

Development of a passive dehumidification system with energy recovery ventilation for residential buildings by using solar energy

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博 士 論 文

Development of a passive dehumidification system
with energy recovery ventilation for residential
buildings by using solar energy

太陽熱を利用した住宅の自然除湿・
全熱交換換気システムに関する研究

2024 年

Yulu Chen

陳雨露

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Chapter 1 Introduction

1.1 Research background

Final energy consumption in the household sector is defined as energy consumption at home, excluding transportation-related energy consumption, such as that of private vehicles. The 2022 Energy White Paper from the Japan Agency for Natural Resources and Energy reports that the household sector accounted for 15.8% of the overall final energy consumption in 2020, amounting to 1908.64 PJ [1]. Household energy consumption is categorized into five areas: cooling, heating, hot water supply, general electricity, and lighting. In 2020, heating and cooling represented 27.5% of the total household energy consumption. Notably, the proportion of energy consumption for cooling has been increasing annually [1]. A study predicts that the cooling demand in 2100 will be 40 times higher than in 2000 [2].

The heat loads on the air-conditioning (AC) system during the cooling season include sensible and latent heat loads. The latent heat load is the energy adsorbed or released by moisture in the air as it undergoes a phase change (from liquid to gas or vice-versa), essentially the energy the AC consumes to condense the indoor air's moisture during summer [3]. Consequently, the indoor humidity level is a crucial factor affecting AC energy consumption in summer; higher indoor humidity leads to greater latent heat loads and increased energy consumption [4]. It has been shown that 40% of a building's cooling load is dedicated to providing the required humidity level in the supply air in hot and humid regions [5]. Moreover, high indoor humidity levels pose several challenges:

1. Dehumidification in conventional residences is primarily achieved through vapor compression air conditioners, which use refrigerants causing significant environmental pollution, including ozone layer depletion and global warming [6].
2. High humidity levels contribute to the proliferation of dust mites, adversely affecting human health and exacerbating conditions such as asthma, allergic symptoms, and airway infections [7–9]; a previous study indicated that relative humidity above 70% in heat ventilation and air conditioning (HVAC) system supply air ducts can rapidly amplify the impact of bacteria, viruses, fungi, and mites [10].
3. Moisture condensation can compromise the durability of building envelopes and lead to the deterioration of building materials [11,12], negatively affecting building performance and the quality of life for occupants [13].

Therefore, dehumidification, as an effective method to reduce latent cooling loads during summer, plays a pivotal role in decreasing building energy consumption and maintaining indoor hygrothermal comfort.

1.2 Previous research of dehumidification technologies

1.2.1 Residential air-conditioning system and integrated desiccant system

Conventional residential air-conditioning systems address both sensible and latent heat loads by cooling the supply air below the dew point temperature to facilitate condensation dehumidification. However, this supercooled air always needs to be reheated to meet the indoor temperature setpoint, an approach that consumes additional energy.

The coefficient of performance (COP) serves as a critical and universal metric for assessing the energy efficiency of AC systems. The Carnot inverse cycle sets the thermodynamic ceiling for the maximum achievable COP in a refrigeration cycle. This is accomplished by either lowering the heat sink temperature (refrigerant condensing temperature) or raising the heat source temperature (refrigerant evaporating temperature) [14]. Guided by this principle, recent research has focused on operating the cooling cycle of indoor units at higher refrigerant evaporation temperatures to attain higher COP values and reduce greenhouse gas emissions [15]. However, a higher refrigerant evaporation temperature diminishes the moisture condensed by the indoor unit, thereby weakening the AC's dehumidification capability. Field studies have demonstrated that buildings solely relying on the condensing dehumidification of conventional AC systems fail to maintain indoor humidity within a comfortable range [16].

The limitations of conventional residential AC systems have prompted researchers to explore advanced technologies for cooling and dehumidifying indoor spaces, such as separate sensible and latent cooling (SSLC) technology [5]. SSLC technology separates temperature and humidity control, eliminating energy wastage and enabling precise, independent management of indoor humidity levels. The sensible heat treatment system in SSLC technology is typically managed using conventional AC systems, such as the vapor compression cycle (VAC) air-conditioning system. The most prevalent technologies for latent heat management are desiccant systems, which extract water vapor from the indoor air by transferring it to desiccant materials through adsorption or absorption [17]. Desiccant materials form the core of these systems and are classified in various ways, including porous solid desiccants, liquid desiccants, and natural fibers such as dried agricultural waste [18].

- Solid desiccant system

Porous solid desiccant systems, known for their satisfactory efficiency, compact structure, and reduced risk of corrosion and carryover, have attracted significant attention from researchers [16]. Common solid desiccant materials include silica gel, zeolite, lithium chloride (LiCl), and molecular sieves [4], which directly adsorb water vapor [2].

Ding et al. [16] conducted a comparative analysis of the annual performance of solid desiccant dehumidifiers, revealing potential energy savings of up to 4.9% and 7.7% for an office and a hotel, respectively, compared to traditional supercooling and reheating condensation dehumidification. Chai et al. [19] introduced a fresh air dehumidification system featuring a heat pump with a desiccant-coated heat exchanger (DCHE) and conducted experimental measurements. The DCHE system achieved a maximum dehumidification effect of 9.98 g/kg (DA) and exhibited a COP of 5.36 for the heat pump. Zhao et al. [20] proposed a novel solid desiccant dehumidification cycle utilizing an internal heat recovery-based sorptive material coated heat exchanger, which could maintain

dehumidified air below 8 g/kg (DA). Ali et al. [20] investigated an integrated absorption-solid desiccant air-conditioning system, demonstrating that the integrated system had a COP 50–55% higher than that of a conventional desiccant system. Eakvanich et al. [21] developed a radiant cooling panel combined with a solid desiccant dehumidifier for tropical climates, showing that their system improved comfort levels (predicted mean vote, PMV), reduced indoor humidity, and saved 16.27% of daily electricity consumption for AC. Dadi et al. [22] assessed the performance of a silica gel-based solid desiccant wheel, including its humidity ratio, adsorption rate, and effectiveness, in Vadodara, Gujarat, India. Their experiments indicated a humidity reduction from 10.138 g/kg (DA) to 5.983 g/kg (DA), approximately a 59.01% decrease. Su et al. [23] conducted a comprehensive analysis of desiccant wheels' dehumidification effectiveness based on operational parameters and desiccant materials, finding optimal performance with a posterior diversion of regeneration air and synthesized metal silicate material. Ahmed et al. [24] examined the impact of various parameters on the performance of a solar desiccant wheel, identifying optimal wheel speeds for each wheel thickness to maximize performance.

Despite these advancements, solid desiccant systems have limitations, including high regeneration temperatures (approximately 90 to 260 °C) [2], elevated material costs, and challenges in achieving high dehumidification rates and stability over prolonged service periods [6].

- *Liquid desiccant system*

In recent years, liquid desiccant systems have garnered significant interest from researchers due to their ability to operate at lower regeneration temperatures using low-grade energy sources such as condensation heat, industrial waste heat, and renewable energy such as solar energy, geothermal energy [2], while also demonstrating relatively high efficiency [16]. Liquid desiccants absorb moisture through chemical and physical processes.

Bhowmik et al. [25] explored a combined system of liquid desiccant dehumidification with a VAC system for energy-efficient AC applications, comparing its performance to a standalone VAC system. Their findings indicated that the integrated system could enhance the COP by 40.8%–74.8% under various conditions. Qi et al. [26] assessed the annual energy consumption of a liquid desiccant cooling system in the hot and humid climate of Hong Kong, revealing that the solar-assisted system could achieve significant energy savings, especially in winter, with the least savings observed in July and August. Liang et al. [27] developed a liquid desiccant dehumidification system combined with shallow geothermal energy, which showed potential power consumption savings of 86% and 82% compared to condensation systems and traditional liquid desiccant dehumidification systems, respectively. Chen et al. evaluated a hybrid system combining liquid desiccant dehumidification and evaporative cooling, noting a 23.5% energy-saving ratio that could reach up to 70% with the utilization of waste heat [28]. Li et al. [29] designed a heat pump-driven liquid desiccant dehumidification system aimed at regulating indoor air temperature and humidity in high moisture load buildings such as natatoriums, achieving a COP increase of 3.19%–10.57% compared to systems without air heating processes. Cui et al. [30] simulated the performance of a carbon dioxide heat pump-driven liquid dehumidification system, finding a 51.67% reduction in power consumption and a 106.89% improvement in the dehumidification effect compared to conventional condensation dehumidification in a VAC system.

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Despite these advancements, liquid desiccant systems face challenges in wider adoption within residential buildings due to issues such as corrosion caused by the liquid desiccant and high investment costs [16,31].

Additionally, desiccant systems encounter inherent limitations difficult to surmount, including the operational process of dehumidification and regeneration, where regeneration involves vaporizing and releasing moisture back into the air, consuming extra energy. Since dehumidification and regeneration cannot occur simultaneously, most desiccant systems cannot provide continuous 24-hour dehumidification, further limiting operational time.

1.2.2 Residential ventilation system

Ventilation plays a crucial role in ensuring good indoor air quality by removing harmful substances such as volatile organic compounds (VOCs), smoke, odors, dust, and bacteria [32]. Recent concerns over the spread of the SARS-CoV-2 virus in confined spaces and the heightened requirements for indoor air quality have underscored that minimum ventilation standards may no longer suffice [33–35]. However, increasing ventilation exacerbates the conflict between maintaining air quality and energy efficiency, as more energy is required to dehumidify fresh air and maintain indoor humidity within the comfort zone [36], given that ventilation and infiltration significantly influence indoor humidity levels [37].

Energy recovery ventilation (ERV) systems, utilized in buildings for heat and moisture recovery between outdoor supply air and indoor exhaust air, mitigate this issue. An ERV system's primary components include a total heat exchanger (for both sensible and latent heat), a supply fan, and an exhaust fan. By recovering energy from one airstream to pre-treat fresh air, ERV systems reduce the load on heating or cooling systems, making AC more economical and efficient. As part of global decarbonization efforts, ERVs are crucial for enhancing energy efficiency in commercial, industrial, and residential buildings [38]. Zhang et al. reported that a membrane-based energy recovery ventilator (MERV) could save energy by accounting for 58% of the ventilation air treatment requirements and reduce the total cooling load by up to 12.4% during Hong Kong's summer [32]. Wu et al. found that an ERV can decrease AC energy consumption by 17.4% [39]. Fan et al. [40] further illustrated that integrating ERV with AC systems and CO₂-demand control can decrease AC loads by 20–30% and outdoor air load energy consumption by 60–70%. Zhang et al. discovered that the use of MERV in Hong Kong could save approximately 58% of the annual energy required for conditioning ventilated air [32].

To enhance dehumidification efficiency in indoor environments, researchers have combined ERV with various dehumidification technologies. This synergy is effective because ERV can limit outdoor moisture intrusion, enhancing the dehumidification system's performance. The more effective the dehumidification, the greater the humidity differential between indoors and outdoors, allowing ERV to recover more moisture and improve its pre-dehumidification effect [10,38,41]. Innovations [42] such as the integration of MERVs with mechanical air dehumidification systems have shown a 33% reduction in primary energy consumption. Chai et al. [43] proposed a DCHE for highly efficient dehumidification, which can be used with a heat pump together, and they found that the optimal average moisture removal and COP of the proposed system are 7.02 g/kg (DA) and 4.10. In addition, the moisture removal effectiveness of the proposal system was 1.60–2.88 times that of the conventional condensation dehumidification method.

1.2.3 Passive dehumidification technologies

Active dehumidification technologies, including AC dehumidification and desiccant dehumidification, face limitations due to issues such as the need for regeneration, corrosiveness, high heat demand, and elevated operation and maintenance costs. As a result, there is a growing interest in passive dehumidification technology, which is celebrated for its efficiency, cost-effectiveness, and simplicity in maintenance. Passive dehumidification is characterized by two main features: 1) it minimizes energy consumption by relying on low-energy fans or pumps and predominantly harnesses natural energy sources; and 2) it leverages building materials with significant moisture adsorption and desorption capabilities. These materials can release or absorb moisture based on the ambient humidity, thus passively regulating indoor humidity levels [44–47]. Thoughtful design and application of moisture buffer materials can notably reduce building energy consumption and enhance indoor hygrothermal comfort [48], with some studies indicating a 30% reduction in cooling energy consumption in systems integrating hygroscopic materials [49–51].

Liu et al. [5] developed a heat and moisture transfer model for a solar-powered ventilated roof desiccant bed system aimed at zero-energy buildings, with simulations showing that the system's regeneration process could remove 80 g of water per unit in approximately 3 h [5]. Ran et al. [52] introduced a passive solar shading and dehumidification system using silica gel as the desiccant material, analyzing the impact of various factors such as airflow velocity and material thickness on desiccant regeneration. Ida [53] proposed a building double-skin system incorporating innovative hygrothermal materials for daylighting blinds within the wall cavity, with a dynamic simulation model demonstrating improvements in annual energy loads and consumption in hot and humid climates. Yang et al. [54] designed a solar-powered dehumidification window for residential use, a solid desiccant packed bed with a photovoltaic panel to offset the electricity used by the fan for ventilation and dehumidification. This system achieved a maximum dehumidification of 7.1 g/kg (DA), a dehumidification efficiency of 58.60%, and a peak power output from the photovoltaic panels of 39.83 W.

The implementation of strategic ventilation and dehumidification materials in attic spaces has proven effective in managing the moisture balance within indoor living environments [55,56]. Abaza et al. innovated a cooling and dehumidification system by optimizing both direct and indirect ventilation within the attic space of buildings [57,58]. The materials used in the attic serve dual purposes: acting as a desiccant for dehumidification at night and as a passive heating collector to regenerate the desiccant during daytime hours. Simulations outcomes indicated that this novel dehumidification approach could slash the overall annual latent cooling load by approximately 83% [58]. Areemit et al. [59,60] introduced a passive room dehumidification system leveraging the moisture buffering properties of wooden attic spaces, demonstrating significant dehumidification effectiveness, particularly under conditions of high daily solar radiation accumulation. Parker et al. [61] developed a night sky cooling system integrated with an enthalpy-based ventilation system in the attic, incorporating solar drying of low-temperature regenerated desiccants. This system showcased the potential to harness solar energy for dehumidification with minimal power input, as evidenced by experimental measurements.

However, these studies have tended to overlook the challenge posed by the intrusion of outside water vapor through ventilation [62]. ERV systems, which facilitate the exchange of energy—

including heat and moisture—between incoming supply air and outgoing exhaust air, are increasingly employed to maintain required ventilation rates while capturing energy-saving benefits. They are frequently combined with energy-efficient technologies, including dehumidification systems, to enhance overall performance. Additionally, it is worth noting that previous research has often relied on the physical properties of water vapor, such as vapor pressure, absolute humidity, or water content, to describe moisture transfer processes. This approach, while useful, offers limited accuracy because, from a thermodynamic perspective, the equilibrium state of water transfer is determined by a complex interplay of temperature, pressure, and concentration factors [63].

To address the challenge of dehumidifying indoor environments efficiently and with minimal energy loss, we propose an innovative indoor hygrothermal environment control system that integrates ERV system with a passive envelope dehumidification system, specifically the passive dehumidification and solar collection (PDSC) system, as introduced by Lee et al. [64]. This system facilitates comprehensive passive dehumidification and radiative cooling during summer months, while enabling solar heating in the winter, using renewable energy sources [65]. Additionally, the PDSC system effectively manages indoor humidity—dehumidifying in summer and preventing over-drying in winter—by using the moisture adsorption and desorption properties of fibrous insulation in the roof, coupled with air circulation between the indoor space and the roof ventilation layer. Lee et al. [64] confirmed the effectiveness of combining the PDSC system with exhaust ventilation equipment, demonstrating a reduction in relative humidity by approximately 5%–11% and absolute humidity by approximately 1.5–3.2 g/kg (DA) compared to a conventional house. Simulation results further indicated that when paired with an exhaust ventilation system, the PDSC system could decrease the latent heat load by 80%–95% across various regions, in comparison to conventional residential settings [66]. Notably, Japanese residential buildings typically maintain a ventilation rate of 0.5 air changes per hour (ACH) to ensure indoor comfort and health. However, employing the PDSC system alongside an exhaust ventilation strategy throughout the day introduces the challenge of admitting hot and humid outdoor air directly into the indoor environment during summer, resulting in an increased ventilation load for cooling. Therefore, integrating the PDSC system with ERV in a balanced ventilation system is expected to significantly enhance the efficiency of dehumidification and latent heat load reduction.

Consequently, this study introduces a merged PDSC and ERV system, designated as the PSE (PDSC and ERV) system, to alleviate the heightened latent cooling load attributable to ventilation within the PDSC framework. The ERV component of the advanced integrated PSE system capitalizes on the humidity differential established by the PDSC system between the indoor and outdoor settings. It exchanges moisture between incoming supply and outgoing exhaust air streams, thereby ensuring the provision of fresh air with reduced humidity during ventilation.

1.3 Main contents of this study

In this study, we explore the summer passive dehumidification and energy-saving performance of the PSE system, using both measured and simulated approaches.

Chapter 2 delves into the thermodynamic energy that drives moisture transfer, known as water potential, providing a comprehensive explanation. It outlines the basic theory behind the existing roof dehumidification system, the PDSC system, which is driven by water potential. After summarizing the limitations of current PDSC systems, the chapter proposes the PSE system—a hybrid of the PDSC system and an ERV. The air circulation paths of the PSE system during both daytime and nighttime are then detailed.

Chapter 3 presents an in-depth look at a residential building equipped with a PSE system, covering the climate of the target region, construction and specifications of the building, and details on the equipment used. To assess the dehumidification performance of the PSE system during the hot and humid summer months, actual experiments were conducted in an experimental house under various ventilation systems (exhaust ventilation, ERV, and PSE systems). The experiments were divided based on whether air-conditioning cooling was considered, with indoor temperatures and humidity levels (both absolute and relative) monitored under three different system operations in these conditions.

Chapter 4 focuses on simulations conducted using the high-accuracy THERB for HAM software. Three residential models (exhaust ventilation, ERV, and PSE systems) were established and to validate their simulated data against experimental values. Five case simulations (Cases A through E) were performed. Case A analyzed indoor temperature and humidity for the four systems (exhaust, PDSC, ERV, and PSE) during summer without air-conditioning. Case B looked into the indoor climate under constant temperature and humidity settings for cooling. Due to limitations in calculating indoor air-conditioning power consumption and condensate moisture content with THERB for HAM alone, the software was coupled with a heat pump characteristic model for validation. Case C through E involved simulations of indoor climate, load, and power consumption for the four systems under various conditions, employing the coupled simulation method. Case D involved simulating the PSE system's indoor temperature, humidity, load, and power consumption under high airflow conditions using a coupled simulation method. Case E examined the indoor temperature, humidity, load, and power consumption of PSE systems under varying air conditioner performance and airflow conditions, also employing the coupled simulation approach. These simulations were conducted considering both the presence and absence of the building envelope's moisture adsorption and desorption properties.

Chapter 5 concludes the study by summarizing the key findings from the preceding chapters, discussing the limitations of this study, and outlining future research directions.

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Chapter 2 Passive dehumidification system combined with energy recovery ventilation system

2.1 Basic theory of passive dehumidification

Existing theories on moisture transfer typically consider water vapor pressure (or absolute humidity) as the driving force of water flux. However, from a thermodynamic perspective, a system reaches thermodynamic equilibrium when the temperature, pressure, and chemical potential of its components are uniform throughout. Therefore, water vapor pressure alone is insufficient as a driving force for moisture transfer if mass transfer occurs in a non-equilibrium state. The chemical potential serves as an indicator of mass balance conditions, implying that mass transfer involves concentration gradients and forces between flowing molecules and their surroundings. In essence, the driving force for moisture transfer is a function of temperature, pressure, and concentration. Therefore, this study utilizes thermodynamic energy (chemical potential) to describe moisture transfer within a system.

2.1.1 Driving force of moisture transfer

Air is a multi-component system, but the phase transition of water does not alter the composition of other components, allowing water in moist air to be treated as a hypothetical single-component system with coexisting gas-phase and liquid-phase water. Consequently, the driving force for moisture transfer in an isobaric field is the chemical potential of water, known as the water potential. Water potential [1], is thus considered the driving force for moisture transfer, based on non-equilibrium thermodynamic energy, facilitating the combined analysis of heat and moisture movement. Water potential applies a chemical potential to moisture (1 kg of water vapor) as an indicator of mass equilibrium conditions (thermodynamic state quantity). It is defined as a function of water vapor concentration, internal energy of water vapor molecules, forces acting on them (such as adhesive power or capillary action), and field energy (external stresses). At standard atmospheric pressure and a given temperature T and vapor pressure p , water potential μ_w is expressed as the sum of the saturated water potential μ_w^0 and unsaturated water potential μ using Eqs. (2.1–2.3):

$$\mu_w(p, T) = \mu_w^0(T) + \mu(p, T) \quad (2.1)$$

$$\mu_w^0(T) = 6.44243 \times 10^5 + c_{p,w}(T - 273.15) \quad (2.2)$$

$$\begin{aligned} & -Tc_{p,w} \ln \frac{T}{273.15} + R_w T \ln \frac{p_s}{1.01325 \times 10^5} \\ \mu(p, T) &= R_w T \ln \frac{p_w}{p_s} \end{aligned} \quad (2.3)$$

where T_o and p_o represent the standard temperature [0 °C (273.15 K)] and pressure [1 atm (1.01325×10^5 Pa)], respectively; p_w and p_s denote the vapor pressure and the saturated vapor pressure of the humid air, respectively; $c_{p,w}$ is the specific heat in J/(kg K); and $R_w = 461.50$ J/(kg K), calculated by dividing the gas constant $R = 8.31441$ J/(mol K) by the molecular weight of water

18.016×10^3 kg/mol. Using standard temperature T_o , pressure p_o , and standard enthalpy ($h_{o,w}$) and entropy ($s_{o,w}$), the Eq. (2) can be simplified as $h_{o,w} - Ts_{o,w} = 6.44243 \times 10^5$.

Figure 2.1 illustrates the principal relationship among water potential (left y-axis), temperature (x-axis), relative humidity (diagonal lines from lower left to upper right), and absolute humidity (lines sloping downward to the right) [2]. The saturated water potential represents the thermodynamic energy in saturated water vapor air at a given temperature. The unsaturated water potential indicates the work done when air at vapor pressure p_w is compressed to saturated vapor pressure p_s [3], symbolizing the negative energy reflecting the dryness of humid air from saturated status. Therefore, the water potential under unsaturated conditions of humid air (or dry air) is lower than in the saturated state at the same temperature; that is, unsaturated humid air (or dry air) has a high potential for water vapor gain. If non-equilibrium thermodynamic energy exists within a system (i.e., a water potential difference), the deviation from a homogenous state will propel water vapor flow, diffusing moisture from higher to lower water potential areas until thermodynamic equilibrium is achieved.

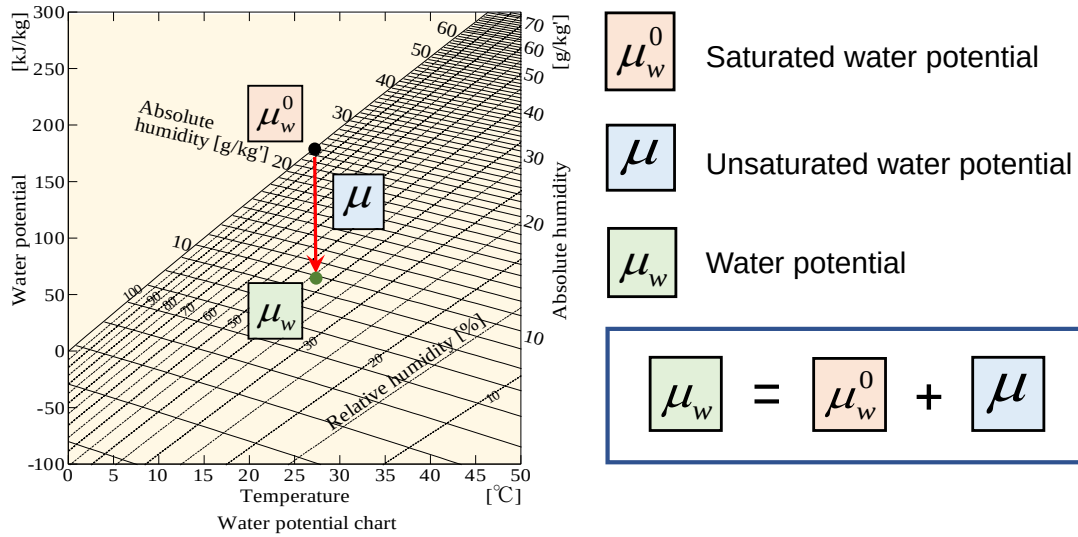


Figure 2.1 Water potential chart portraying the relationship between water potential, temperature, relative humidity, and absolute humidity [4]

Additionally, when considering the effects of gravity and external forces, the force water potential is denoted as follows:

$$\mu_f = gz + p\overline{V}_m \quad (2.4)$$

where g represents gravitational acceleration, z is the height of the water from a reference position, $\overline{V}_m$ is the volume per unit weight of water, and $p\overline{V}_m$ is equivalent to $R_w T$. Eqs. (2.1–2.4) together constitute the complete equations capable of expressing the potential energy of moisture.

2.1.2 Heat and moisture transfer model by considering water potential as driving force

Unsteady-state heat transfer modeling of building walls typically addresses convective and radiative heat transfer in one-dimensional multilayer flat walls using the concepts of total heat transfer coefficients and equivalent air temperatures for simplified heat flow calculations. If physical phenomena are to be accurately modeled, the surface heat transfer layer should be distinguished from the wall's heat conductive system, which, in turn, must be integrated with other systems via the wall surface heat balance equation. In this approach, the heat balance equation for each indoor surface acts as the junction between the respective wall surface heat transfer system and the room temperature fluctuation system, described by a simultaneous equation involving each indoor surface temperature and indoor air temperature as variables. Nonetheless, the wall heat transfer systems rely on certain assumptions:

- (1) Each wall layer is considered homogeneous, with constant specific heat, specific gravity, and thermal conductivity.
- (2) Moisture transfer within the walls is disregarded.
- (3) Walls are presumed to be planar, with the temperature gradient only occurring in the direction normal to the surface. Multidimensional heat transfer at corners and in irregularly shaped sections is not accounted for.

Consequently, the wall heat transfer system simplifies to basic heat transfer, excluding latent heat transfer. Most thermal environment and heat load analysis software approximate the effects of hygroscopicity and desorption by adding a virtual humidity (latent heat) capacity to the indoor air, due to the complexities involved in analyzing indoor temperature and humidity fluctuations considering wall hygroscopicity and desorption. This is challenging because humidity transfer is a nonlinear phenomenon influenced by heat, air (temperature, concentration, and external forces), and varying physical properties depending on moisture content. Nonetheless, some software such as THERB for HAM, can simulate the combined movement of heat and moisture in a building, providing detailed analysis of the temperature and moisture environment. The P-model, as proposed in THERB for HAM, accounts for the water potential difference as a driving force for moisture transfer. It fully integrates heat and moisture in calculations, considering both internal energy and external forces such as gravity. This model forms the basis for accurately predicting moisture movement within the building envelope and between building materials and the surrounding air [5]. By using the P-model, the hygrothermal environment of the entire building and its ventilation layer within the envelope can be accurately evaluated.

- *Conductive heat and moisture transfer balance*

The conductive heat and moisture transfer balance equations are presented as Eqs. (2.5) and (2.6), with the process illustrated in Figure 2.2.

Heat balance equation:

$$\frac{\partial C_p T}{\partial t} + C_{lw} j_{lw} \nabla T = \nabla \lambda \nabla T + r_v \nabla \lambda'_g (\mu_w + \mu_f) \quad (2.5)$$

Moisture balance equation:

$$\rho_{lw} \frac{\partial \phi}{\partial \mu} \frac{\partial \mu}{\partial t} = \nabla \lambda'_g \nabla (\mu_w + \mu_f) + \nabla \lambda'_l \nabla (\mu + \mu_f) \quad (2.6)$$

where C and ρ represent specific heat and specific weight of water in hygroscopic materials, respectively. C_{lw} , ρ_{lw} , and j_{lw} denote the specific heat, specific weight, and flux of liquid-phase water, respectively. λ symbolizes thermal conductivity, while λ'_g and λ'_l indicate the conductivities for gaseous and liquid-phase water, corresponding to gradients μ_w and μ . r_v is heat of sorption, which is the latent heat of evaporation. The term ϕ stands for the material's equilibrium water content, with $\partial \phi / \partial \mu$ illustrating the water capacity, which is the gradient of the tangent to the equilibrium water content-unsaturated water potential curve. This gradient reveals the extent of moisture adsorption and desorption by a material, expressed in (kg/kg)/(kJ/kg).

In the heat balance equation, the initial term on the right side accounts for heat transfer, while the subsequent term considers the latent heat effects stemming from moisture movement. The moisture balance equation's initial and subsequent terms on the right side correspond to gaseous and liquid moisture conductance, respectively.

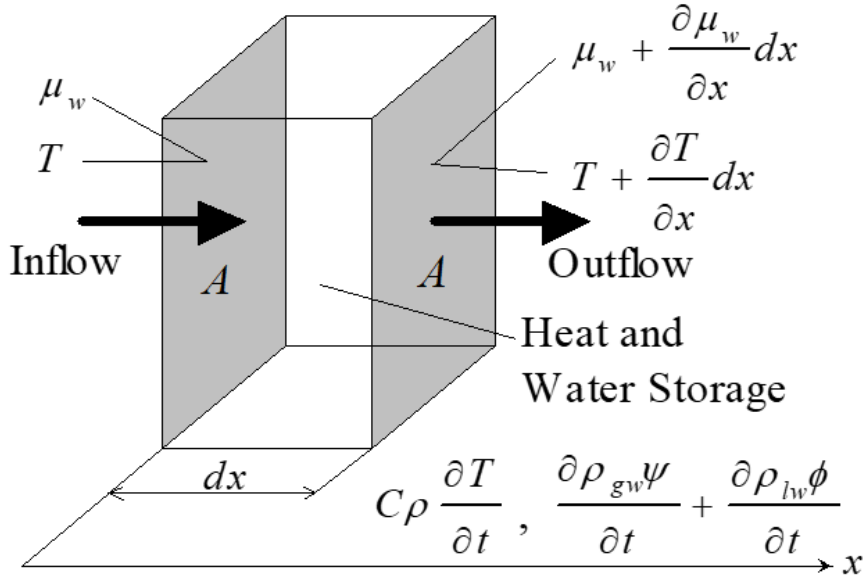


Figure 2.2 Schematic of conductive heat and moisture transfer of P-model

- Heat and moisture transfer balance in ventilation layer

Equations (2.7) and (2.8) showcase the balanced equations of the P-model for computing the heat and water transfer flux within the ventilation layer, taking into account water potential. Figure 2.3 depicts the diagram of heat and moisture transfer in the ventilation layer. Under the assumption that heat and humidity are instantaneously and uniformly diffused, temperature and humidity levels are derived from convective heat and moisture transfer (the first term on the right), along with heat and humidity variations due to airflow inside the ventilation layer (the second and third terms on the right), respectively.

Heat balance equation:

$$V_{vol,c} \cdot c\gamma \frac{dT}{dt} = \sum_{j=1}^2 S_{c,j} \cdot \alpha_{c,j} (T_j - T_c) + V_{af} \cdot c\gamma (T_o - T_c) \quad (2.7)$$

Moisture balance equation:

$$V_{vol,c} \cdot r_{w,c} \frac{d\mu_c}{dt} = \sum_{j=1}^2 S_{c,j} \cdot \alpha'_{\mu,j} (\mu_{w,j} - \mu_{w,c}) + V_{af} (\gamma_{w,o} \cdot \mu_o^* - \gamma_{w,c} \cdot \mu_c) \quad (2.8)$$

where $c\gamma$ represents volumetric specific heat of air, $V_{vol,c}$ denotes the volume of ventilation layer c , $S_{c,j}$ and $\alpha_{c,j}$ denote the surface area and heat convective transfer coefficient of ventilation layer c , respectively. T_j and T_c are the surface and air temperatures of ventilation layer c , respectively, and T_o is the outdoor temperature. $r_{w,c}$ signifies water capacity of ventilation layer c , $\gamma_{w,o}$ and $\gamma_{w,c}$ represent the water vapor specific weight of the outdoor surface and ventilation layer, respectively. $\alpha'_{\mu,j}$ is the moisture convective transfer coefficient, $\mu_{w,j}$ and $\mu_{w,c}$ are the surface and air water potentials of ventilation layer c , respectively. μ_o^* denotes the unsaturated water potential of the outdoor air.

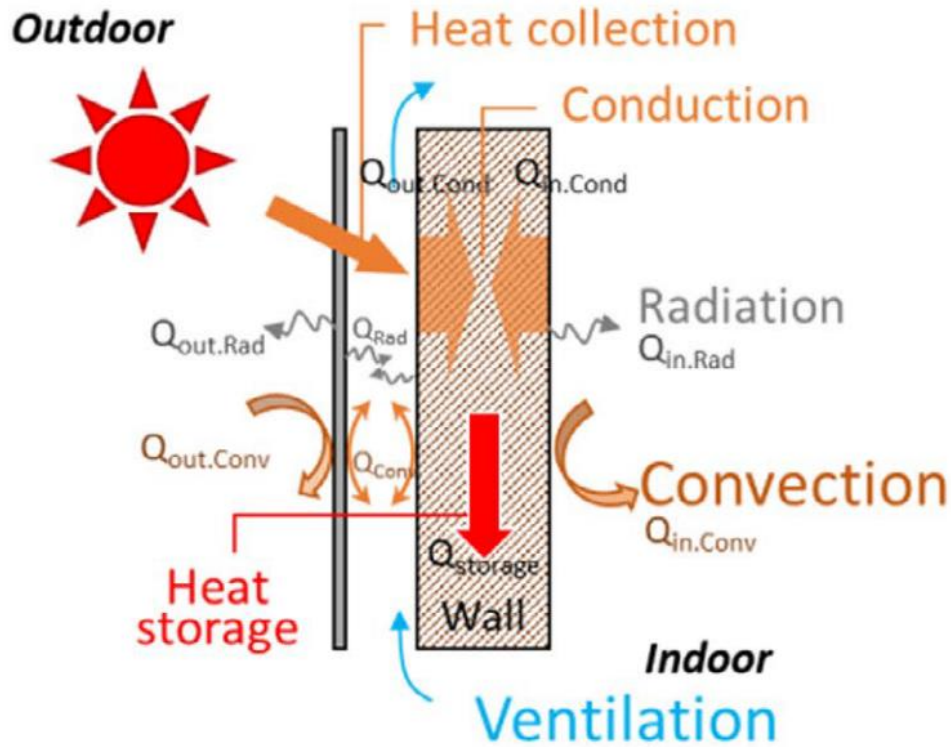


Figure 2.3 Schematic of heat and moisture transfer in the ventilation layer [6]

Chapter 2 Passive dehumidification system combined with energy recovery ventilation system

Utilizing high water capacity insulation and ventilated layers within the building envelope's construction can effectively regulate indoor humidity levels [6]. When exposed to the atmosphere, hygroscopic materials engage in moisture exchange—either adsorption or desorption—with the surrounding air. Consequently, these materials can adsorb moisture from the interior environment and then transfer it through the ventilated layer to the exterior. Once equilibrium between the moisture in the material and that in the surrounding atmosphere is achieved, the material's water content stabilizes over time.

2.1.3 Conceptual diagram of roof passive dehumidification during day and night

Our research team has developed a passive dehumidification roof system grounded in non-equilibrium thermodynamics, leveraging the concept of water potential and the heat and moisture transfer phenomena between building hygroscopic materials and the surrounding air. This subsection will elaborate on the fundamental principles of moisture transfer within a roof passive dehumidification system. A key component of the passive dehumidification and solar collection (PDSC) system is the ventilated roof. The thermal insulation, made of fibrous insulation material, possesses a high water capacity, enabling it to adsorb and desorb moisture. Between this fibrous insulation and the roof's exterior material is a ventilation layer that facilitates air movement from indoors. Air circulation from indoor to outdoor (or vice versa) through the roof ventilation layer can be achieved with a low-power fan. The roof environment's temperature fluctuates in response to solar radiation or radiative cooling, causing the fibrous insulation to desorb moisture through solar heating or adsorb moisture during radiative cooling phases. Temperature and humidity changes affect the water potential of the roof components. Given a constant indoor temperature and humidity, the water potential within the room remains unchanged, leading to a water potential difference between the indoor environment and the roof, thus driving moisture movement.

Figure 2.4 illustrates the detailed variation of water potential in the roof component and indoors for the PDSC system during summer, with Figure 2.4(a) showing a one-day cycle. Schematic diagrams depict moisture movement in daytime mode (Figure 2.4(b)) and nighttime mode (Figure 2.4(c)), assuming constant indoor temperature and relative humidity (dry-bulb temperature = 27 °C, relative humidity = 60%). These conditions are recommended for comfortable indoor environments as suggested by the Building Health Law in Japan [7])

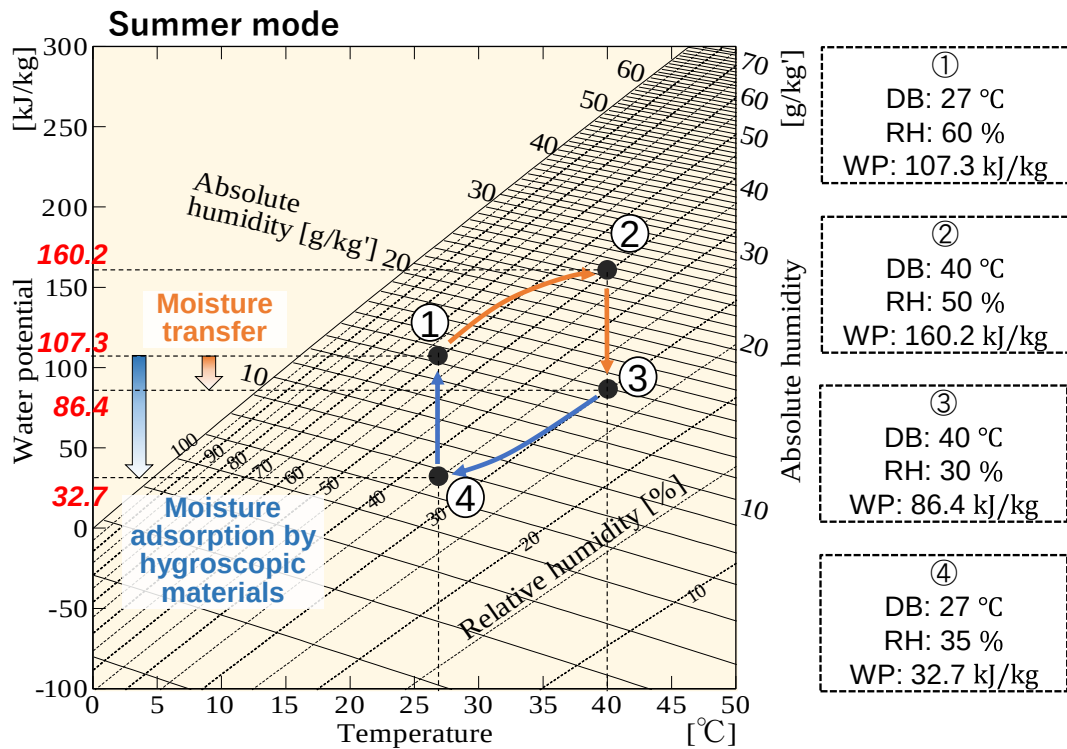
- *Day mode:*

During day mode (moisture desorption and water vapor transfer process), solar irradiation heating causes the roof ventilation layer's temperature to rise, leading to the desorption and diffusion of moisture molecules from the fibrous insulation to the roof ventilation layer. Consequently, the absolute humidity of the roof ventilation layer gradually increases, altering the water potential (Figure 2.4 $\Phi \rightarrow \textcircled{2}$: dry-bulb temperature = 40 °C, relative humidity = 50%, based on assumed values from our previous study [2]). Upon initiating air circulation in day mode, indoor air enters the roof ventilation layer, equating its absolute humidity with that of the indoor air and thereby reducing the roof ventilation layer's water potential (Figure 2.4 $\textcircled{2} \rightarrow \textcircled{3}$: dry-bulb temperature = 40 °C, relative humidity = 30%, based on assumed values from our previous study [2]). This creates a water potential difference between the indoor space and the roof ventilation layer (Figure 2.4 Φ and $\textcircled{3}$), facilitating the spontaneous transfer of moisture flux from the indoors to the roof ventilation layer. Ultimately, the transferred water vapor is expelled outdoors along with the flowing air through the roof ventilation layer.

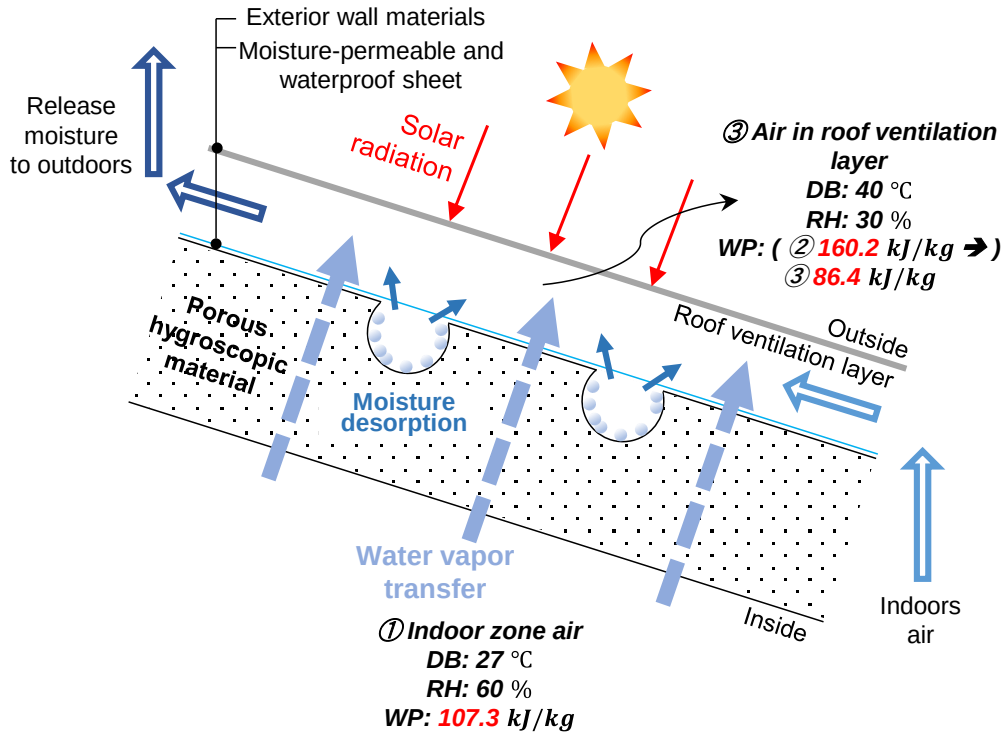
- *Night mode:*

During night mode, which is characterized by moisture adsorption, the temperature in the roof ventilation layer drops due to radiative cooling, promoting moisture adsorption by the fibrous insulation. This results in a decrease in the absolute humidity of the roof ventilation layer (including the fibrous insulation). The water potential of the roof components drops as both temperature and absolute humidity decrease (Figure 2.4 ③→④: dry-bulb temperature = 27 °C, relative humidity = 35%, based on assumed values from our previous study [2]). Consequently, the roof components' water potential becomes lower than that of the indoors, causing moisture to flow from the indoor environment to the roof due to the water potential difference (Figure 2.4 ①→④). The roof gradually become moist due to moisture adsorption, with its water potential slowly increasing until it reaches equilibrium with the indoor water potential (Figure 2.4 ④).

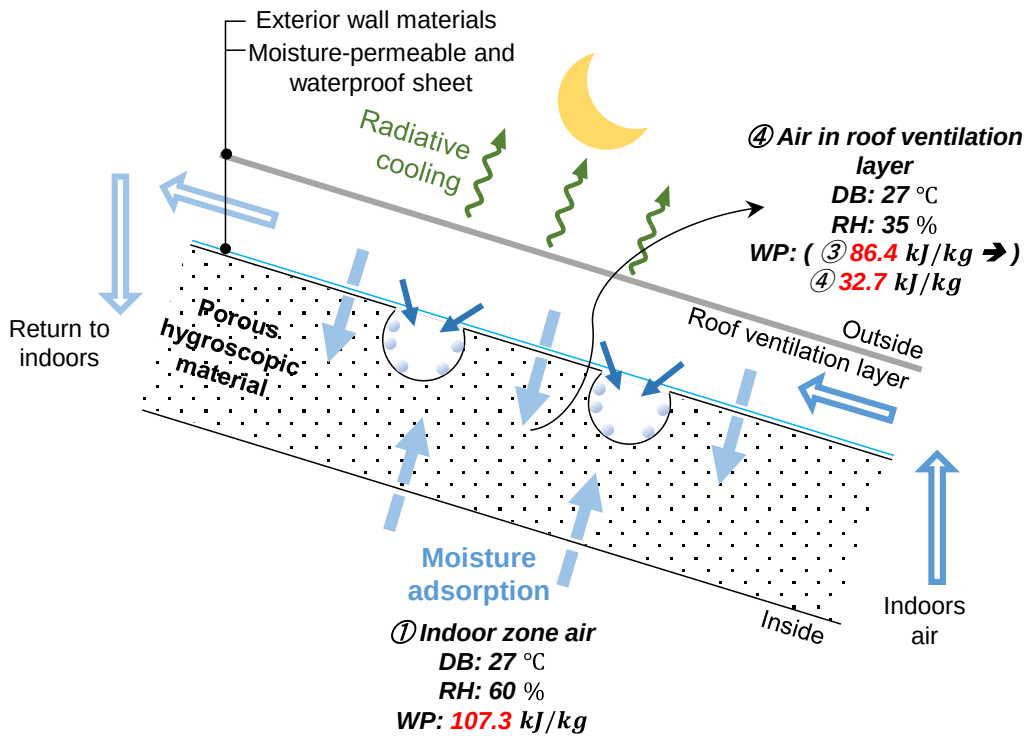
In summary, during the day, advection brings air into the roof ventilation layer, and the thermodynamic energy of the roof is diminished due to the increased dryness from solar radiation heating. Moisture is transferred from indoors to the roof ventilation layer and subsequently expelled to the exterior. Conversely, at night, moisture is adsorbed from indoors, and the thermodynamic energy of the roof components gradually equilibrates with the indoor environment.



(a) Variation in water potential with respect to temperature, water potential, relative humidity, and absolute humidity over one-day



(b) Schematic of moisture desorption and moisture transfer in day mode



(c) Schematic of moisture adsorption in night mode

Figure 2.4 Variation in water potential and moisture movement for one-day of passive dehumidification and solar collection (PDSC) in summer

2.2 Passive dehumidification system installed on roof of residential building

To assess the PDSC system's effectiveness for dehumidification and radiant cooling during the hot and humid summer months, our research team constructed two full-scale residential buildings for comparative testing—one equipped with the PDSC system and the other a conventional Japanese house [8–10]. The results demonstrated that the house with the PDSC system experienced a reduction in both relative and absolute humidity by approximately 11% and 3.2 g/kg (DA), respectively, compared to the house without the system during the summer season, thereby confirming the dehumidification efficiency of the PDSC system.

This subsection elaborates on the PDSC system's air circulation pathway, differentiated into daytime and nighttime modes.

2.2.1 Moisture desorption and transfer during daytime

Figure 2.5 illustrates the daytime air circulation pathway for the PDSC system installed in a residential building during the summer [8].

- (1) During the day, the roof ventilation layer's temperature increases due to solar radiation, leading to moisture desorption from the fibrous insulation to the roof ventilation layer, thereby increasing both temperature and absolute humidity within the layer.
- (2) When the roof ventilation layer's air temperature exceeds the indoor air temperature by more than 1 °C, Fan A activates to channel indoor air into the roof ventilation layer. As indoor air traverses the roof ventilation layer, its water potential becomes lower than that of the room. This process equilibrates the absolute humidity levels between the indoor environment and the roof ventilation layer, facilitating moisture transfer to the roof ventilation layer. Venting this air outdoors helps mitigate temperature increases and simultaneously dehumidifies the space.
- (3) The water vapor transferred from indoors is expelled outdoors along with the air through the fan unit.
- (4) Concurrently, Fan C operates to circulate dehumidified air from the attic space back into the living area, effectively reducing the air conditioning system's latent heat load.
- (5) An opening between the living space and the attic ensures the circulation of indoor air, with Fan C promoting the distribution of air into the attic for a uniform indoor air distribution.
- (6) The air conditioner processes the dehumidified air in the attic before distributing it throughout the rooms, thus reducing the air conditioning system's latent heat load. The central air conditioning system ensures a consistent distribution of temperature and humidity across all rooms.

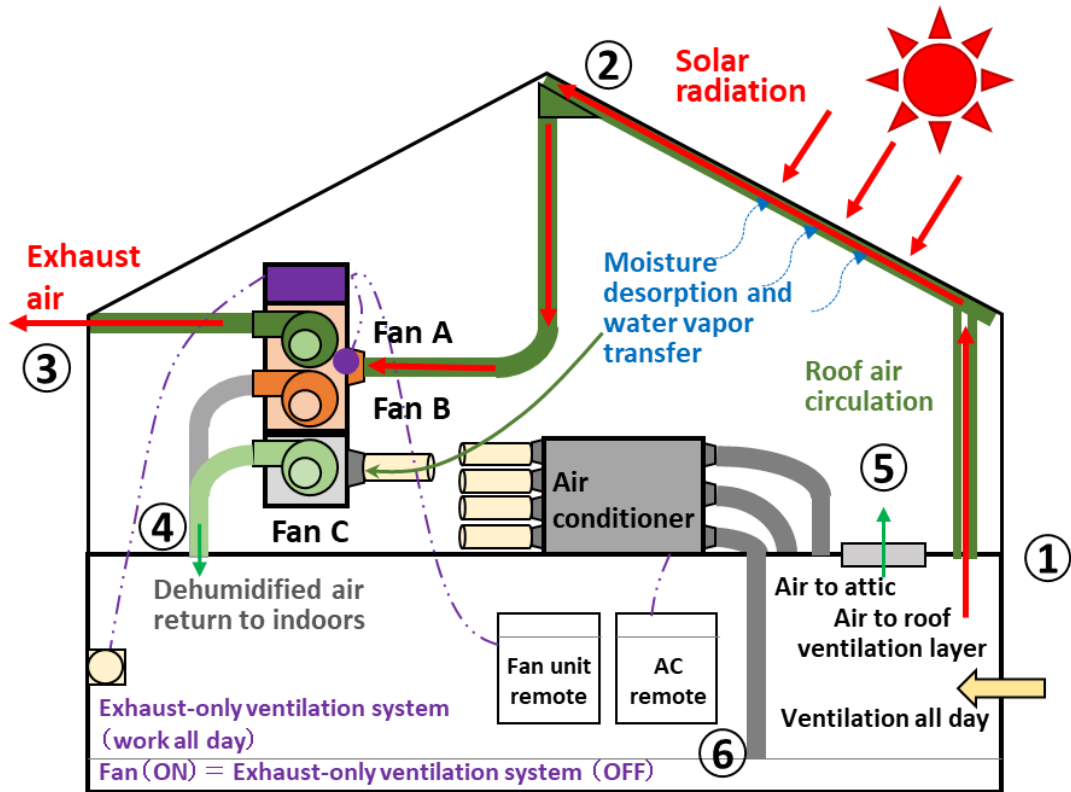


Figure 2.5 Air circulation pathway for PDSC system installed in a residential building during the daytime in summer [8]

2.2.2 Moisture adsorption and radiative cooling at night

Figure 2.6 shows the nighttime air circulation pathway for the PDSC system installed in a residential building during the summer [8].

- (1) At night, the roof ventilation layer's temperature decreases due to radiative cooling, prompting the hygroscopic insulation within the roof component to adsorb moisture, reflecting the ambient temperature drop.
- (2) When the roof ventilation layer's air temperature falls below the indoor temperature by more than 1 °C, Fan B is activated, pulling indoor air into the roof ventilation layer. This circulated air is cooled and dehumidified by the fibrous insulation material through moisture adsorption, attributable to the temperature drop.
- (3) Subsequently, the cooled and dehumidified air from the roof ventilation layer is reintroduced into the living space, aiding in the regulation of the indoor thermal environment.
- (4) With the air conditioner in the attic operational, it directly treats the cooled and dehumidified attic air, thereby reducing both the sensible and latent heat loads.

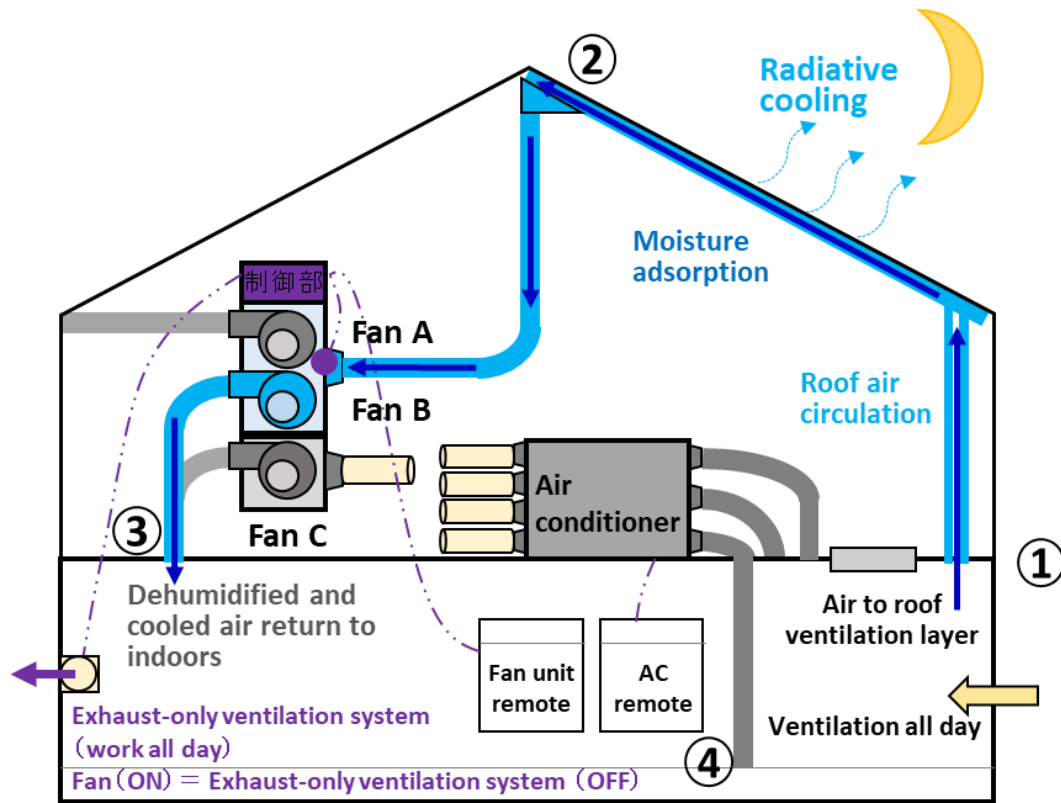


Figure 2.6 Air circulation pathway for PDSC system installed in a residential building at night in summer [8]

Unlike conventional dehumidification systems that necessitate regeneration for desiccant materials, the PDSC system facilitates continuous dehumidification, operating round-the-clock. It is noteworthy that during daytime, the PDSC system dehumidifies by venting outdoors with an approximate ventilation rate of 0.5 ACH. This process hinges on comparing the temperature of the roof ventilation layer with the indoor temperature to determine operation feasibility. Should the PDSC fail to meet operational criteria, an exhaust ventilation system located in the toilet activates for immediate ventilation. At night, this exhaust system continues to operate independently of the PDSC system's status, ensuring uninterrupted 24-hour ventilation. Natural vent openings in each room of the PDSC house guarantee consistent ventilation.

2.3 Passive dehumidification system combined with energy recovery ventilation system installed in residential building

It is crucial to recognize that Japanese building codes mandate continuous ventilation in residential buildings at a constant rate of 0.5 ACH throughout the day. By integrating the PDSC system with an ERV system capable of recovering energy (heat and moisture), air conditioning cooling loads can be further reduced.

2.3.1 Energy recovery ventilation system

Building loads, aside from internal gains, mainly stem from envelope losses or gains and ventilation loads [11]. The increasing demand for superior indoor air quality has amplified buildings' ventilation loads. Employing heat recovery technology is the most effective strategy for mitigating ventilation loads. An ERV is an air-to-air system that efficiently recovers energy (heat and moisture) from one air stream to another, known as enthalpy recovery processes [12]. Differing from sensible heat recovery ventilators, an enthalpy recovery ventilator supports heating and cooling, preventing over-drying in winter and aiding in summer dehumidification. A typical ERV structure, as shown in Figure 2.7, consisting of an energy exchange core and two fans, pre-treats fresh air.

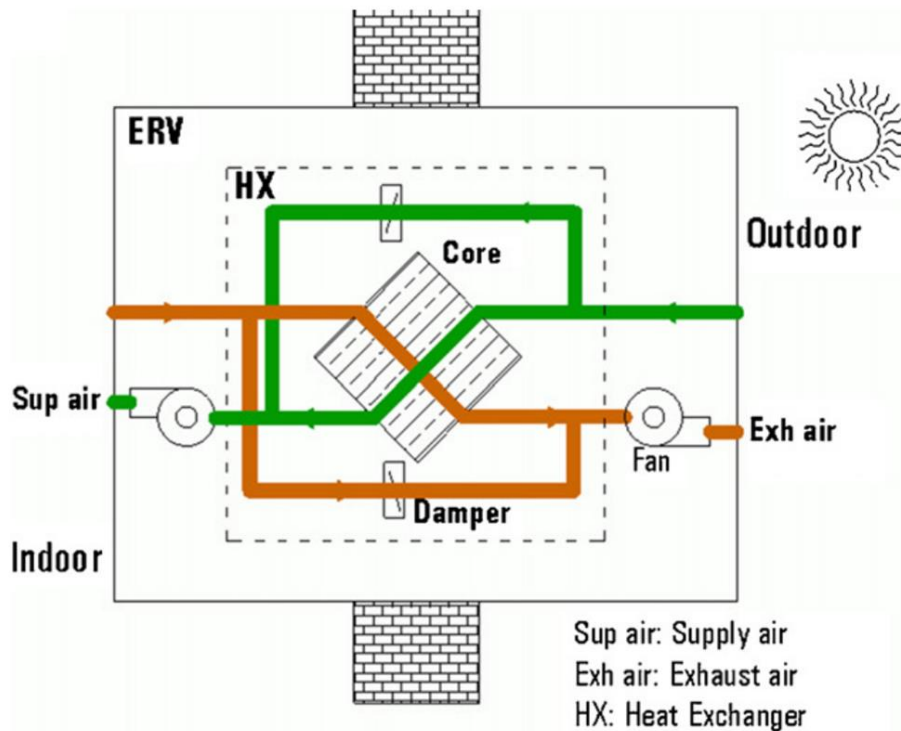


Figure 2.7 Structure of an ERV [13]

ERV performance evaluation indicators include heat or absolute humidity or enthalpy recovery coefficients, calculated from the inlet and outlet air's temperature, absolute humidity, and flow rate. According to ASHRAE Standard 84 [14], the sensible, latent, and total heat exchange coefficients of ERV systems are defined based on the conditions of supply, return, and outdoor air [15].

Sensible heat exchange coefficient:

$$\varepsilon_s = \frac{T_{OA} - T_{SA}}{T_{OA} - T_{RA}} \times 100\% \quad (2.5)$$

Latent heat exchange coefficient:

$$\varepsilon_l = \frac{W_{OA} - W_{SA}}{W_{OA} - W_{RA}} \times 100\% \quad (2.6)$$

Total heat exchange coefficient:

$$\varepsilon_t = \frac{h_{OA} - h_{SA}}{h_{OA} - h_{RA}} \times 100\% \quad (2.7)$$

where ε_s , ε_l , and ε_t are sensible, latent, and total heat exchange coefficients, respectively. T , W , and h represent the temperature, absolute humidity, and enthalpy of the measured air, respectively. The subscript OA signifies outdoor air, SA signifies supply air, and RA signifies return air.

The effectiveness of heat and moisture recovery varies with the climate and indoor environment. A significant absolute humidity difference between indoor and outdoor ($W_{OA} - W_{RA}$) environments, coupled with a lower W_{SA} , enhances the ERV's pre-dehumidification impact, as shown in Eq. (2.6).

This paper introduces the PSE (PDSC combined with ERV) system to improve dehumidification efficacy. The PDSC system's capacity to reduce indoor absolute humidity leverages the ERV's ability to pre-humidify incoming fresh air further. This synergy between the PDSC and ERV systems results in a pronounced dehumidification effect.

In the PSE system, the ERV facilitates ventilation and air circulation, linking the roof component with the indoor and outdoor environments, and conducting moisture and heat exchange between supply and return air. This exchange reduces the latent and sensible load of incoming fresh air, thereby saving energy when the air conditioning system utilizes the preconditioned supply air, achieving lower latent and sensible heat loads. Same as the PDSC system, the PSE system operates in distinct day and night modes.

2.3.2 Daytime mode for PSE system

Figure 2.8 illustrates the daytime air circulation pathway for the PSE system installed in a residential building during summer:

- (1) During the day, solar radiation increases the roof's temperature. When the air temperature in the roof ventilation layer exceeds the indoor temperature by more than 1 °C, the ERV's EA fan activates, pulling indoor air into the roof ventilation layer.
- (2) The water potential difference between the indoor environment (attic) and roof ventilation layer causes moisture from the indoor air to transfer toward the roof ventilation layer, dehumidifying the air in the attic.
- (3) The EA fan then expels the hot air, now carrying the transferred moisture, directly from the roof ventilation layer to the outdoor without any energy exchange, achieving approximately 0.25 ACH of indoor air through the PDSC system.
- (4) Concurrently, the ERV conducts heat and moisture exchange between incoming outdoor air and outgoing indoor air via a separate mechanical ventilation system (exhausting air at approximately 0.25 ACH). The return indoor air is sourced from the living space on the second-floor, allowing the ERV to mitigate increases in indoor water vapor and heat.
- (5) Vents are maintained in the ceiling of the second-floor living space, enabling air exhausted by the ERV from the second-floor to circulate back to the attic, promoting even distribution of indoor air.
- (6) The PDSC system dehumidifies the attic air, while the ERV supplies the attic with fresh air that is both low in humidity and temperature. The attic-installed air conditioner directly processes this cooled and dried air, reducing its sensible and latent heat load before distributing it uniformly to each room, ensuring consistent indoor temperature and humidity levels.

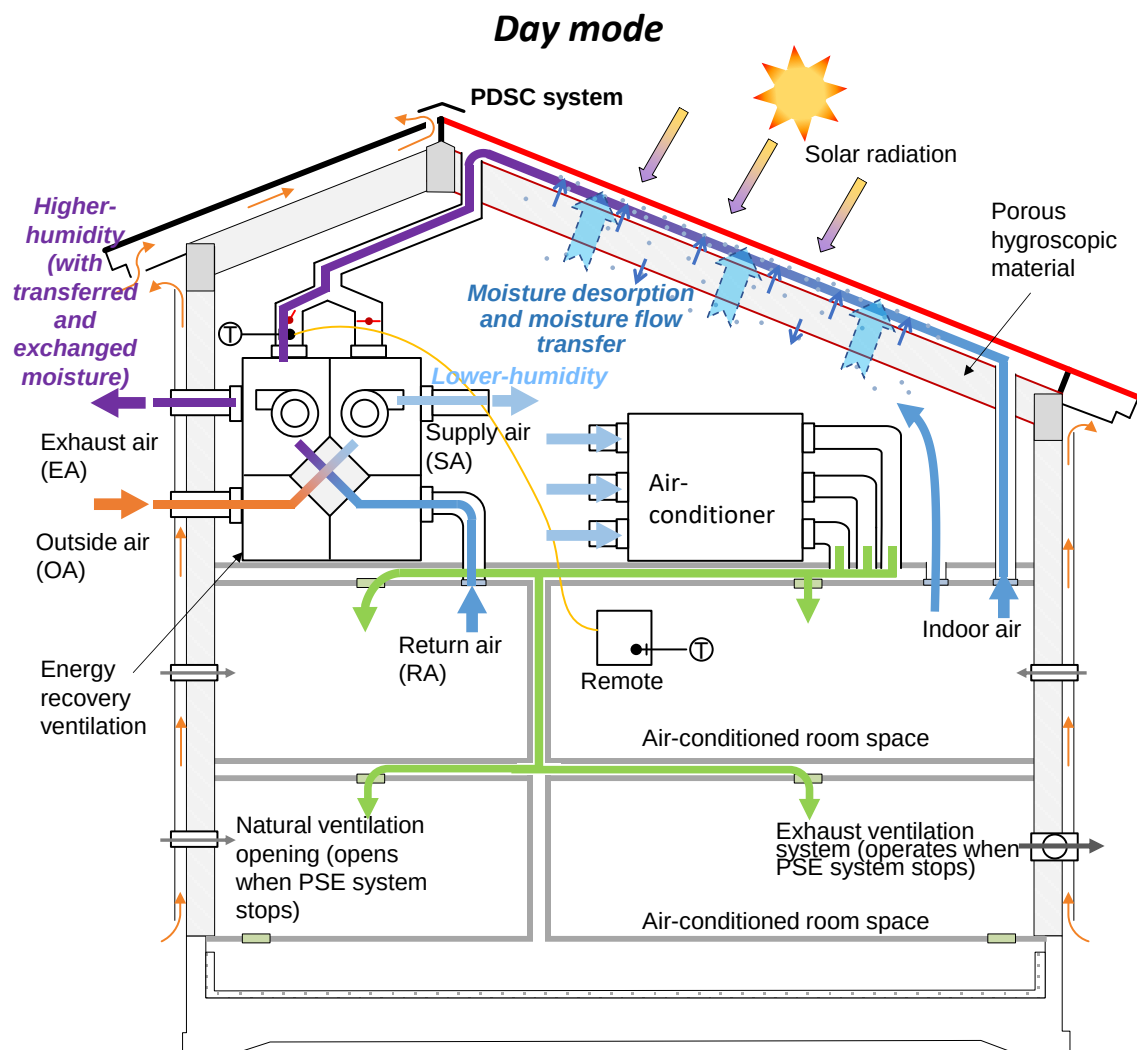


Figure 2.8 Air circulation pathway for PSE system installed in a residential building during the day in summer [16]

2.3.3 Nighttime mode for PSE system

Figure 2.9 depicts the nighttime air circulation pathway for the PSE system in a residential building during summer:

- (1) At night, radiative cooling decreases the roof's temperature. If the air temperature in the roof ventilation layer drops below the indoor temperature by more than 1 °C, the ERV's SA fan activates, drawing indoor air into the roof ventilation layer.
- (2) The roof's hygroscopic materials, dried during the day, start to adsorb moisture from the air entering the roof ventilation layer due to the radiative cooling effect.
- (3) The SA fan then channels the cooled and dried air from the roof ventilation layer directly into the room without any energy exchange.
- (4) Simultaneously, the ERV exchanges heat and moisture between the outdoor and indoor air while performing ventilation (at approximately 0.5 ACH), helping to control the rise in indoor water vapor and heat and maintaining indoor thermal comfort while lessening the air conditioning system's sensible and latent heat loads.
- (5) Vents in the ceiling of the second-floor living space allow air exhausted by the ERV from the second-floor to return to the attic, facilitating even air distribution throughout the indoor space.
- (6) Through the PDSC system's dehumidification and radiative cooling, coupled with the ERV's supply of low-humidity and low-temperature fresh air to the attic, the attic-installed air conditioner directly treats this air, further diminishing its sensible and latent heat load. This treated air is then distributed uniformly across rooms, ensuring a consistent indoor thermal environment.

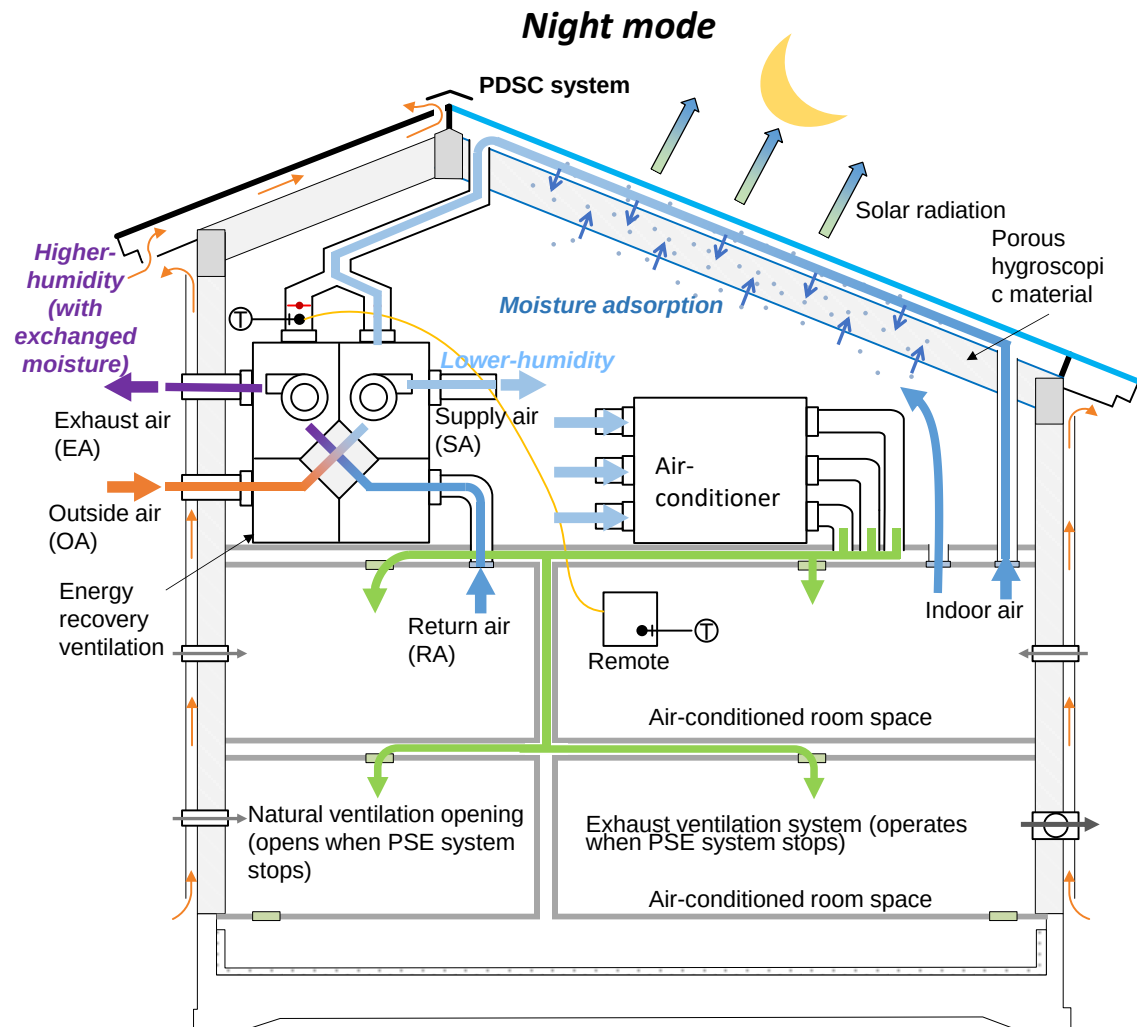


Figure 2.9 Air circulation pathway for PSE system installed in a residential building at night in summer [16]

2.4 Reference

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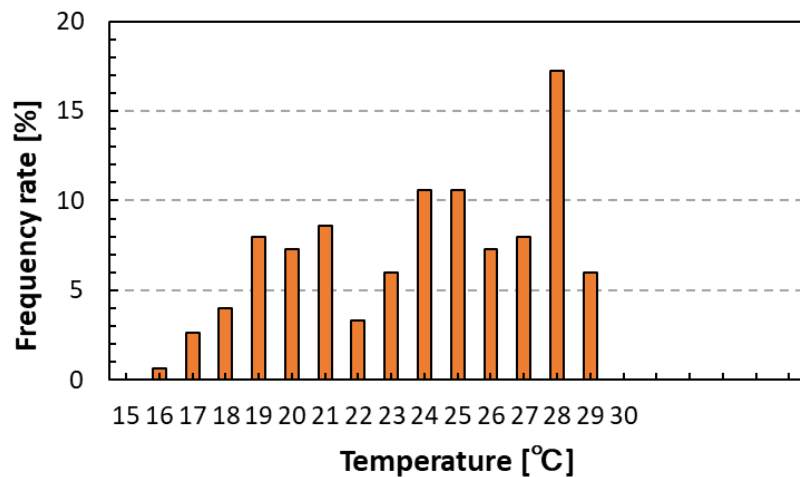
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Chapter 3 Measurement analysis on hygrothermal environment of detached house with PSE system

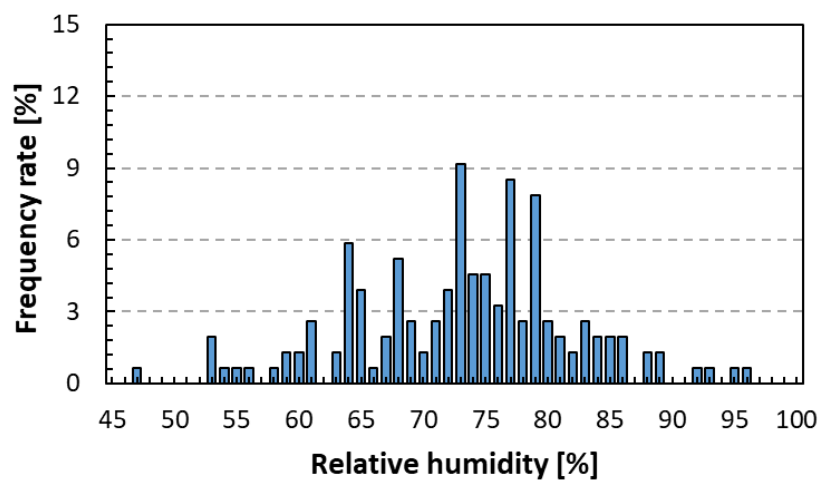
3.1 Reference residential building outline

3.1.1 Climate of target region

A wooden-built house in Shimonoseki City, Japan, served as the subject of our study. Shimonoseki City, characterized by a humid subtropical climate (Cfa) according to the Köppen climate classification [1]. Figure 3.1 illustrates the daily average summer temperature and relative humidity in Shimonoseki City over a 30-year period (May–September, 1991–2020) [2]. It can be seen that the weather was hot, with the highest frequency of daily mean temperatures at approximately 28 °C and relative humidity typically between 70–80%, occasionally reaching up to 95%. The climate data underscores the significant need for latent cooling in this region during the summer months.



(a) Average outdoor temperature



(b) Average outdoor relative humidity

Figure 3.1 Average daily outdoor air temperature and relative humidity in Shimonoseki City during summer (May–September, 1991–2020) [2]

3.1.2 Building construction and specifications

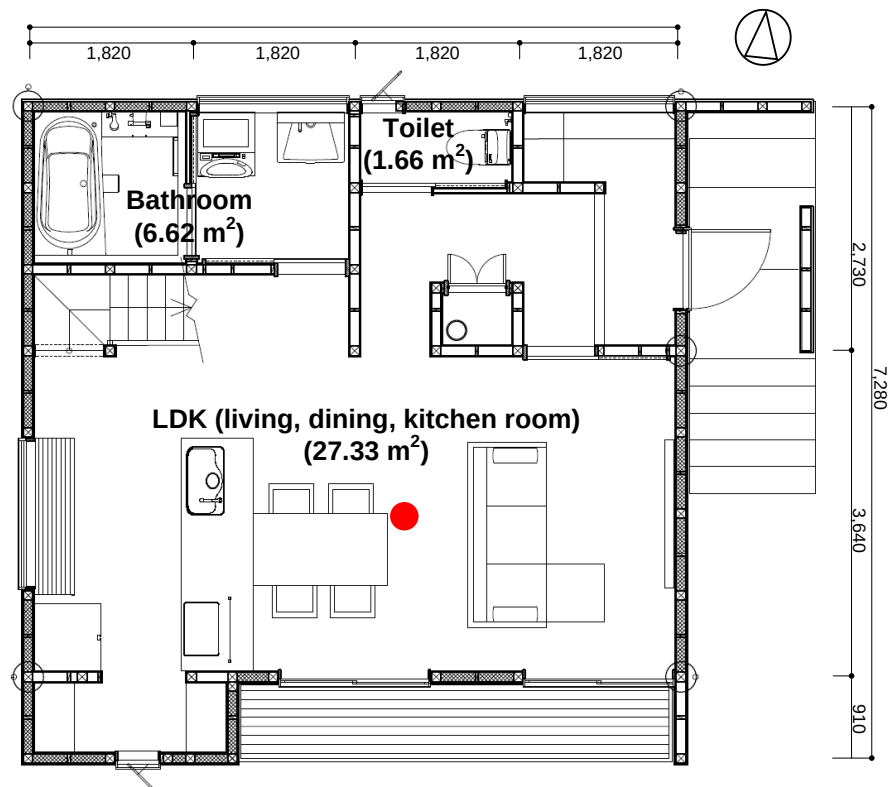
Table 3.1 provides detailed information and descriptions of the experimental house. Exterior photos and floor plans are presented in Figures 3.2 and 3.3. The PDSC roof area, highlighted by dashed red frames in Figures 3.3(b) and (c), spans 25.17 m². Notably, the building envelope lacks a vapor barrier, implying that indoor humidity fluctuations are solely attributable to ventilation and moisture interaction with building materials. Figure 3.4 illustrates a cross-sectional view of the experimental house, indicating all temperature and humidity measurement locations.

Table 3.1 Descriptions of the experiment house

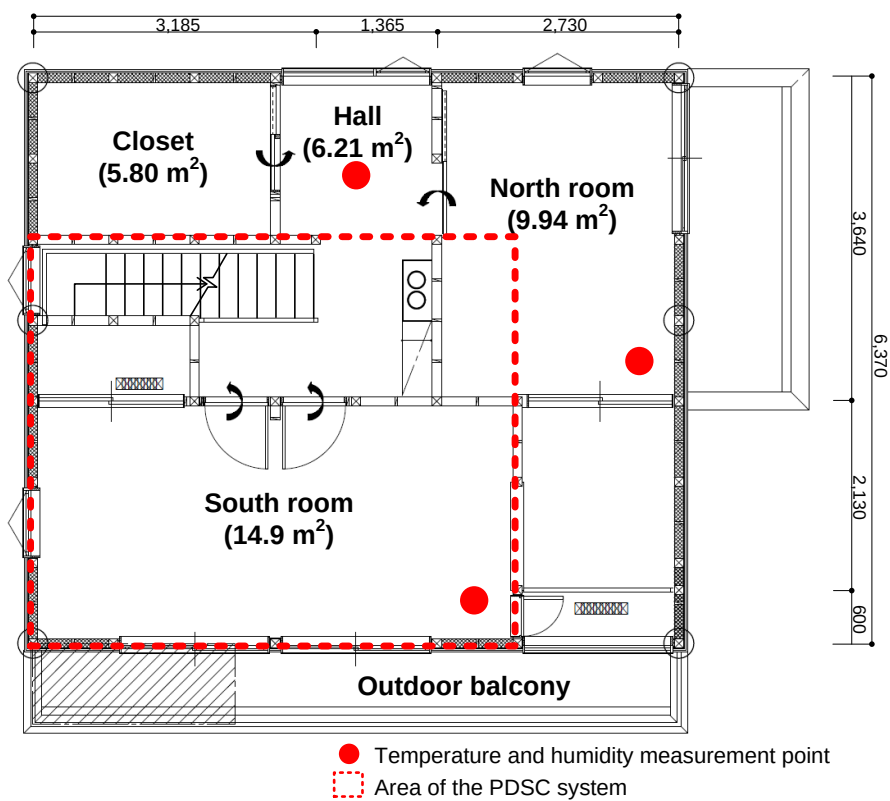
Items	Contents
Location	Shimonoseki City, Yamaguchi Prefecture, Japan
Total floor area	90.93 m ²
height	First floor: 3.150 m; Second-floor: 2.800 m; Second-floor ceiling to ridge: 2.020 m.
Insulation of Windows	Low-e double pane glazing; total window area: 24.578 m ² ; WWR (window-to-wall ratio) = 15.7%.



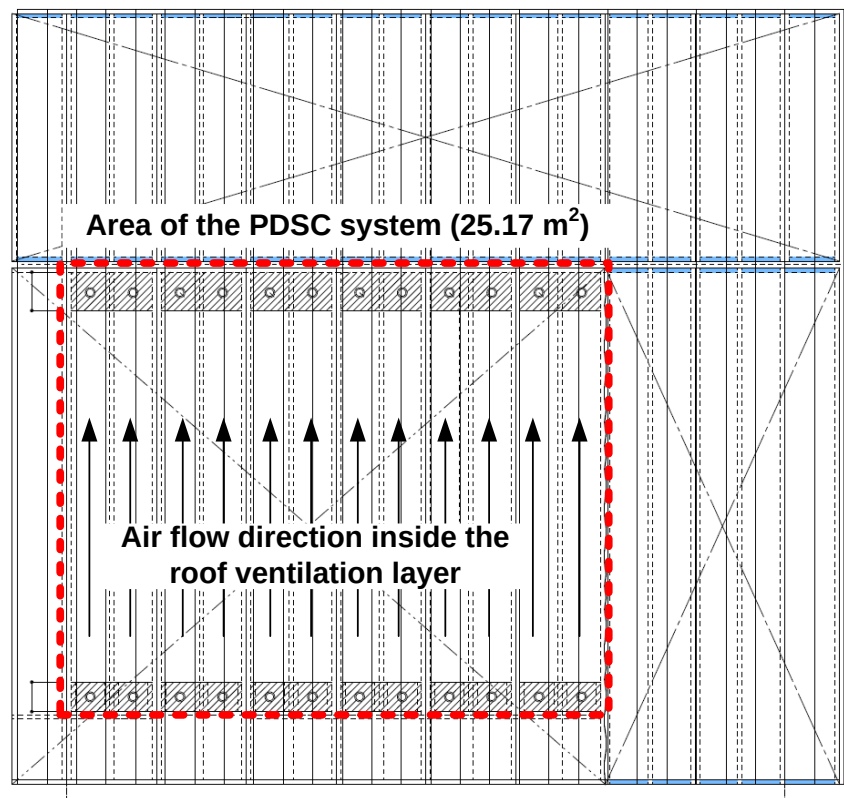
Figure 3.2 Photograph of the target experimental house



(a) First floor plan



(b) Second-floor plan



(c) Roof plan

Figure 3.3 Plans of the experimental house

- Temperature and humidity measurement point by thermo-hygrometer
- Voltage measurement point by voltmeter

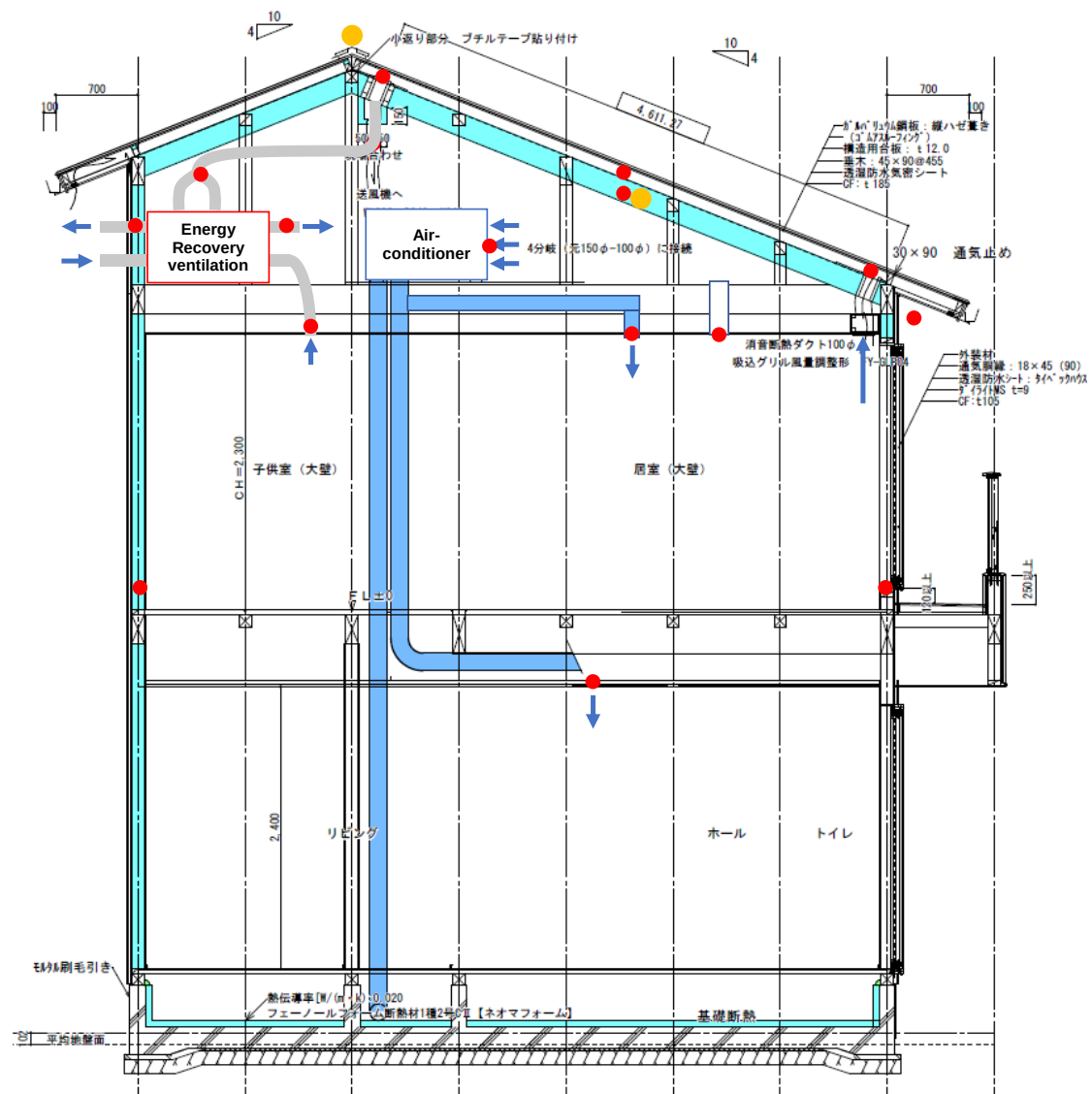


Figure 3.4 Cross-section of the experimental house

Cellulose fiber insulation (CFI), selected for its porous hygroscopic properties and thermal insulation capabilities, was utilized within the envelope, including the PDSC system's air circulation roof. CFI's ability to adsorb and desorb significant amounts of moisture is crucial for managing indoor humidity levels [3]; its physical properties, relevant for simulation purposes, are documented in Table 3.2. The experimental house features a ventilation layer within its external walls and a standard roof, designed to prevent interstitial condensation and maintain indoor humidity control over the long term [4].

Table 3.2 Physical properties of cellulose fiber insulation

Items	Contents
Thickness (m)	In roofs: 0.185, in walls: 0.105
Thermal conductivity [W/ (m K)]	0.039
Specific heat [J/ (kg K)]	1880.0
Specific weight (kg/m ³)	35.0
Moisture conductivity [kg/ (m s Pa)]	1550e-10
Moisture capacity [kg/ (m ³ (kJ/kg))]	3.219e-02

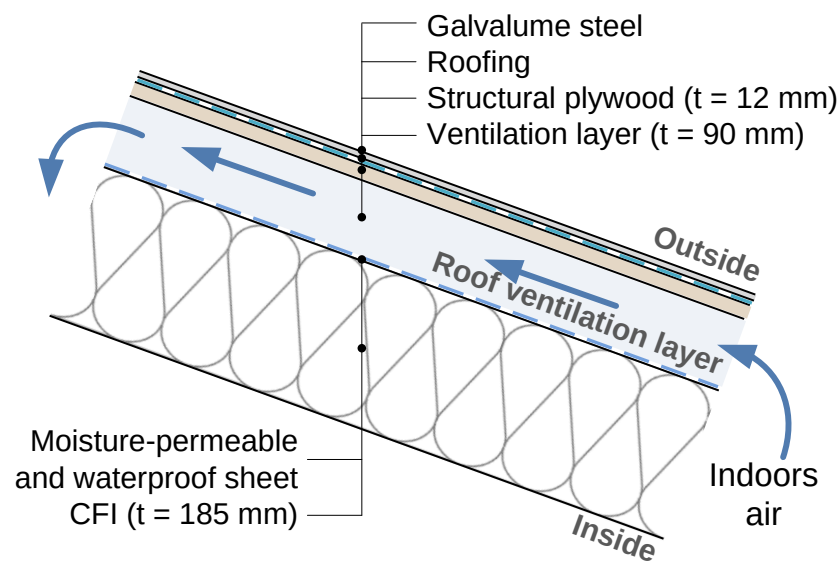
Table 3.3 outlines the experimental house's building envelope structure and materials. It is an airtight, well-insulated structure with U-values of 0.20, 0.32, and 0.39 for the roof, exterior walls, and ground floor, respectively, indicating its energy efficiency in insulation.

Table 3.3 Building envelope structure of the experiment house

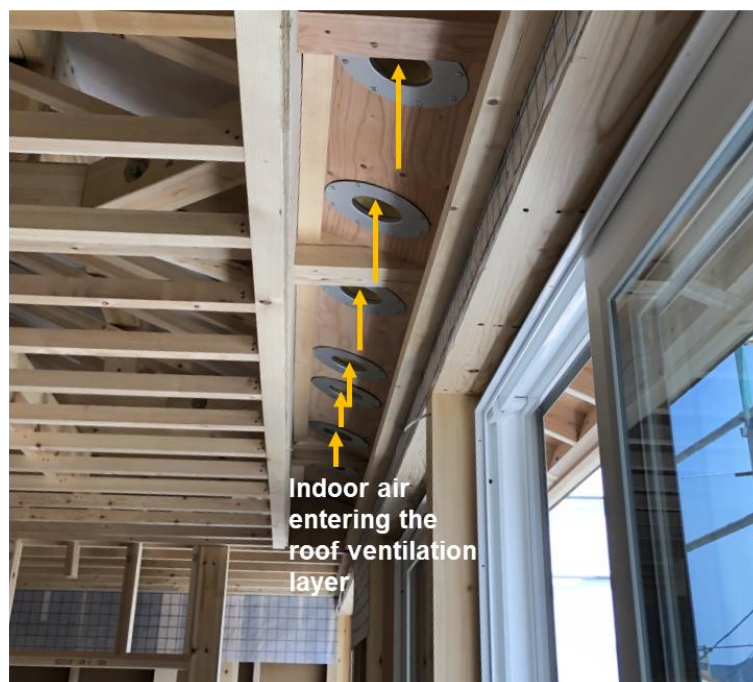
Items	Contents
Roof [area = 49.95 m ² ; U-value = 0.20 W/(m ² K)]	Inner to outer: cellulose fiber insulation (t = 0.185 m), moisture-permeable and waterproof sheet, roof ventilation layer (t = 0.090 m), structure plywood (t = 0.012 m), roofing, galvalume.
Exterior wall [area = 156.77 m ² ; U-value = 0.32 W/(m ² K)]	Inner to outer: interior finishing material, plasterboard (t = 0.012 m), cellulose fiber insulation (t = 0.105 m), structural plywood (t = 0.009 m), moisture-permeable and waterproof sheet, ventilation layer (t = 0.018 m), exterior finishing material.
Ground floor [area = 48.44 m ² ; U-value = 0.39 W/(m ² K)]	Inner to outer: phenolic foam insulation (t = 0.050 m), concrete (t = 0.160 m).
Floor	Top to bottom: flooring (t = 0.015 m), plywood (t = 0.012 m), air space (t = 0.235 m), plater board (t = 0.012 m).

3.1.3 Equipment specifications

Figure 3.5 depicts a section of the roof and interior photographs of the ventilation layer within the PDSC system. The system alternates between daytime (6 a.m. to 6 p.m.) and nighttime modes (6 p.m. to 6 a.m.), with the daytime operation triggered when the air temperature in the roof ventilation layer exceeds the indoor air temperature by more than 1 °C. Conversely, nighttime operation commences when the roof ventilation layer's air temperature falls below the indoor temperature by more than 1 °C. Air circulation is assessed every 20 min for temperature checks before achieving PDSC system, with a test circulation mimicking the day mode and lasting 1 min to ensure accurate air temperature measurements by the EA fan's built-in thermistor from the roof ventilation layer.



(a) Roof section of experimental house



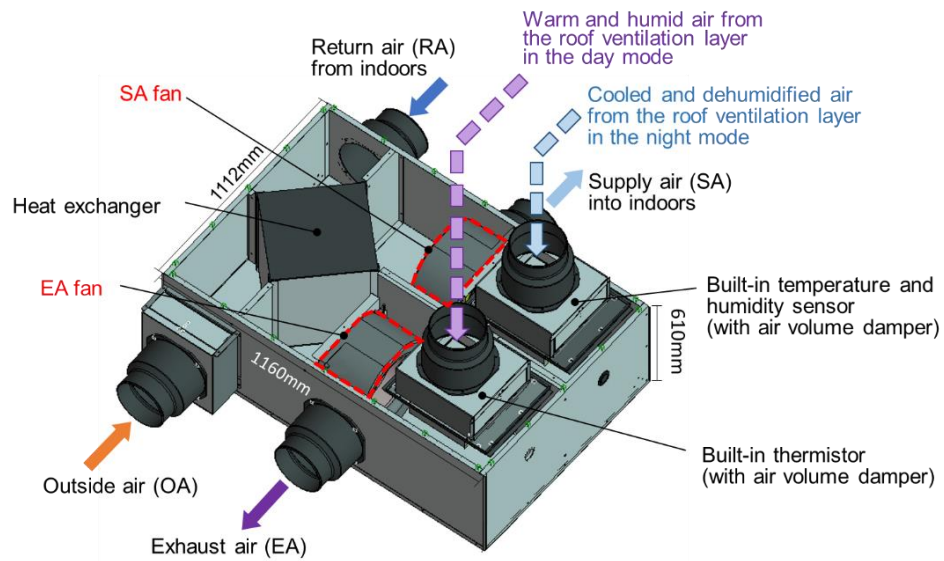
(b) Roof ventilation layer inlet



(c) Roof ventilation layer outlet

Figure 3.5 Roof sections and site photographs

Figure 3.6 presents a composition diagram of the ERV within the PSE system, featuring a heat exchanger core capable of exchanging both water and heat, the SA fan, and the EA fan. These components collectively manage the circulation of air within the roof and facilitate whole-house mechanical ventilation. Additionally, the experimental house includes an exhaust ventilation system, consisting of an exhaust fan in the toilet and natural ventilation openings in each room, to serve as a basis for comparison in the experiment.



(a) ERV's construction



(b) ERV's photograph before installation



(c) ERV's photograph after installation

Figure 3.6 Composition diagram and photographs of ERV unit

Figure 3.7 illustrates the airflow schematic of the ERV during the PSE system's operational period. The ERV's ventilation rate is maintained at 0.5 ACH (approximately $170 \text{ m}^3/\text{h}$). Notably, the airflow through the ERV's heat exchanger is unbalanced during the day due to the direct exhaust of roof air to the outdoors, bypassing heat exchange. During the day, ventilation rates for the roof and ERV are set at approximately $85 \text{ m}^3/\text{h}$ each. At night, both the ERV and roof air circulation rates increase to approximately $170 \text{ m}^3/\text{h}$.

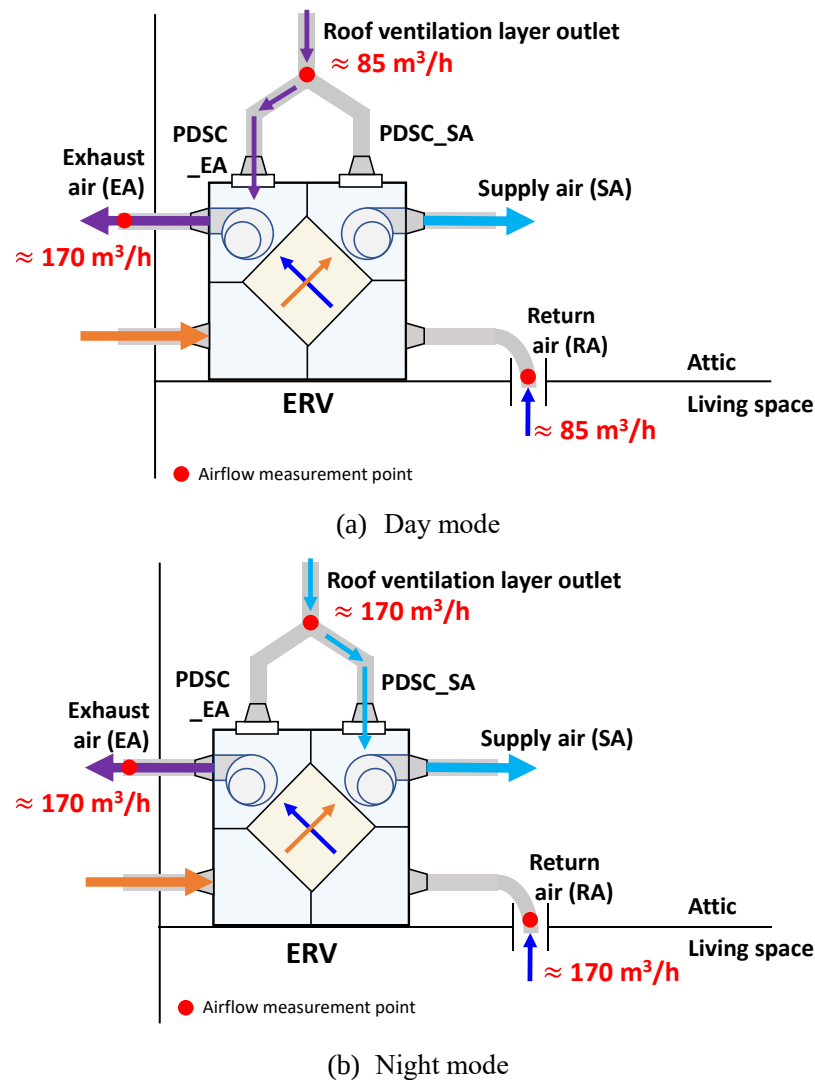


Figure 3.7 Schematic of ERV airflow during operating period of the PSE system

Throughout the measurement period, the temperature and relative humidity inside the experimental house were monitored using high-precision temperature-humidity sensors at 1-min intervals, boasting an accuracy of $\pm 0.3 \text{ }^{\circ}\text{C}$ and $\pm 5\%$ relative humidity (at $25 \text{ }^{\circ}\text{C}$, 50% relative humidity). These sensors were strategically placed in main rooms, air inlets and outlets of the ERV, and within the roof ventilation layer. Solar radiation data was collected at 1-min intervals using a pyranometer installed on the roof.

3.2 Measurements without operation of air-conditioning system

To evaluate the basic dehumidification performance of the PSE system, a series of controlled experiments were undertaken under unconditioned conditions, comparing it against common ventilation systems found in Japanese homes—exhaust ventilation and ERV systems.

3.2.1 Measurement conditions

Table 3.4 details the descriptions of the measurement cases. Exhaust ventilation, frequently found in bathrooms and kitchens, represents the standard system in Japanese residences. Meanwhile, the adoption of ERV system has grown, spurred by the push toward zero-energy buildings. Therefore, the study aimed to assess the dehumidification efficiency of the PSE system relative to these traditional systems. Case 1 featured the operation of an exhaust ventilation system; Case 2 involved the use of an ERV system; and Case 3 focused on the PSE system's operation. The experiments took place in May and June, coinciding with Shimonoseki's rainy season, a period marked by high outdoor humidity levels, allowing for the observation of indoor humidity fluctuations in a humid outdoor environment. Throughout the measurement period, the air conditioner was set to circulate air without cooling to prevent any impact from air-conditioning condensation and dehumidification on the experimental results. This approach also aimed to maintain consistent temperature and humidity levels across all rooms. Additionally, there were no internal moisture or heat gains recorded during the experiment.

Table 3.4 Measurement conditions for Case A-1, A-2, and A-3 without air-conditioning

Cases	System	Measurement period	Indoor conditions	Ventilation rate and equipment
Case 1	Exhaust ventilation system	May 18–30, 2022	No cooling but air circulation mode of air conditioner. No internal moisture, and heat gain	125 m ³ /h (rated data), ventilation fan: Panasonic FY-24CDTK8
Case 2	ERV system	May 31–June 15, 2022		Approximately 170 m ³ /h (measured data), ERV
Case 3	PSE system	June 16–22, 2022		Refer to Figure 3.7

3.2.2 Measurement results of dehumidification effect

Figures 3.8, 3.9, and 3.10 illustrate the changes in temperature, absolute humidity, and relative humidity during three sunny days representative of the operation periods of exhaust ventilation, ERV, and PSE systems, respectively, during the test period.

Case 1 (exhaust ventilation system): Early summer, without indoor air-conditioning, leads to indoor temperatures slightly exceeding outdoor levels—with outdoor air approximately 20 °C and indoor air approximately 25 °C. The indoor temperature peaks occur later than those outdoors, highlighting the experimental house's effective thermal insulation. Indoor absolute humidity trends follow temperature variations, increasing during the day and decreasing at night due to the high moisture capacity of the CFI material and the lack of a vapor barrier, allowing the envelope to modulate room moisture in response to ambient temperature changes (Figures 3.8(a) and (b)). Therefore, given the moisture buffering effect of the building materials mentioned, the relative humidity in the room can remain relatively stable, even when operating under the exhaust ventilation system (Figure 3.8(c)). Despite the introduction of humid fresh air by the exhaust system, causing a mean absolute humidity difference of 0.27 g/kg (DA) between LDK and outdoor air (Figure 3.7(b)), the moisture buffering effect stabilizes relative humidity approximately 45% with minimal fluctuations (Figure 3.8(c)). The relative humidity fluctuates slightly, with a difference between day and night of approximately 5%.

Case 2 (ERV system): Activation of the ERV, when indoor temperatures are higher than outdoor, (as shown in Figure 3.8(a)), facilitates sensible heat exchange that warms incoming fresh air. The ERV also recovers moisture from incoming fresh air due to lower indoor humidity levels compared to outdoors, (as shown in Figure 3.8(b)), effectively pre-dehumidifying the air (Figures 3.9(a) and (b)). This results in a mean absolute humidity difference between LDK and outdoor air increasing to 0.87 g/kg (DA), stabilizing indoor relative humidity approximately 48%—a reduction in variability compared to exhaust ventilation (Figure 3.8(c)).

Case 3 (PSE system): During daytime operation, the roof ventilation layer's temperature can reach approximately 58 °C, and the absolute humidity can soar to approximately 19.5 g/kg (DA) at the outlet (Figure 3.10(b)), as indoor moisture is transferred to the roof layer and expelled outdoors. At night, the CFI in the roof adsorbs moisture, reducing absolute humidity at the outlet. At night, moisture from the air circulating in the roof is adsorbed, leading to consistently low absolute humidity levels in the roof ventilation layer. Additionally, due to the radiative cooling effect of the roof, the air temperature at the outlet of the roof ventilation layer significantly decreases at night. Hence, the distinct daytime and nighttime modes of the roof dehumidification system are evidenced by the observed measurements of temperature and absolute humidity. The PDSC system's dehumidification throughout the day ensures indoor air's absolute humidity remains below outdoor levels, magnifying the ERV's pre-treatment effect on fresh air. Consequently, indoor relative humidity remains consistently at approximately 50%, evidencing the PSE system's superior dehumidification performance.

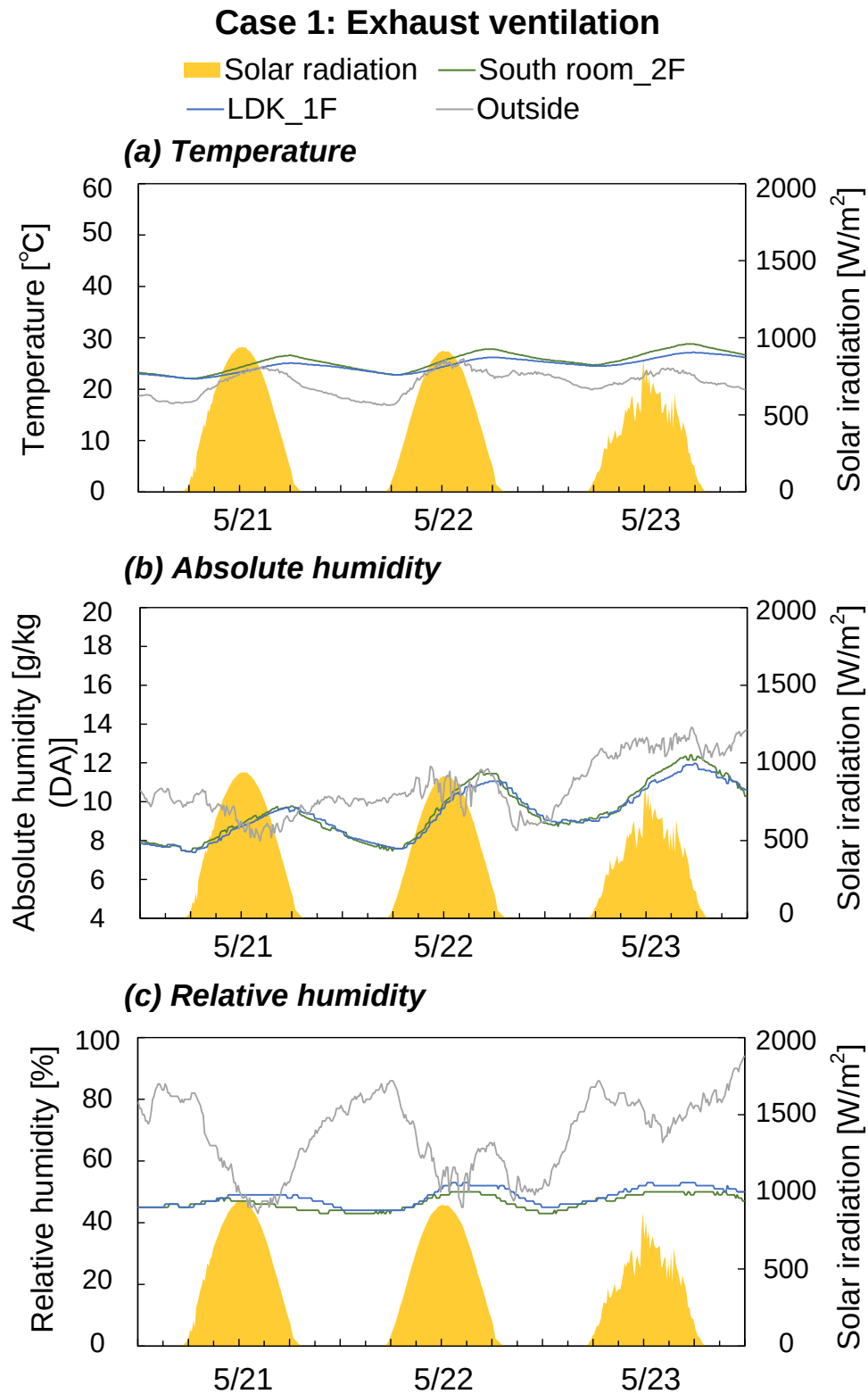


Figure 3.8 Temporal variations of temperature and humidity in Case 1: (a) temperature, (b) absolute humidity, and (c) relative humidity.

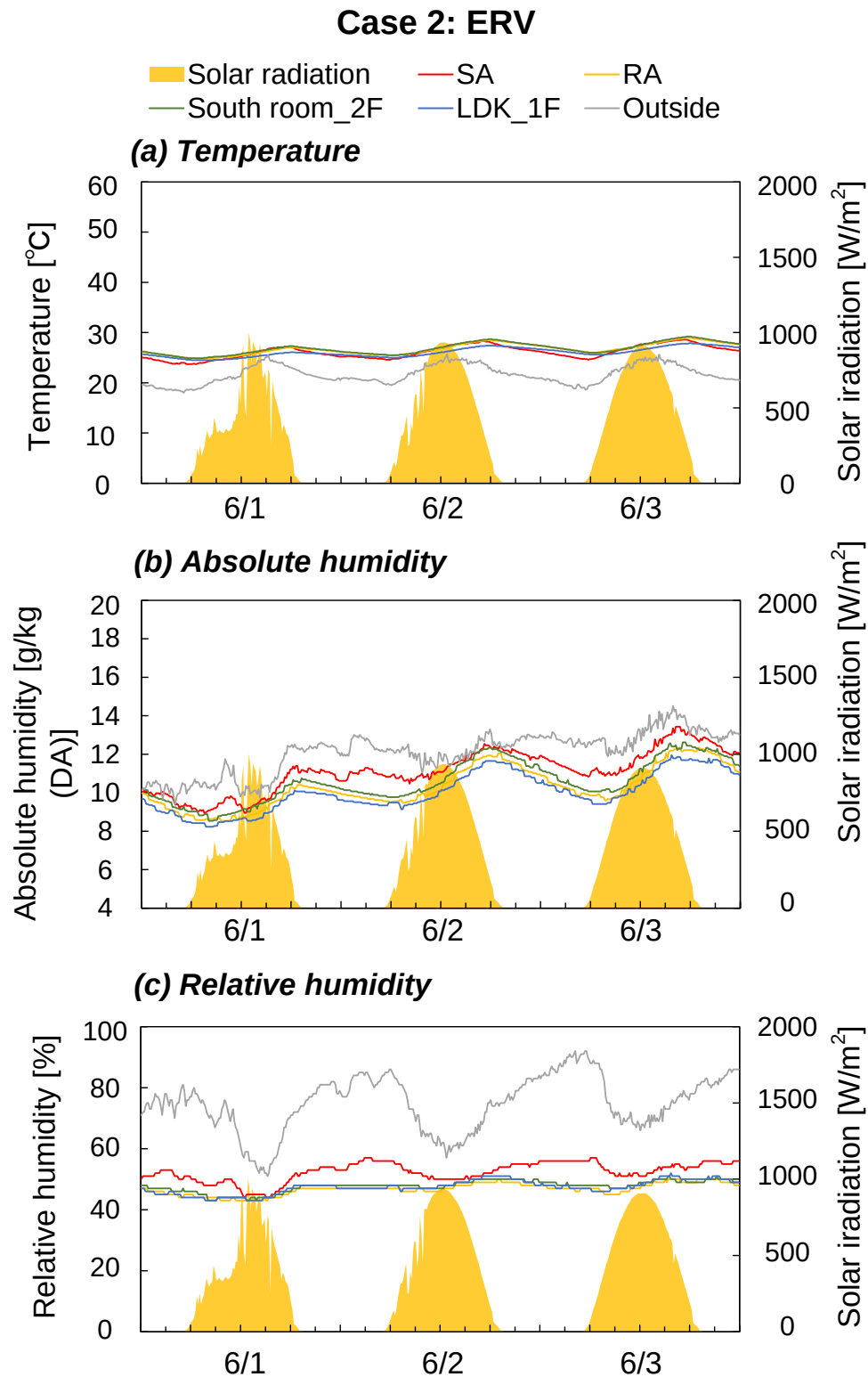


Figure 3.9 Temporal variations of temperature and humidity in Case 2: (a) temperature, (b) absolute humidity, and (c) relative humidity.

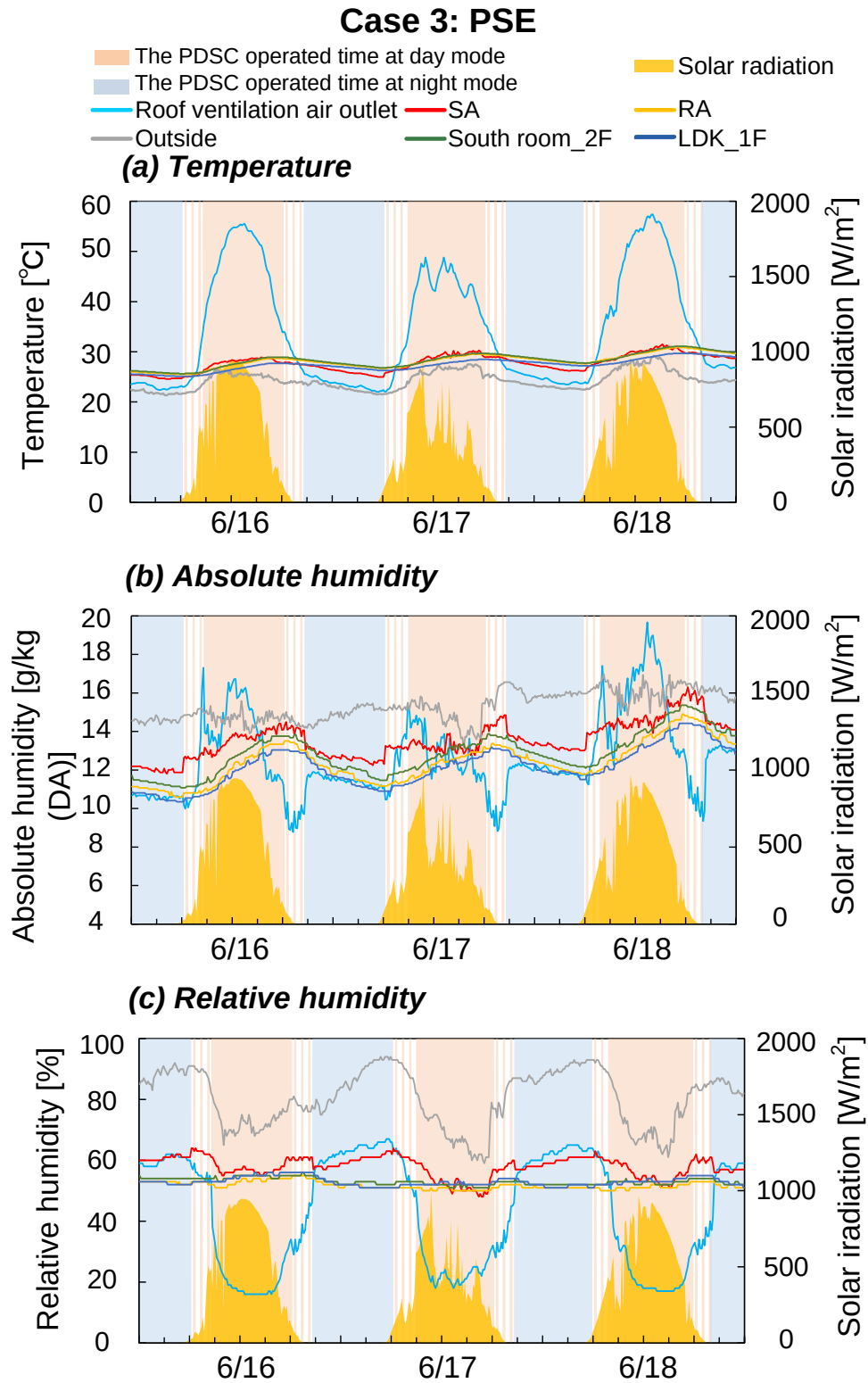


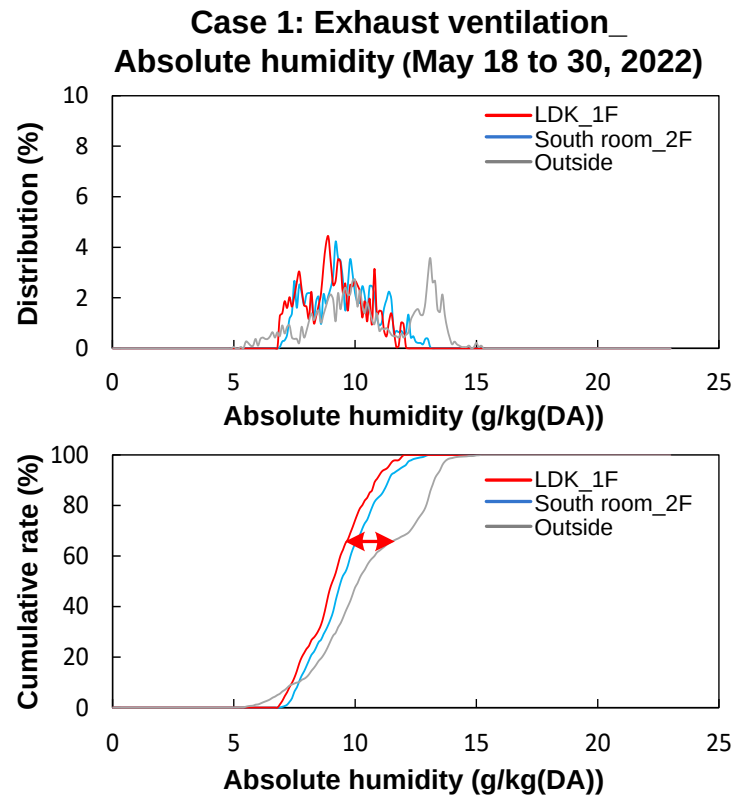
Figure 3.10 Temporal variations of temperature and humidity in Case 3: (a) temperature, (b) absolute humidity, and (c) relative humidity.

Given the challenge of assessing the differences in humidity across various cases solely through temporal variation graphs, it is crucial to aggregate and compare the distribution and cumulative humidity rates for humidity data across each case.

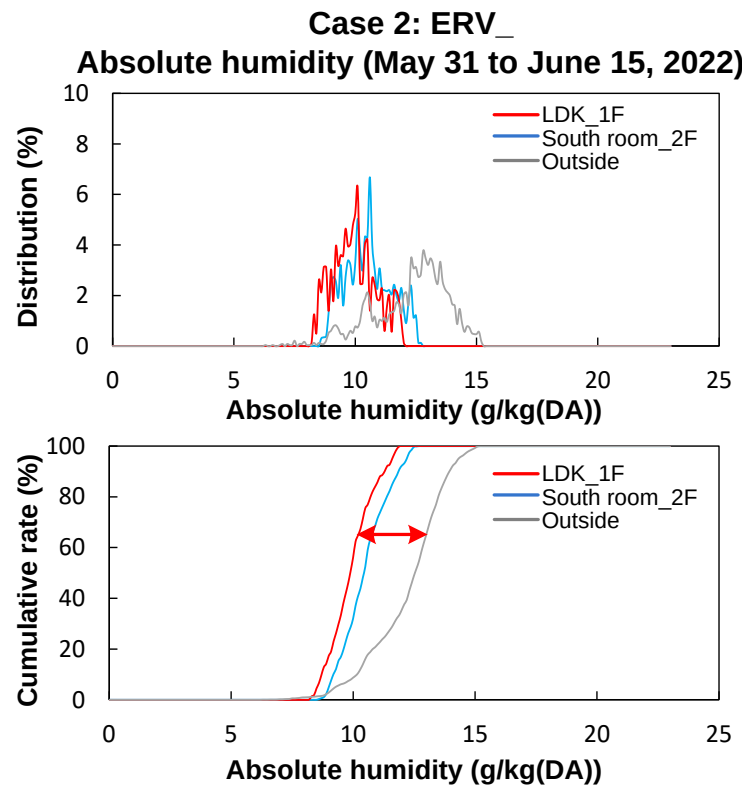
Figures 3.11 and 3.12 illustrate the distributions and cumulative rates for absolute and relative humidity, respectively, throughout the experimental period during the humid summer for Cases 1, 2, and 3. Table 3.5 presents the mean indoor and outdoor relative and absolute humidity levels for each case during the measurement period, alongside the differences between the mean indoor and outdoor absolute and relative humidity values.

The distribution and accumulation rate graphs for absolute humidity align with the temporal variation graph trends. Notably, Case 3 (PSE system) exhibited the largest difference between indoor and outdoor absolute humidity, with a significant separation of approximately 3.4 g/kg (DA) as detailed in Table 3.5. This findings illustrates the superior dehumidification performance of the PSE system over the exhaust ventilation and ERV systems throughout the extended measurement period.

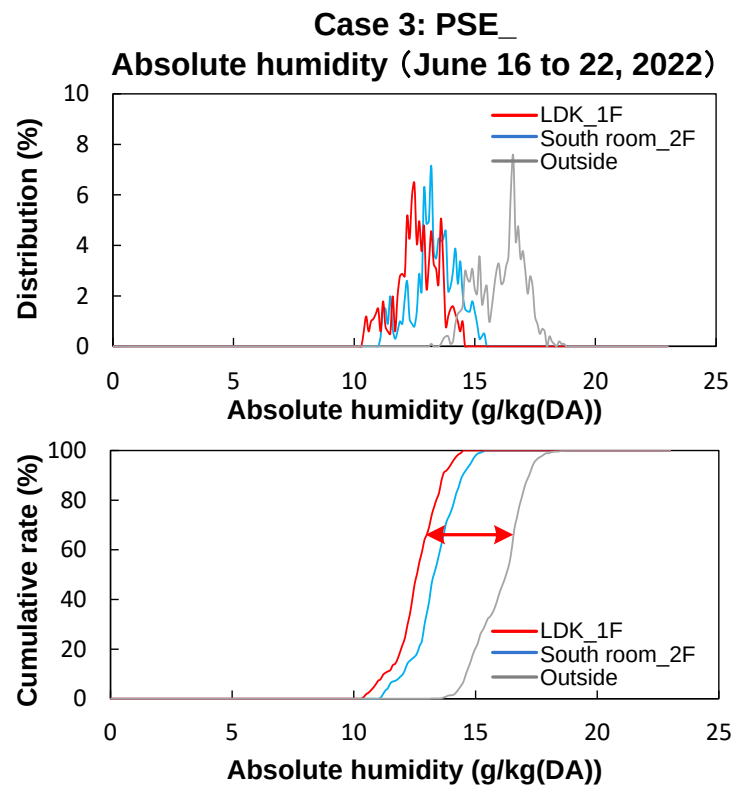
As depicted in Figure 3.12, Case 3 showcased the most constrained range of relative humidity fluctuations. With the PSE system in operation, despite the outdoor relative humidity experiencing substantial variation between 60 and 100%, indoor relative humidity remained impressively stable, fluctuating within a narrow 7% range. Indoor relative humidity consistently stayed below 56%, aligning with the comfortable humidity range. In contrast, the relative humidity in Cases 1 and 2 exhibited more pronounced variability, with fluctuations exceeding 15%. This stark contrast highlights the PSE system's exceptional capability to stabilize indoor relative humidity, ensuring a more consistent and comfortable indoor environment.



(a) Case 1: Exhaust ventilation system

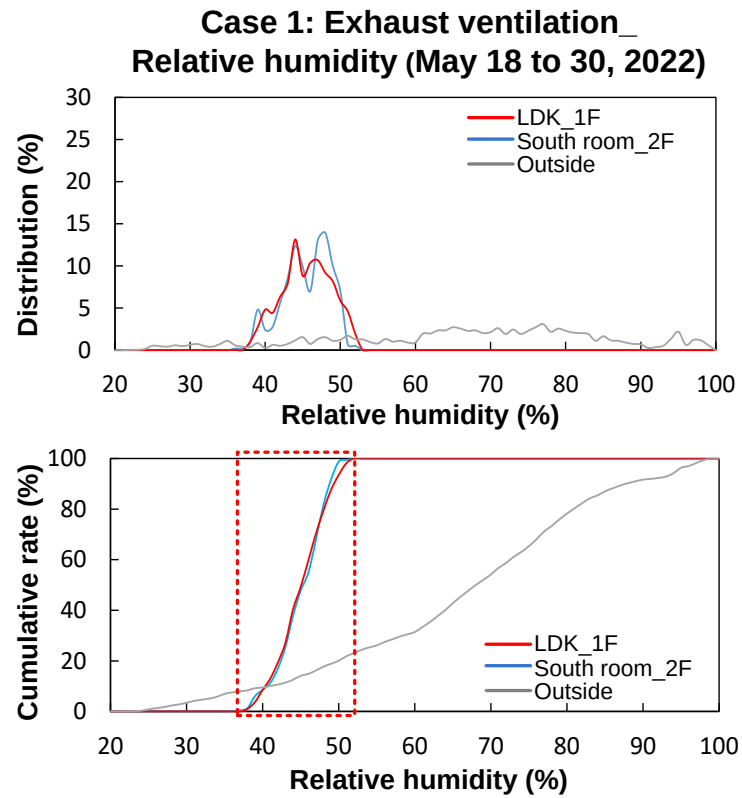


(b) Case 2: ERV system

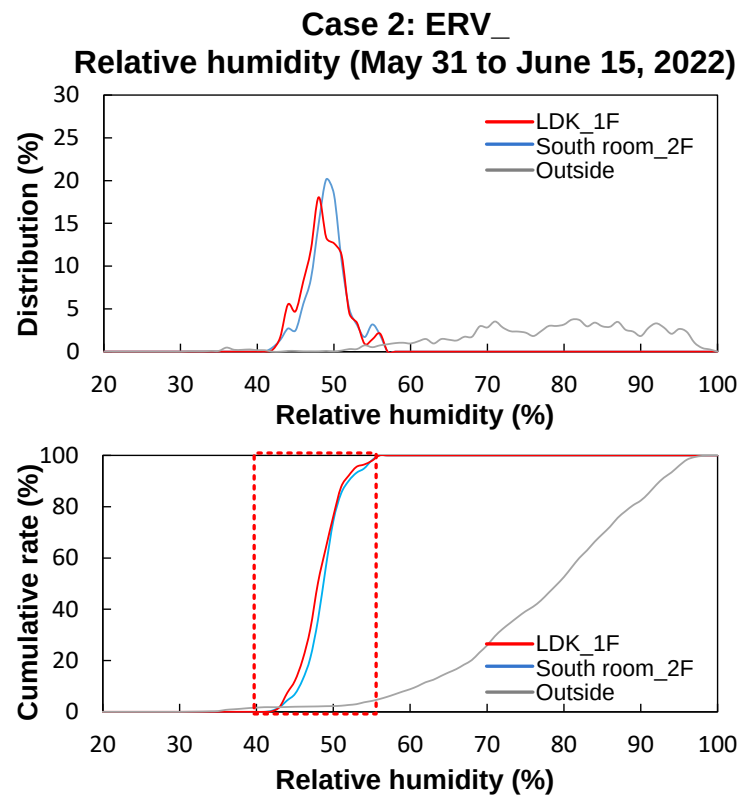


(c) Case 3: PSE system

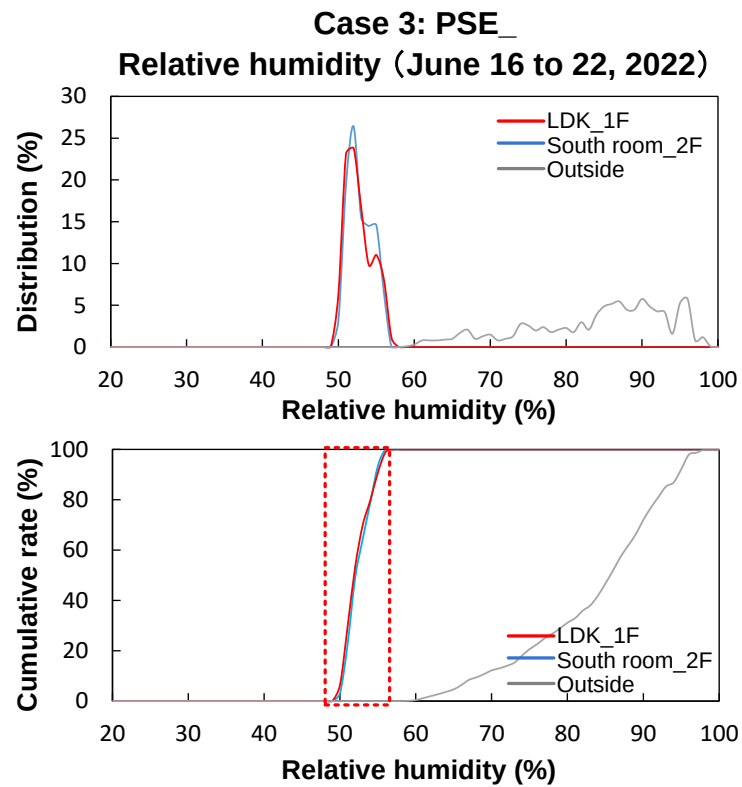
Figure 3.11 Distribution and accumulation rate of absolute humidity for Cases 1–3
[0.1 g/kg (DA) absolute humidity interval]



(a) Case 1: Exhaust ventilation system



(b) Case 2: ERV system



(c) Case 3: PSE system

Figure 3.12 Distribution and accumulation rate of relative humidity for Cases 1–3
(1% relative humidity interval)

Table 3.5 Mean indoor and outdoor temperature and humidity for Case 1, 2, and 3

Cases	Systems	Outdoors		Indoors (LDK)		Difference between indoors and outdoors	
		Mean AH [g/kg (DA)]	Mean RH (%)	Mean AH [g/kg (DA)]	Mean RH (%)	Mean AH [g/kg (DA)]	Mean RH (%)
Case 1	Exhaust ventilation system	10.4	66.6	9.1	45.5	1.3	21.1
Case 2	ERV system	12.2	77.9	9.9	48.7	2.3	29.2
Case 3	PSE system	16.0	84.0	12.6	52.7	3.4	31.3
*Temp., temperature; RH, relative humidity; AH, absolute humidity							

3.3 Measurements with operation of air-conditioning system

While summer experiments were conducted, they took place indoors without the influence of air-conditioning. Therefore, it is crucial to perform additional summer experiments incorporating the cooling effects of air-conditioning.

3.3.1 Measurement conditions

Table 3.6 outlines the conditions for Cases 4, 5, and 6, which include air-conditioning. These cases use the same operational systems as the unconditioned experiments: the exhaust ventilation system, the ERV system, and the PSE system. The experiments were scheduled during the hot and humid months of July and August, with the indoor air-conditioning system operational around the clock at a set temperature of 27 °C. Notably, there were no internal sources of moisture or heat observed during these experiments. The exhaust ventilation system's ventilation rate is based on the manufacturer's rated air change value of approximately 125 m³/h, while the ERV and PSE systems feature a slightly higher measured ventilation rate of approximately 170 m³/h.

Table 3.6 Measurement conditions for Cases 4, 5, and 6 with air-conditioning

Cases	System	Measurement period	Indoor conditions	Ventilation rate and equipment
Case 4	Exhaust ventilation system	July 13–25, 2022	Cooling = 27 °C. No internal moisture, and heat gain	125 m ³ /h (rated data), ventilation fan: Panasonic FY-24CDTK8
Case 5	ERV system	July 27–Aug 9, 2022		Approximately 170 m ³ /h (measured data), ERV
Case 6	PSE system	June 24–July 11, 2022		Refer to Figure 3.6

3.3.2 Measurement results

3.3.2.1 Dehumidification effect

Figures 3.13, 3.14, and 3.15 display the temporal variations in temperature, absolute humidity, and relative humidity across three sunny days for the operation periods of the exhaust ventilation, ERV, and PSE systems, respectively, under the test conditions without air-conditioning (27 °C).

Case 4 (exhaust ventilation system): Mid to late July sees outdoor temperatures approximately 28 °C, while indoor temperatures are maintained at approximately 27 °C due to continuous air-conditioning. The second-floor room temperatures are approximately 1 °C higher than those on the first floor. Considering practical aspects, the windows of the observation room on the first floor are shaded externally by trees, unlike those of the observation room on the second-floor. Therefore, likely due to direct solar radiation on the second-floor windows, unlike the first floor, which benefits from external shading by trees. Indoor absolute humidity is lower than outdoors, as the air conditioner dehumidifies the indoor air by condensing moisture during cooling, keeping relative indoor humidity relatively stable at approximately 70%.

Case 5 (ERV system): By August, outdoor temperatures rise to approximately 31 °C. The ERV's heat recovery effect slightly cools the fresh air SA entering the room to approximately 30 °C, demonstrating the ERV's pre-cooling benefit. The building material's humidity control and the air conditioner's dehumidification reduce indoor absolute humidity below outdoor levels, with the ERV transferring moisture from incoming fresh air to the exhaust air RA, significantly lowering the absolute humidity of incoming fresh air SA to approximately 2-3 g/kg (DA) below outdoor levels, indicating the ERV's dehumidification effect. The combined dehumidification effects maintain indoor relative humidity almost constant at approximately 65%.

Case 6 (PSE system): the temperature in each room is approximately 27 °C, lower than the outside air temperature due to the cooling effect of the air conditioner. However, the ERV's sensible heat exchange effect did not significantly lower the supply air temperature during the day, which remained slightly below the outdoor air temperature. This occurs because the ERV's OA ducts are positioned on the building's west wall, exposing incoming fresh air to elevated temperatures from solar radiation. Furthermore, the high outdoor air temperature causes the roof ventilation layer's temperature to exceed indoor air temperature even at night, despite the cooling effect of long-wave radiation. Consequently, the PDSC system only operates in night mode for a limited duration. As depicted in Figure 3.15(b), when the PDSC system is inactive at night, the absolute humidity in the roof ventilation layer drops to as low as 10 g/kg (DA) because the lack of air circulation, leaving the remaining air in the roof continuously dehumidified. Upon activating the PDSC system in night mode, indoor air flows into the roof ventilation layer, causing an increase in absolute humidity. Overall, the PSE system's dehumidification effect ensures that the absolute humidity in all indoor rooms remains lower than outdoor levels.

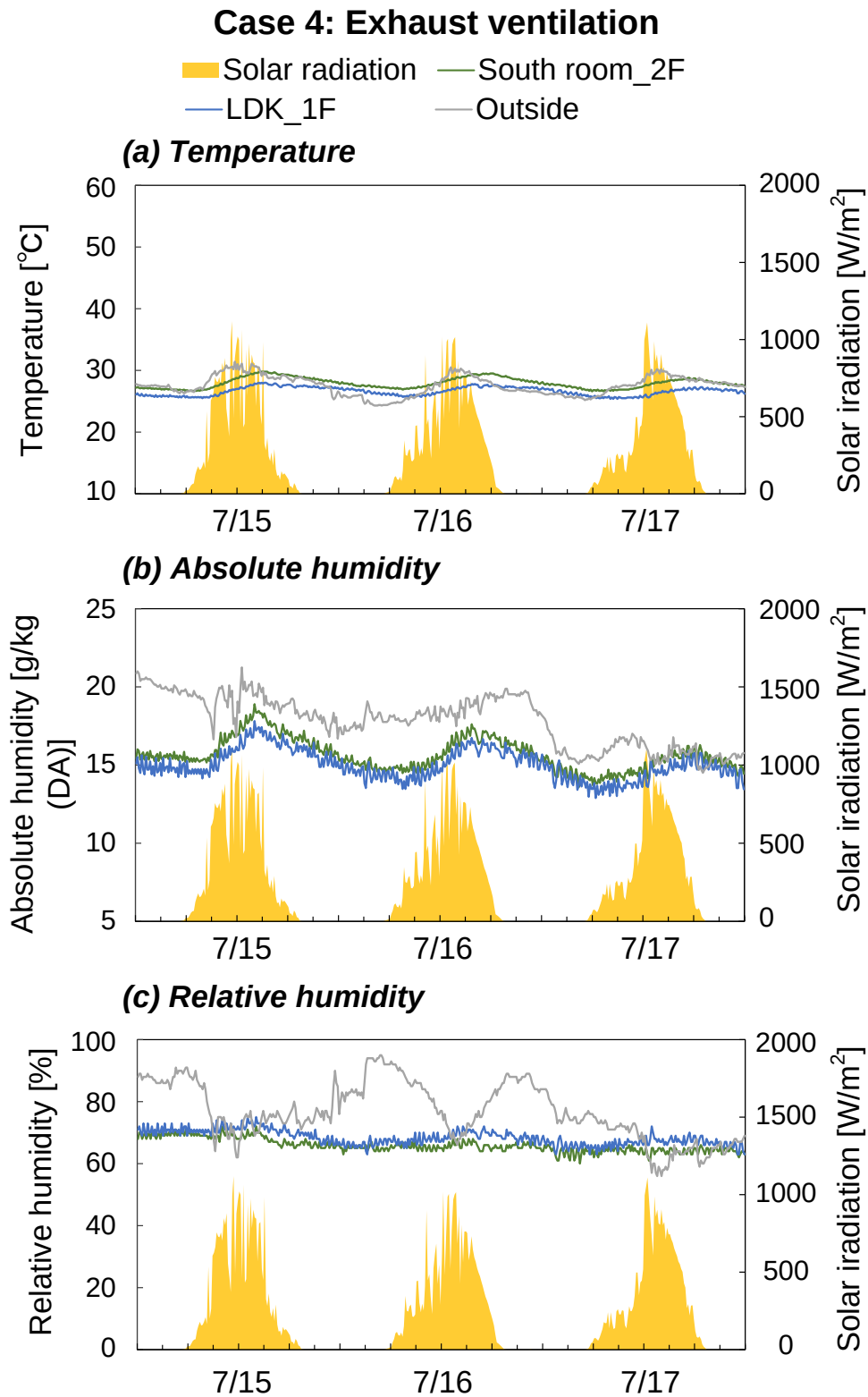


Figure 3.13 Temporal variations of temperature and humidity in Case 4: (a) temperature, (b) absolute humidity, and (c) relative humidity.

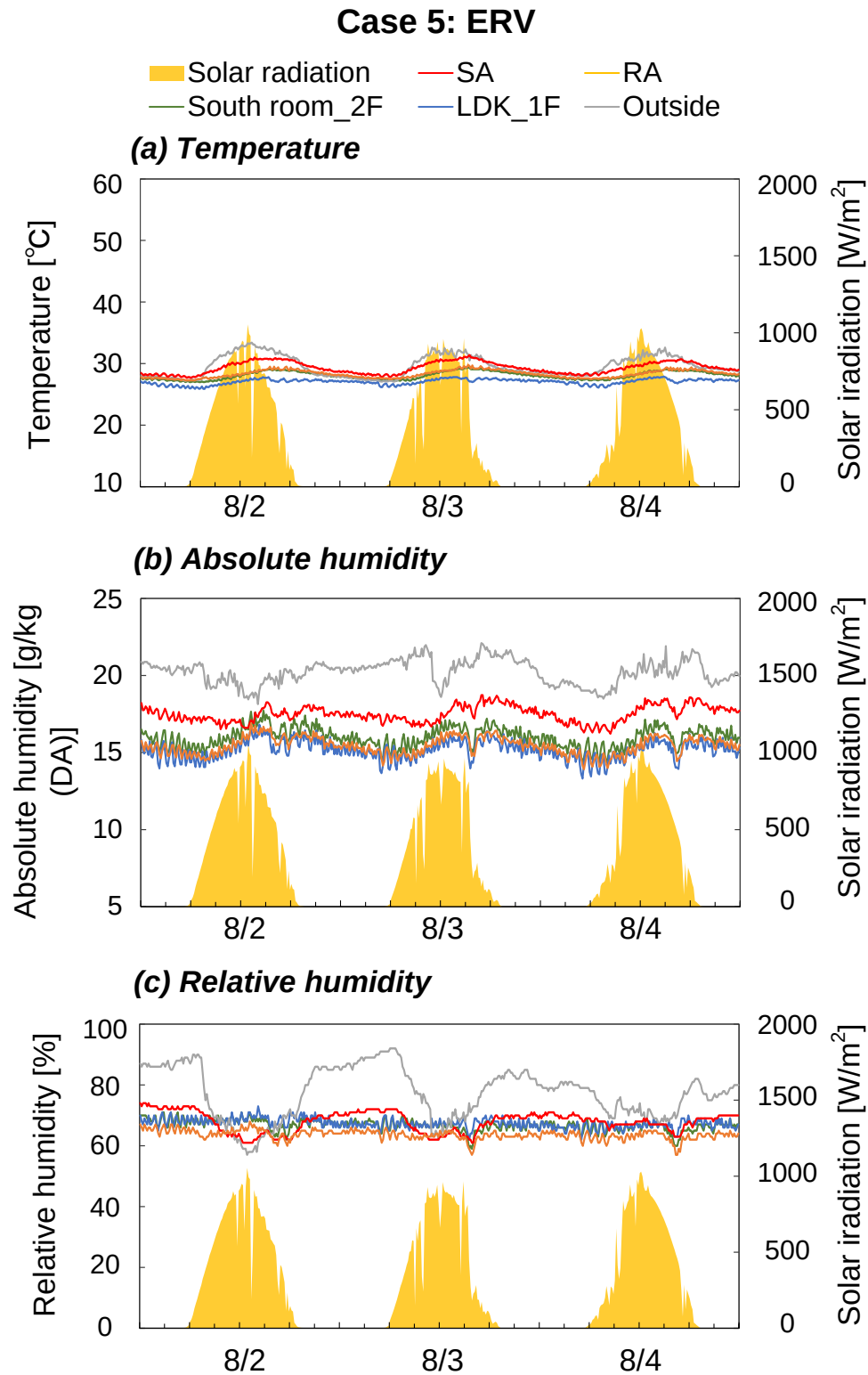


Figure 3.14 Temporal variations of temperature and humidity in Case 5: (a) temperature, (b) absolute humidity, and (c) relative humidity.

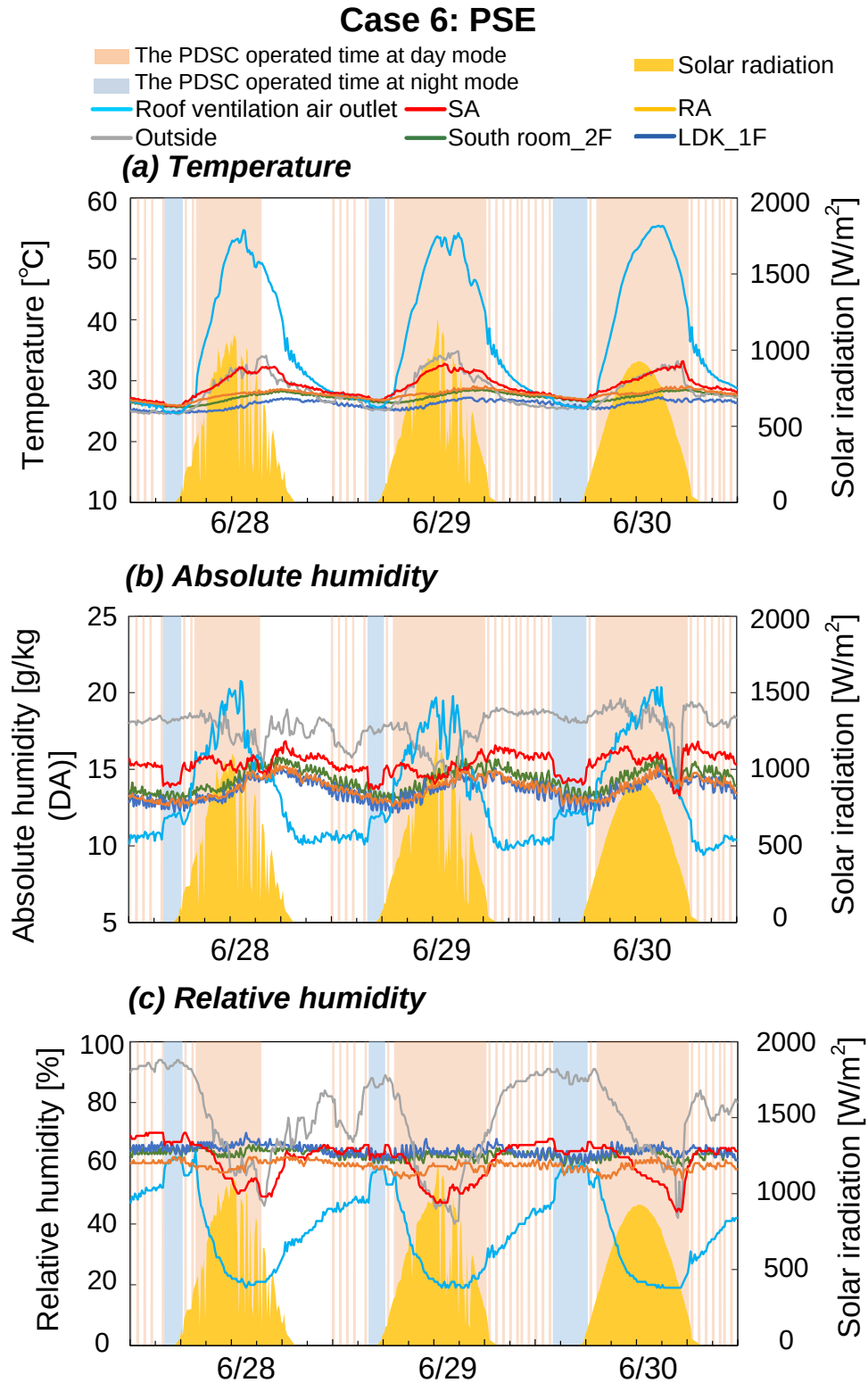
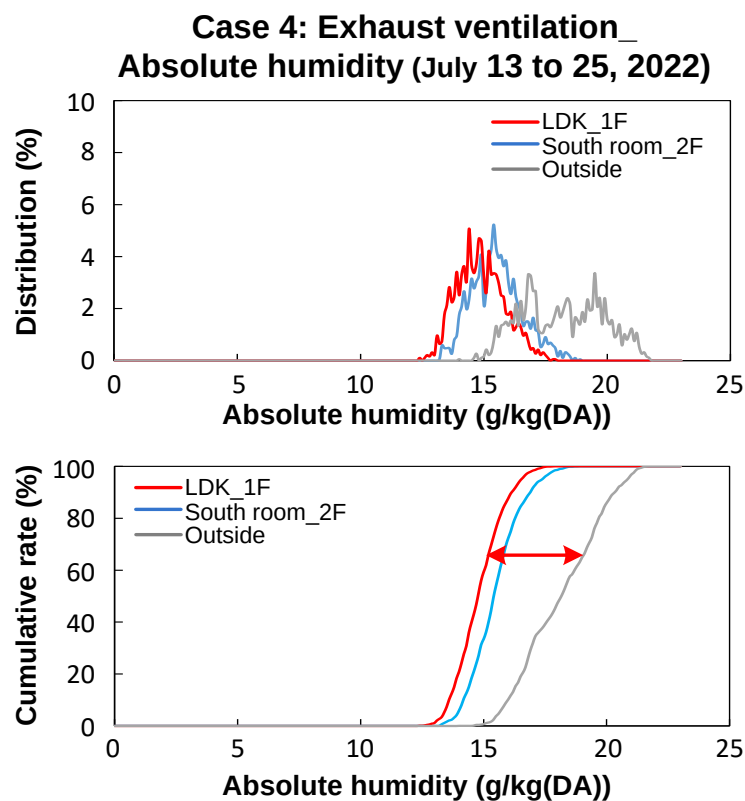


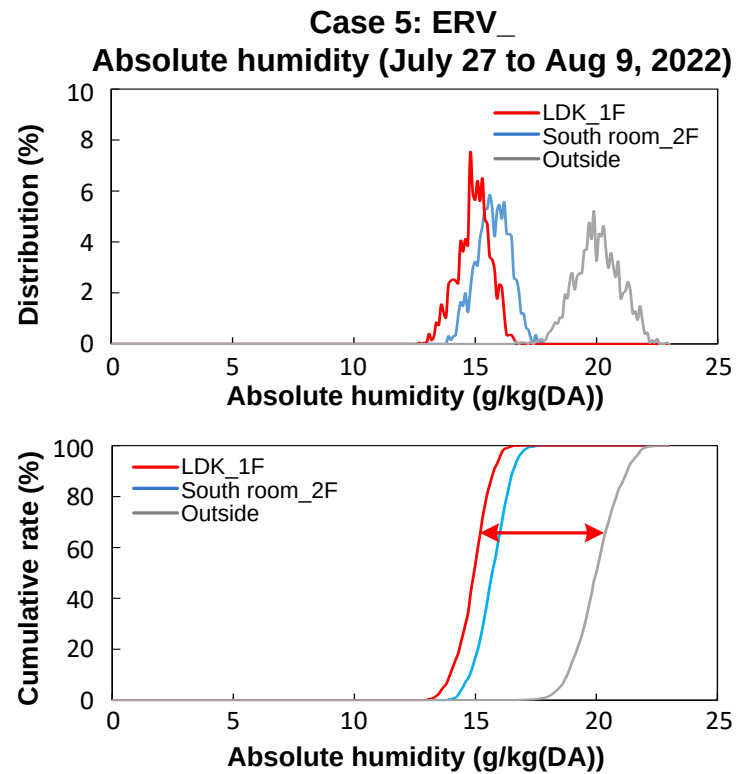
Figure 3.15 Temporal variations of temperature and humidity in Case 6: (a) temperature, (b) absolute humidity, and (c) relative humidity.

Figures 3.16 and 3.17 detail the distributions and cumulative rates for absolute and relative humidity during the humid summer for Cases 4, 5, and 6, demonstrating the effectiveness of each system under air-conditioned conditions. Table 3.7 presents the mean indoor and outdoor relative and absolute humidity levels for each case during the measurement period, including the differences between the mean indoor and outdoor absolute and relative humidity values.

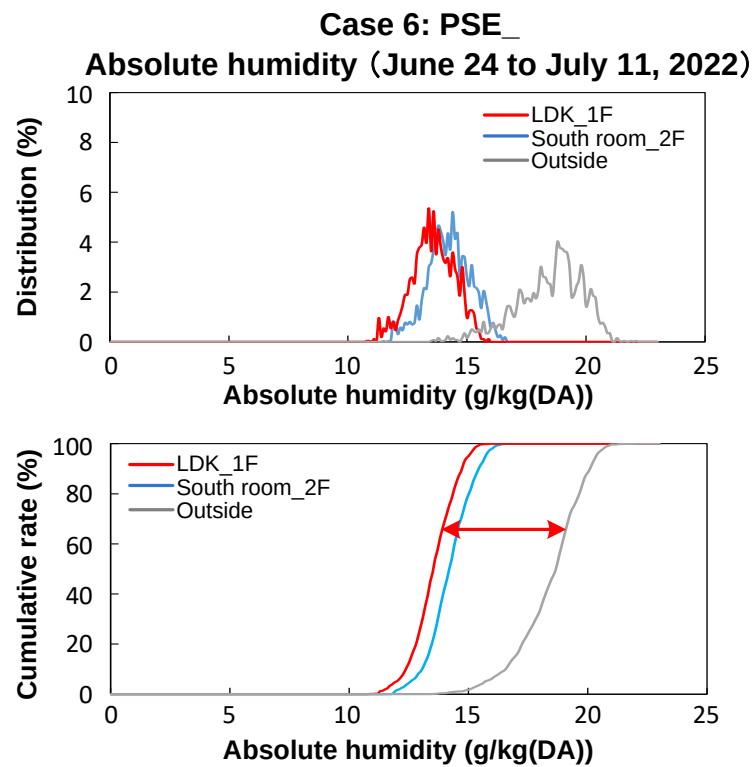
In the scenarios where the exhaust ventilation system operates (Case 4), a higher disparity is observed between the indoor and outdoor absolute humidity compared to Case 1, attributable to the air-conditioning's cooling and condensing dehumidification effects in Case 4. The mean absolute humidity difference between indoor and outdoor in Case 5 (ERV system) surpasses that of Case 4 (exhaust ventilation system), highlighting the ERV's capability to further pre-humidify fresh air by utilizing the indoor-outdoor humidity difference created by the air conditioner's dehumidifying effect. Case 6 showcases the PSE system operation, yielding the most substantial differences in both indoor-outdoor absolute humidity and relative humidity are 4.9 g/kg (DA) and 16.1%, respectively, and the lowest mean indoor absolute and relative humidity levels when compared to the other systems. However, the dehumidification effect of the PSE system is influenced by varying meteorological conditions such as solar radiation and air moisture content, necessitating numerical analysis to accurately predict its indoor humidity control effect under consistent weather conditions.



(a) Case 4: Exhaust ventilation system



(b) Case 5: ERV system

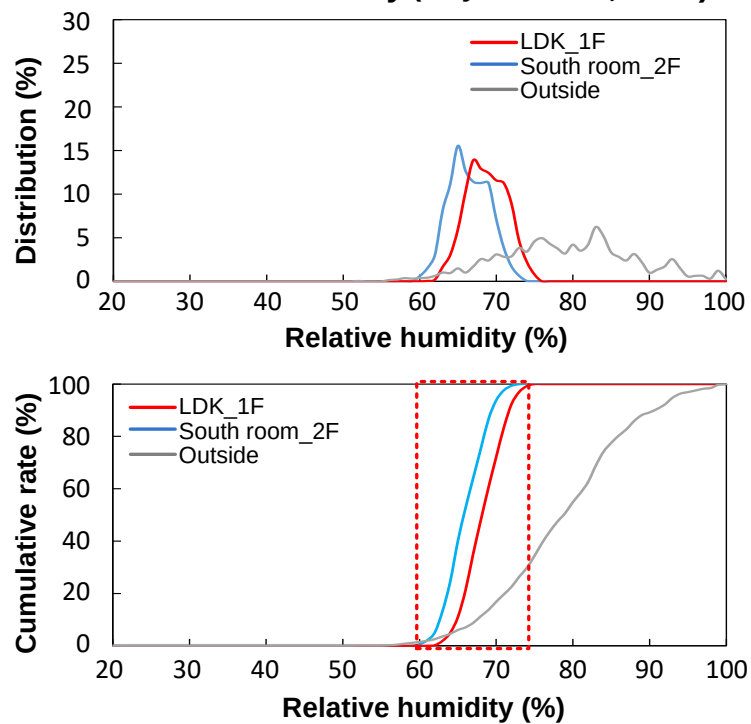


(c) Case 6: PSE system

Figure 3.16 Distribution and accumulation rate of absolute humidity for Cases 4–6

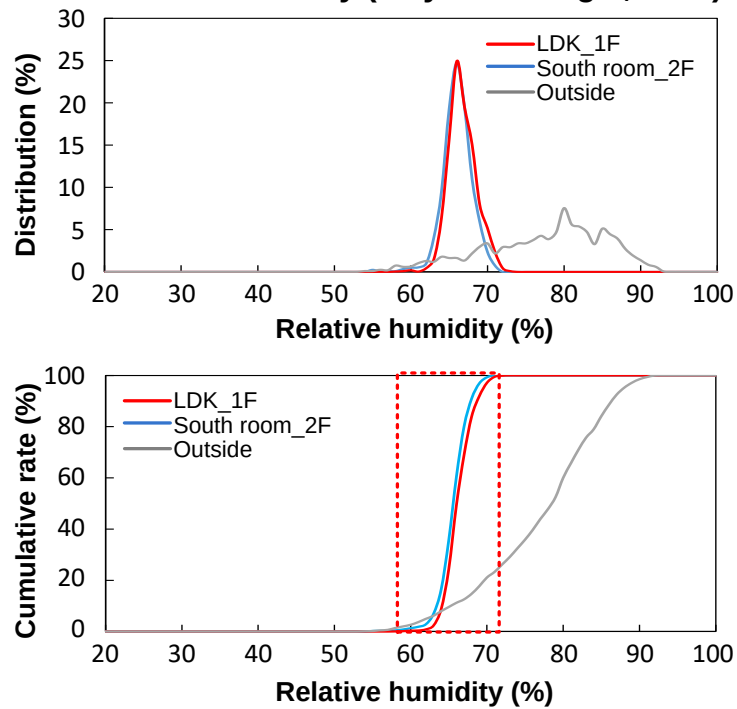
[0.1 g/kg (DA) absolute humidity interval]

**Case 4: Exhaust ventilation_
Relative humidity (July 13 to 25, 2022)**

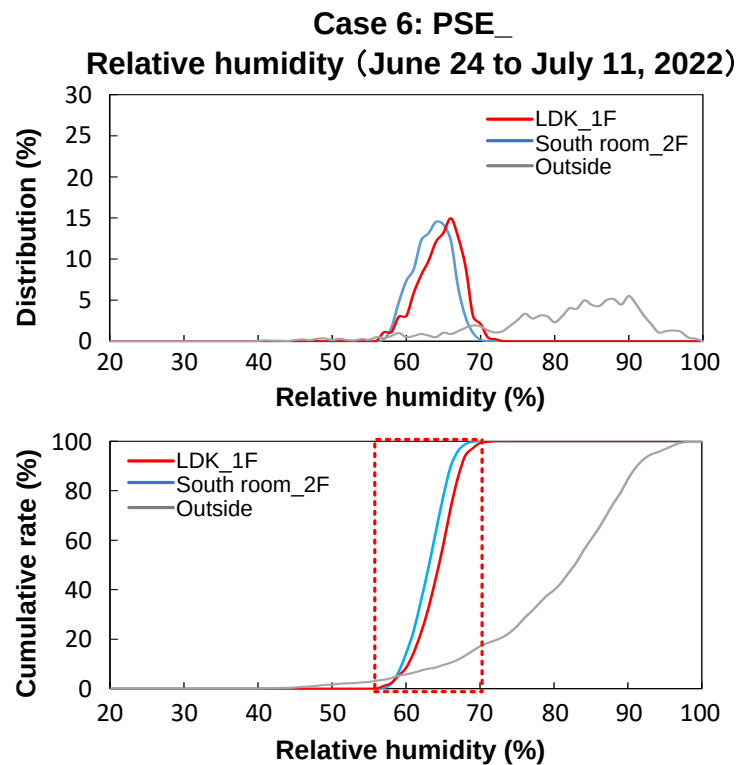


(a) Case 4: Exhaust ventilation system

**Case 5: ERV_
Relative humidity (July 27 to Aug 9, 2022)**



(b) Case 5: ERV system



(c) Case 6: PSE system

Figure 3.17 Distribution and accumulation rate of relative humidity for Cases 4–6
(1% relative humidity interval)

Table 3.7 Mean indoor and outdoor temperature and humidity for Case 4, 5, and 6

Cases	Systems	Outdoors		Indoors (LDK)		Difference between indoors and outdoors	
		Mean AH [g/kg (DA)]	Mean RH (%)	Mean AH [g/kg (DA)]	Mean RH (%)	Mean AH [g/kg (DA)]	Mean RH (%)
Case 4	Exhaust ventilation system	18.1	79.2	14.8	68.7	3.3	10.4
Case 5	ERV system	20.0	77.5	14.9	66.7	5.1	10.7
Case 6	PSE system	18.4	80.8	13.5	64.6	4.9	16.1
*Temp., temperature; RH, relative humidity; AH, absolute humidity							

3.3.2.2 Ventilation latent heat load reduction effect

Given the absence of internal heat and moisture sources during the experiments, the primary source of latent heat load was ventilation. Consequently, the ventilation's latent heat load was calculated for each Case 4, 5, and 6 based on daily experiments under different conditions.

The calculation for summer's ventilation latent heat load under air-conditioning conditions involves considering the ventilation rate and absolute humidity difference between indoor and outdoor air for each system. For the ERV system, (Case A-5), calculations use the humidity difference between return and supply air alongside the ERV's ventilation rate, whereas for the PSE system (Case A-6), calculations adjust for the PDSC system's dehumidification capacity. The daytime dehumidification and nighttime moisture adsorption capacity of the PDSC system can be determined by the airflow rate and the absolute humidity difference between the inlet and outlet of the roof ventilation layer. Detailed equations for these calculations are provided below.

$$q_{ex} = l_w \cdot \gamma_a \cdot Q_{vent}(x_i - x_o) \quad (3.1)$$

$$q_{erv} = l_w \cdot \gamma_a \cdot Q_{vent}(x_{ra} - x_{sa}) \quad (3.2)$$

$$q_{pse_day} = l_w \cdot \gamma_a \cdot Q_{vent}(x_{ra} - x_{sa}) - l_w \cdot \gamma_a \cdot Q_{roof}(x_{roof_out} - x_{roof_in}) \quad (3.3)$$

$$q_{pse_night} = l_w \cdot \gamma_a \cdot Q_{vent}(x_{ra} - x_{sa}) - l_w \cdot \gamma_a \cdot Q_{roof}(x_{roof_in} - x_{roof_out}) \quad (3.4)$$

where l_w and γ_a represent the evaporation latent heat of moisture and the specific weight of air, respectively. Q denotes the ventilation rate and x signifies the absolute humidity. Figure 3.18 compares the ventilation latent heat loads across Cases 4, 5, and 6, showing that as the average absolute outdoor humidity increases, so does the ventilation system's latent heat load due to the ingress of more water vapor. The ERV system significantly reduces the ventilation latent heat load compared to the exhaust system by dehumidifying the high-humidity outside air upon ventilation. The PSE system, through the PDSC's dehumidification effect, creates a significant indoor-outdoor humidity difference, enhancing the ERV's moisture recovery capability and thus presenting the lowest ventilation latent heat load among the systems for comparable outdoor humidity conditions.

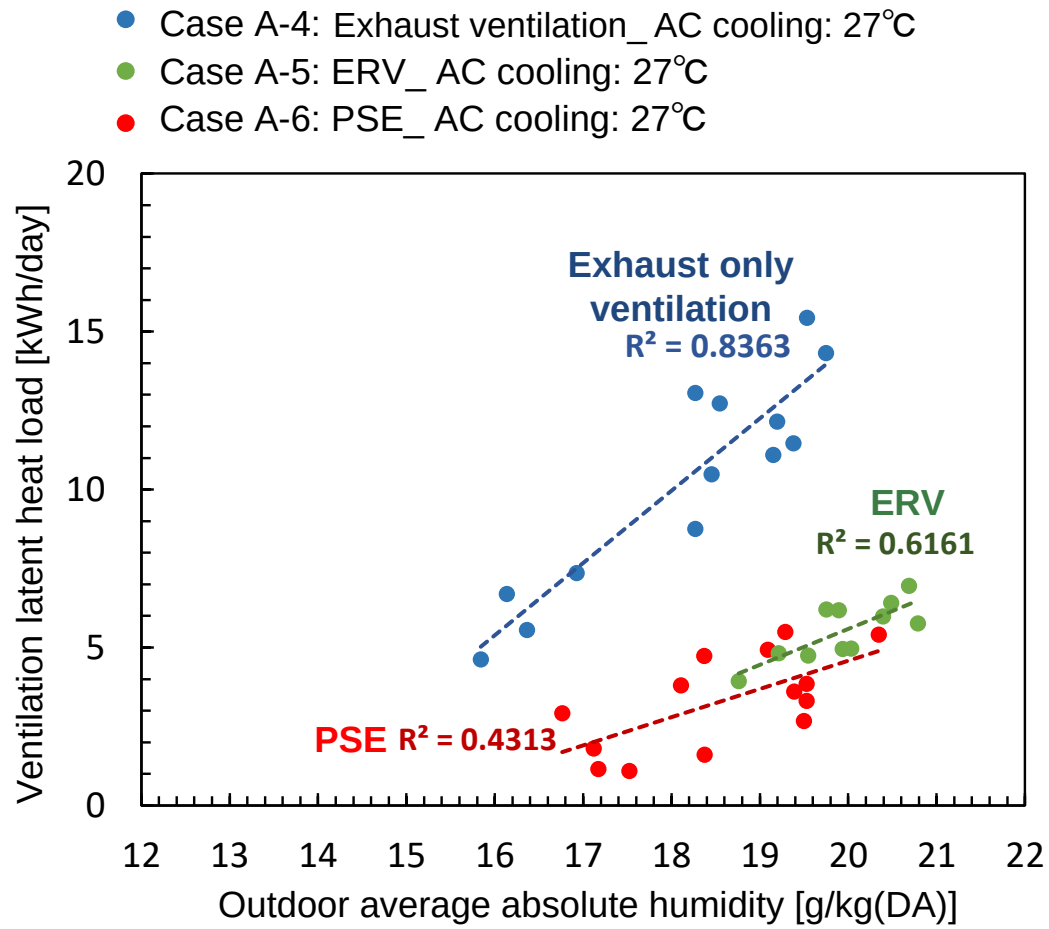


Figure 3.18 Comparison of ventilation latent heat load in summer with air-conditioning

3.4 Reference

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<https://doi.org/10.1016/j.buildenv.2015.09.031>.
- [4] H. Lee, A. Ozaki, M. Lee, T. Yamamoto, Humidity control effect of vapor-permeable walls employing hygroscopic insulation material, *Indoor Air.* (2019) 346–360.
<https://doi.org/10.1111/ina.12622>.

Chapter 4 Simulation analysis by using numerical simulation under several conditions

In Chapter 3, the superior dehumidification performance of the PSE system, compared to other common systems (exhaust ventilation and ERV systems), was analyzed using field experiments. However, the field experiments still have the following limitations:

- 1) Analyzing the field experiment data cannot quantify the energy consumption reduction effect.
- 2) The measurement data from the various cases do not allow for rigorous controlled experiments under same meteorological conditions.
- 3) Due to the limitations in the design of the ERV unit, the PDSC system cannot operate separately from the ERV; therefore, constructing a PDSC-only system using simulation software is necessary. This is crucial for comparing the effects of the PSE and PDSC systems and evaluating the effectiveness of the improved PSE system proposal.
- 4) Simulation can provide season-round data essential for comprehensive evaluations.

Therefore, in Section 4, a high-precision hygrothermal simulation software, THERB for HAM [1], was selected for the simulation analysis. The simulation is conducted in two main parts.

Numerical models of three experimental cases (exhaust ventilation, ERV, and PSE systems) were constructed and validated. Subsequently, six residential models with various envelope characteristics, including PDSC-only, ERV-only, and PSE models, were developed. Two numerical calculations were performed using the high-precision simulation software THERB for HAM, Cases A and B. Case A involved calculating the indoor temperature and humidity of the PSE system compared to the other systems under unconditioned conditions. Case B simulated the indoor temperature, humidity, and loads of the PSE system compared to the other systems under constant temperature and relative humidity setting conditions. The THERB for HAM software was then coupled with a heat-pump characteristic model capable of reproducing the air-conditioned vapor cycle. Three separate calculations were performed under constant temperature-setting conditions: Cases C, D, and E. Case C calculated the indoor temperatures, humidity levels, and heat loads of the PSE system in relation to the other systems. Case D estimated the PSE system's dehumidification effect under various air conditioner airflows. Case E analyzed the PSE system's dehumidification effect at high refrigerant evaporation temperatures.

4.1 Simulations for evaluating PSE performance by using constant temperature and relative humidity air-conditioning model in THERB for HAM

4.1.1 Simulation software THERB for HAM

THERB for HAM (Simulation Software of the Thermal Environment of Residential Building for Heat and Mass Transfer) is a dynamic calculation software designed for evaluating temperature, humidity, thermal comfort, and heat load of indoor environments. It is officially recognized software that has passed standardized testing, such as the Building Energy Simulation Test procedure, and has been approved by the Japanese government [2]. THERB for HAM integrates detailed heat transfer theories—conduction, convection, and radiation—and a sophisticated model of heat, air, and moisture transfer. Notable features of THERB for HAM include [3]:

- The heat and moisture transfer model, known as the P-model, utilizes water's thermodynamic energy (water potential) as the driving force. It accounts for the moisture adsorption and desorption phenomena between hygroscopic materials and the indoor environment, specifically targeting the air layer of envelopes. This method is verified to be more accurate than models based solely on physical properties such as vapor pressure.
- It incorporates time variation in heat and moisture convection.
- It calculates heat and moisture transfer coefficients for both natural and forced convection using dimensionless organizing equations.
- It performs rigorous geometric calculations of sun/shade areas on indoor surfaces.
- It simulates multiple reflections of transmitted solar radiation on indoor surfaces.
- It includes a multi-layer window model.
- It addresses the nonlinearity of radiation heat transfer.
- It simulates long-wave radiation heat transfer between indoor surfaces.

The moisture transfer model (P-model) was first mentioned in THERB for HAM. The P-model uses water potential as the driving force for moisture transfer, considers the effects of temperature and humidity variations on moisture balance. Its application in THERB for HAM enables accurate simulation of the hygrothermal behavior of the building envelope, such as moisture interactions between indoor air (or wall cavity) and moisture-buffering building materials. Proven to be more accurate than other simulation software, THERB for HAM is predominantly used to evaluate the performance of high-performance residential buildings and their components [4–11]. This study utilized THERB for HAM for numerical simulations.

The software's defining equations for water potential, along with heat and moisture conduction equations within walls and heat and moisture transfer equations in wall cavities, are detailed in Section 2.1. The balance equations for heat and moisture transfer within indoor spaces, as per the P-model in THERB for HAM, assume instantaneous and uniform distribution of heat and moisture throughout the space.

Heat balance equation:

$$V_{i,ol} \cdot \overline{c\gamma} \frac{dT_i}{dt} = \sum_{j=1}^J S_{i,j} \cdot \alpha_{c,i,j} (T_{s,i,j} - T_i) + \sum_{k=1}^k V_k \cdot c\gamma (T_k - T_i) \quad (4.1)$$

$$+ V_o \cdot c\gamma (T_o - T_i) + Q_{g,h}$$

Moisture balance equation:

$$V_{i,ol} \cdot \gamma_{w,i} \frac{d\mu_i}{dt} = \sum_{j=1}^J S_{i,j} \cdot \alpha'_{\mu,i,j} (\mu_{w,s,j} - \mu_i) + \sum_{k=1}^k V_k \cdot (\gamma_{w,k} \cdot \mu_k^* - \gamma_{w,i} \cdot \mu_i) \quad (4.2)$$

$$+ V_o \cdot (\gamma_{w,o} \mu_o^* - \gamma_{w,i} \cdot \mu_i) + Q_{g,w}$$

where $\overline{c\gamma}$ represents the apparent volumetric specific heat of the room, incorporating furniture and clothing [J/(m³ K)], $c\gamma$ is the volumetric specific heat of indoor air, $Q_{g,h}$ and $Q_{g,w}$ are the indoor heat generation (J/s) and moisture generation (g/s), respectively. $S_{i,j}$ and $T_{s,i,j}$ are the areas (m²) and surface temperatures (K) of the j surfaces of room i , respectively. T_o and T_k are the outdoor temperature and indoor temperature of room k , respectively. $V_{i,ol}$ is the volume of room i (m³), V_k and V_o are the amount of air circulation between rooms and the ventilation rate (m³/s), respectively; $\alpha_{c,i,j}$ and $\alpha'_{\mu,i,j}$ are the convective heat transfer coefficient [W/(m² K)] and moisture transfer coefficient [kg/(m² s J/kg)] for surface j of room i , respectively. $\gamma_{w,i}$ is the water vapor density [kg/(m³ J/kg)] of air in room i . In Eqs. (4.1) and (4.2), the first term on the right signifies the convective transfer of heat and moisture, the second term represents heat and moisture loss due to air circulation between rooms, the third term accounts for heat and moisture loss due to ventilation, and the fourth term indicates heat and moisture generation inside the room.

The heat load calculation method used in THERB for HAM is based on an ideal air-conditioning model where both sensible and latent heat loads are derived from heat and moisture balances, with indoor temperature and relative humidity maintained at predetermined constant values. Thus, the sensible and latent heat loads can be precisely calculated.

4.1.2 Model validations

4.1.2.1 Validation conditions

Three house models equipped with different systems (exhaust ventilation, ERV, and PSE systems) were constructed for simulation analysis. The basic configuration of the target experimental house and the settings from measurement Cases 1–3 were utilized as inputs to create the three numerical house models.

The simulation conditions for model validation are listed in Table 4.1. The models include an exhaust ventilation system model, an ERV model, and a PSE model. These residential models are identical except for two aspects: the type of ventilation system they employ and the presence or absence of a PDSC system. The models have identical building envelope parameters, detailed in Table 3.3 (Section 3.1.2). Weather data were sourced from real-time monitoring by the Japan Meteorological Agency [12]. The exhaust ventilation system model's ventilation rate adheres to the fan's rated capacity of 125 m³/h, while the ERV and PSE models are aligned with measurement data, both 170 m³/h. Notably, the ERV operated with unbalanced indoor and outdoor airflows during the day, achieving sensible and latent heat exchange efficiencies of 0.40 and 0.50, respectively, and 0.80 and 0.58 at night. These efficiencies were derived from temperature and absolute humidity data collected when the heat and moisture exchange efficiency stabilized, and the ERV was operational during the experiment. The airflow velocity within the dehumidification roof of the PSE model was 0.05 m/s during the day and 0.10 m/s at night, based on converted measured air flow rates. Besides the PDSC roof, THERB for HAM dynamically calculated the airflow velocity through the models' walls and roofs based on a dimensionless buoyant airflow model in the cavity, factoring in temperature differences (buoyancy) [13].

Table 4.1 Simulation conditions used for model validations

Items	Models		
	Exhaust ventilation system model	ERV model	PSE (PDSC+ERV) model
Measurement cases for comparison	Case 1	Case 2	Case 3
Measurement period	2022.5.21-5.25	2022.6.6-6.11	2022.6.17-6.22
Weather data	Measured meteorological data from Japan Meteorological Agency (May 18 to June 22, 2022) [12]		
Indoor condition	No air-conditioning		
Internal moisture and heat gain	No internal moisture and heat generation		
Ventilation rate	125 m ³ /h	170 m ³ /h	
Heat exchange efficiency	-	Sensible heat exchange efficiency: 0.80. Latent heat exchange efficiency: 0.58	Sensible heat exchange efficiency: 0.40 during the daytime, 0.80 at nighttime. Latent heat exchange efficiency: 0.50 during the daytime, 0.58 at nighttime.
Velocity of airflow through walls' cavity	Calculated non-stationarily by THERB for HAM.		
Velocity of airflow through roofs' cavity	Calculated non-stationarily by THERB for HAM.		In the PDSC roof: daytime = 0.05 m/s and nighttime = 0.10 m/s. In the other roofs: calculated non-stationarily by THERB for HAM.

4.1.2.2 Validation results

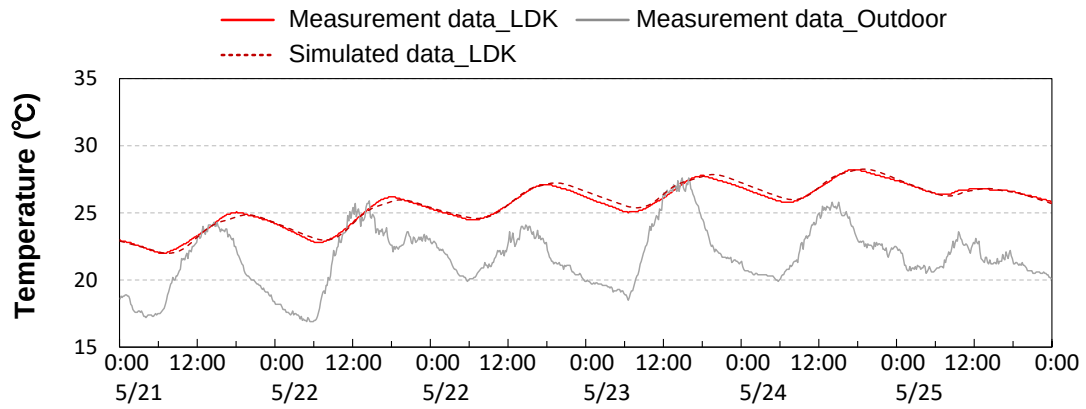
Figures 4.1 and 4.2 compare the measured values and simulated results for the indoor LDK (living room, dining room, and kitchen) temperature, absolute humidity, and relative humidity of Cases 1 and 2, respectively. Moreover, Figures 4.3 and 4.4 compare the measured values and simulated results for the indoor LDK and roof ventilation layer temperatures, absolute humidity, and relative humidity of Case 3.

To assess the reliability of the developed building models, a dimensionless criterion indicator, the mean absolute percentage error (MAPE), was introduced for model validation according to the standards and guidelines of ASHRAE Guideline 14 [14]. The equation is shown as follows:

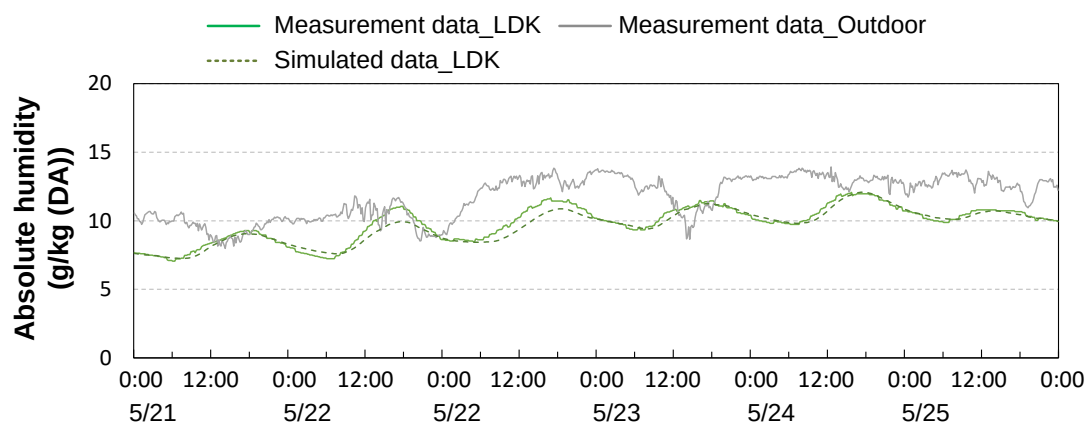
$$MAPE = \frac{1}{n} \sum_{i=1}^n \frac{|\hat{y}_i - y_i|}{y_i} \cdot 100\% \quad (4.3)$$

where n is the total number of measured data, $\hat{y}_i$ is the predicted value, and y_i is the measured value.

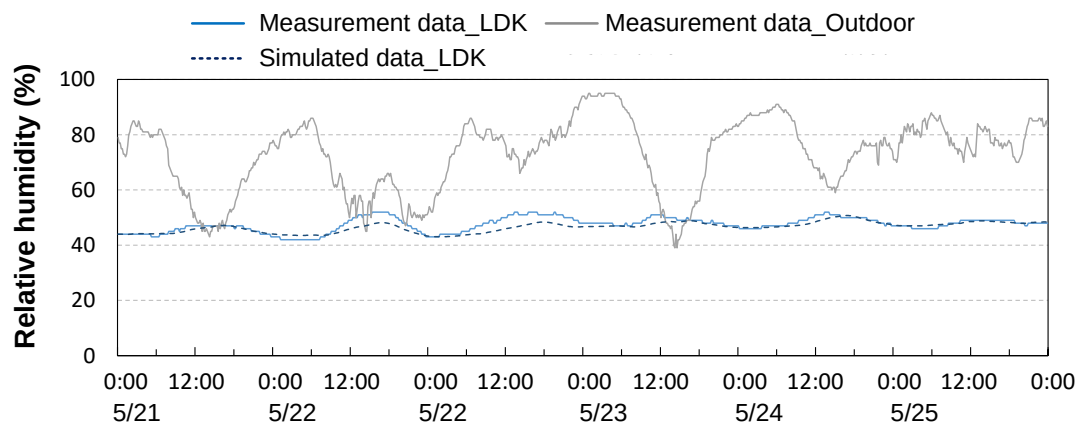
Table 4.2 lists the MAPE between the measured and simulated temperature, absolute humidity, and relative humidity data. The average MAPE of Cases 1 and 2 in the LDK room is 2.02% and 2.17%, respectively, and the average MAPE of Case 3 in the LDK room and roof ventilation layer is 1.78% and 8.79%, respectively. These results demonstrate the high accuracy of the THERB for HAM software in forecasting for all house models [15].



(a) Temperature

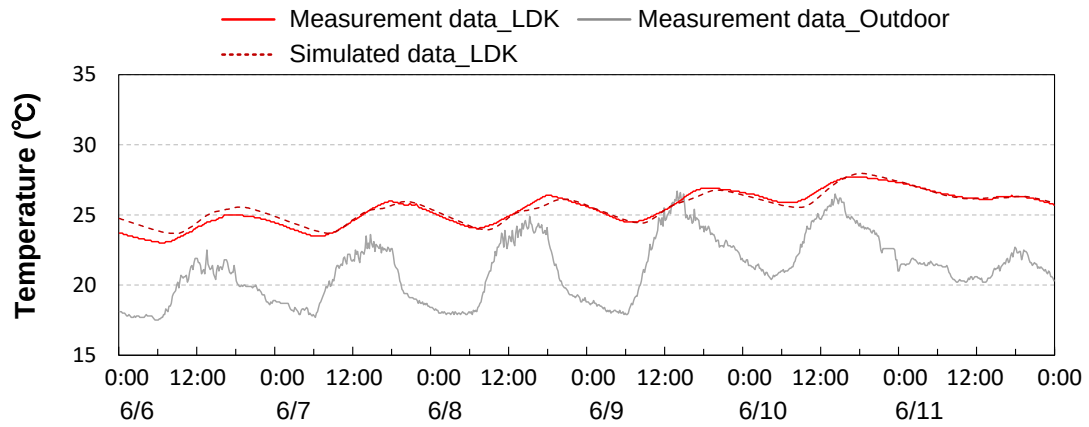


(b) Absolute humidity

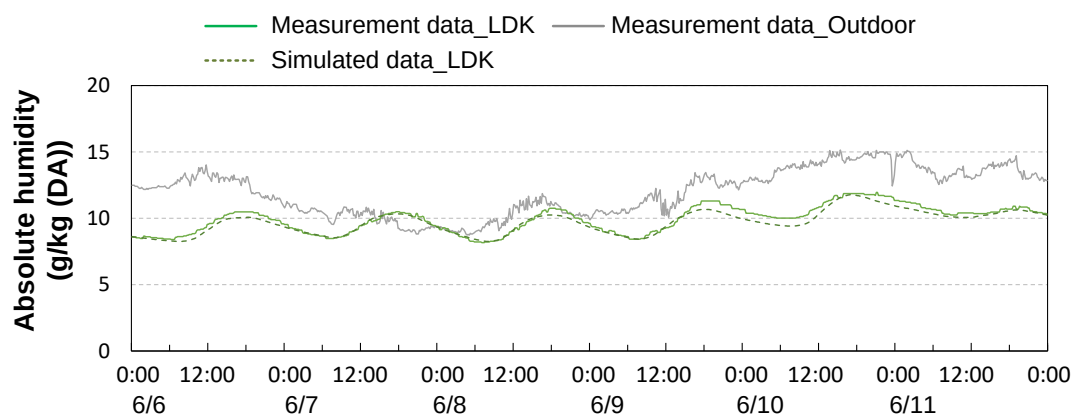


(a) Relative humidity

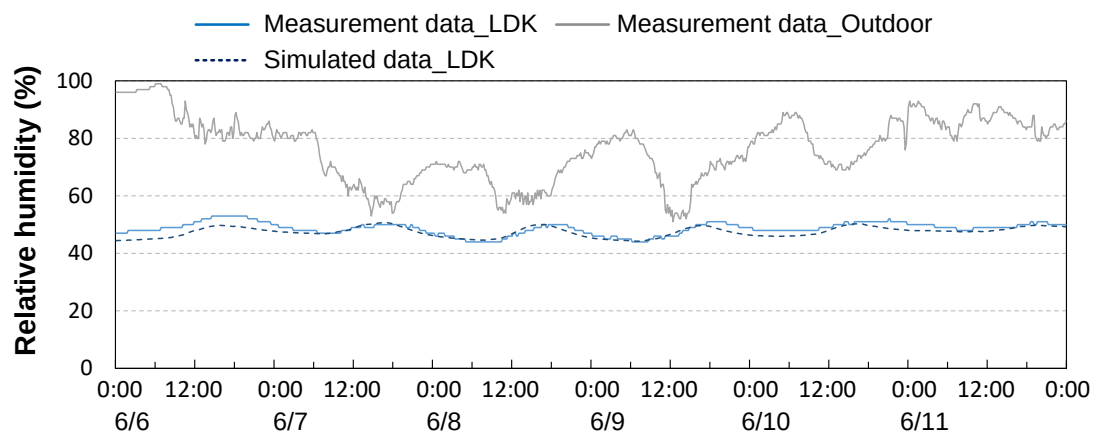
Figure 4.1 Comparisons between the measured values and simulated results for LDK's temperature, absolute humidity, and relative humidity of Case 1 (exhaust ventilation system)



(a) Temperature

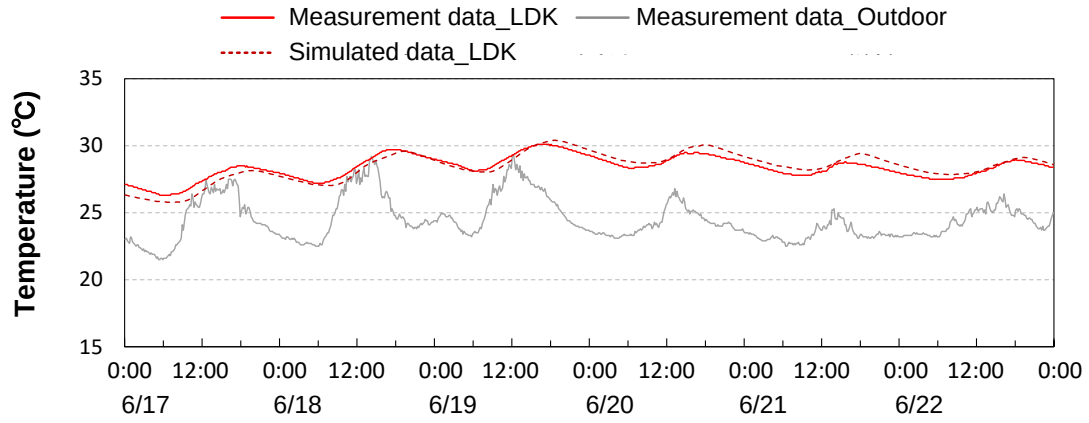


(b) Absolute humidity

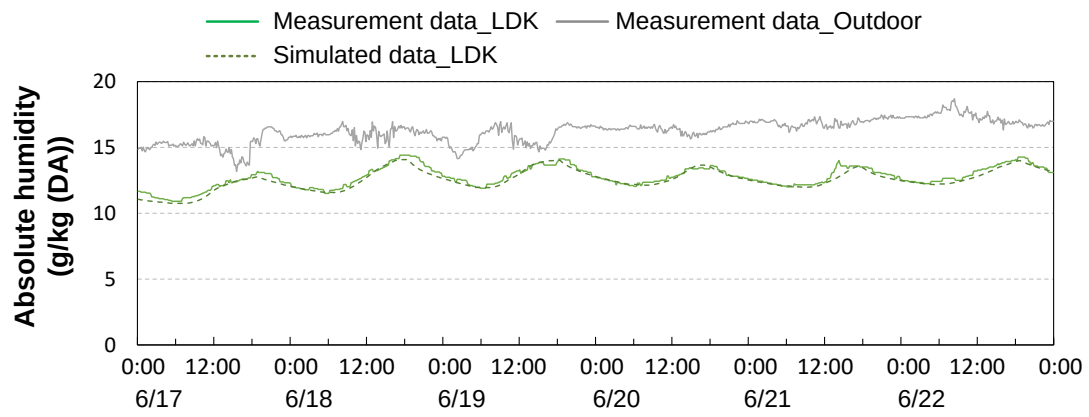


(c) Relative humidity

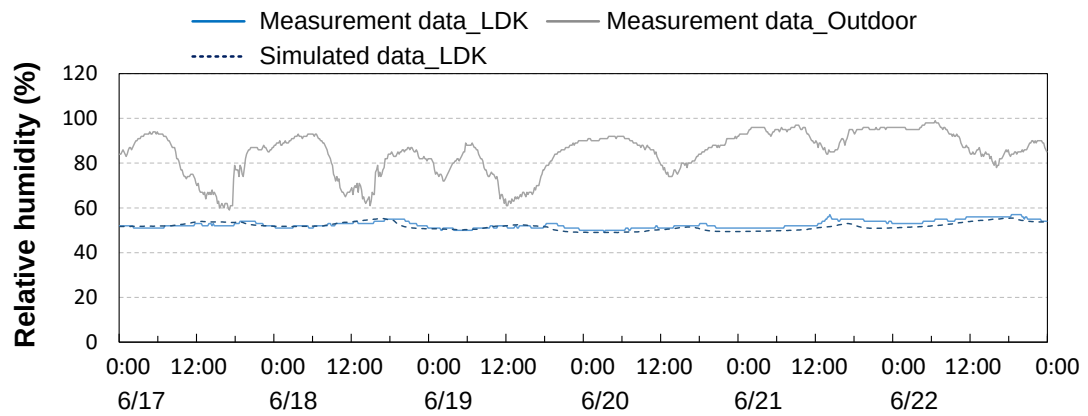
Figure 4.2 Comparisons between the measured values and simulated results for LDK's temperature, absolute humidity, and relative humidity of Case 2 (ERV system)



(a) Temperature

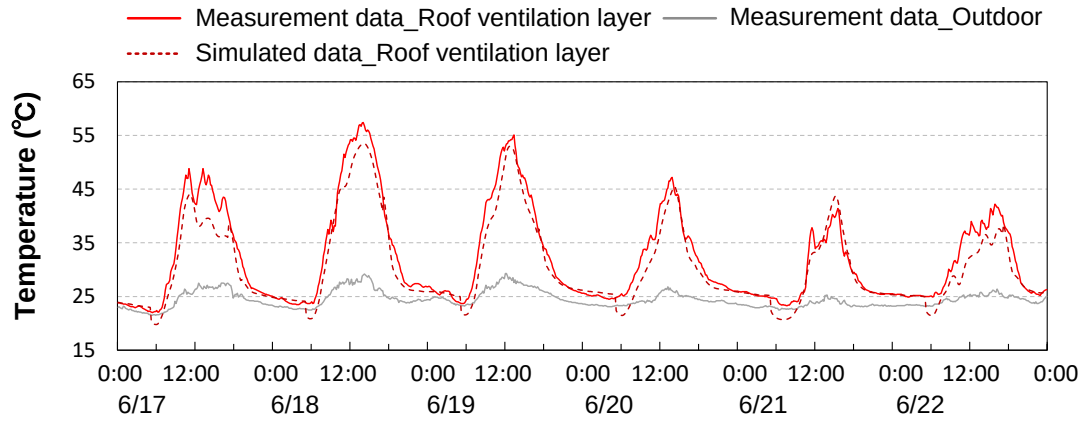


(b) Absolute humidity

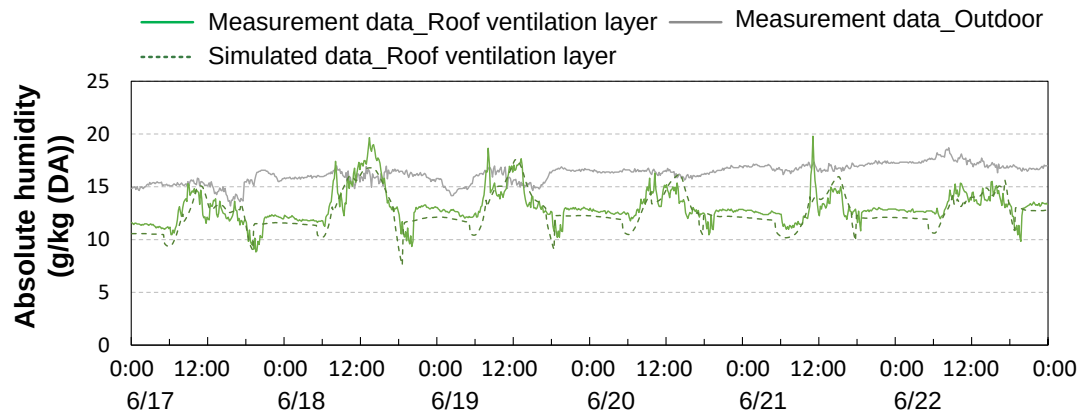


(c) Relative humidity

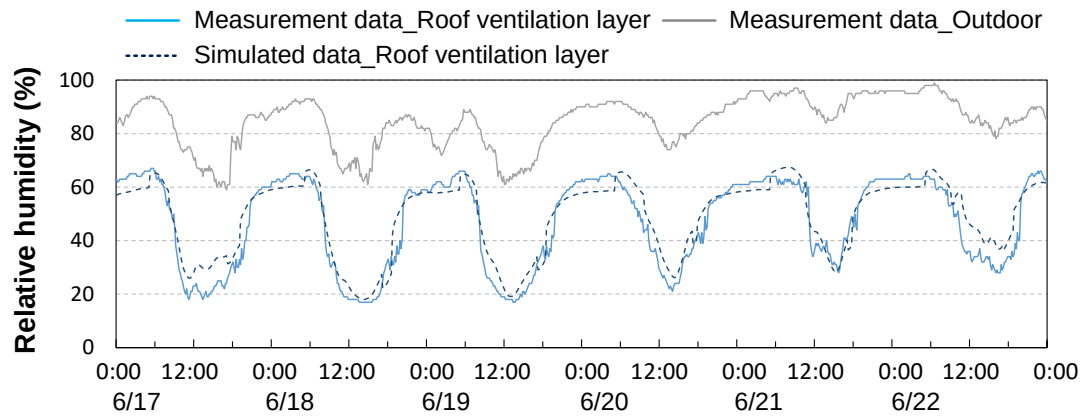
Figure 4.3 Comparisons between the measured values and simulated results for LDK's temperature, absolute humidity, and relative humidity of Case 3 (the PSE system)



(a) Temperature



(b) Absolute humidity



(c) Relative humidity

Figure 4.4 Comparisons between the measured values and simulated results for roof ventilation layer's temperature, absolute humidity, and relative humidity of Case 3 (the PSE system)

Table 4.2 Prediction accuracy of the model

Models	Item	MAPE of temperature (%)	MAPE of absolute humidity (%)	MAPE of relative humidity (%)	Average (%)
Exhaust ventilation system model	LDK	0.63	2.80	2.64	2.02
ERV model	LDK	1.02	2.55	2.94	2.17
PSE model	LDK	1.30	1.65	2.40	1.78
	Roof ventilation layer	5.93	7.42	13.01	8.79

4.1.3 Simulation conditions

Most existing buildings in Japan are constructed with vapor barriers on the interior of the envelope to prevent moisture adsorption and desorption, thereby avoiding condensation within walls. However, the validated house models are based on an experimental house designed with a building envelope that facilitates moisture adsorption and desorption. Consequently, it is necessary to design a standard house model reflecting typical Japanese house envelope characteristics for comparison, specifically a building with an envelope lacking moisture adsorption and desorption functions.

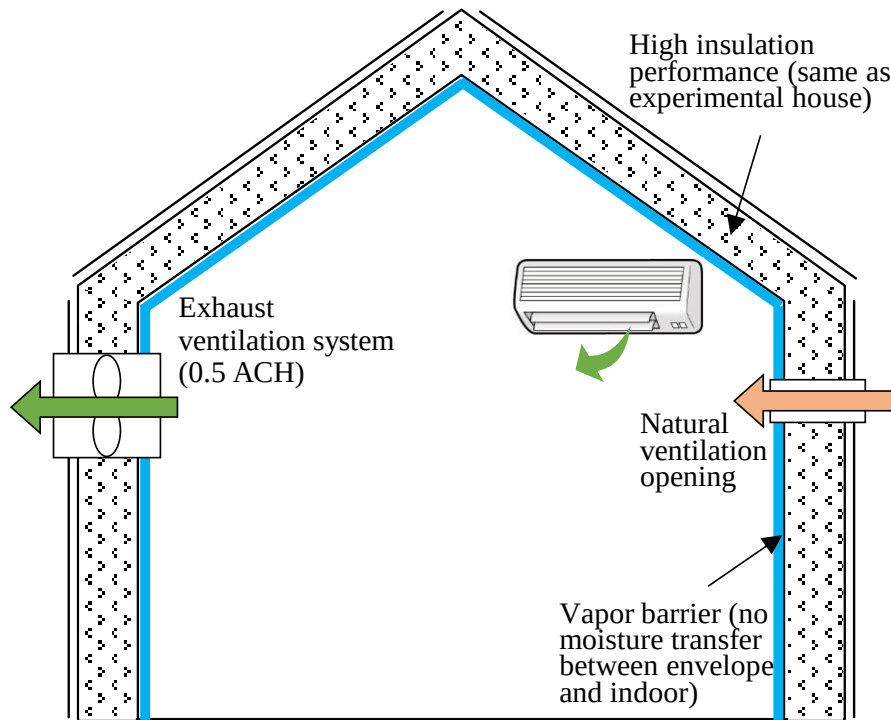
The simulation cases are presented in Table 4.3, divided into Cases A (Case A-1-1~A-2-4) without air-conditioning and Cases B (Case B-1-1~B-2-4) with air-conditioning, each comprising eight separate cases. The conceptual diagram of the numerical model for Cases A-1-1~A-2-4 is shown in Figure 4.5, including the three validated models in Subsection 4.1.2 (Cases A-2-1, A-2-3, and A-2-4). In Case A, the house models are divided into the Case A-1 and Case A-2 series. The Case A-1 series includes four house models with various ventilation systems (exhaust ventilation, PDSC, ERV, and PSE systems), but the building envelope cannot adsorb and desorb moisture. However, while series Case A-2 comprises four house models with same ventilation systems but includes moisture adsorption and desorption properties in their envelopes. Notably, except the envelope characteristics and systems, the design and construction of these eight house models are identical, following the specifications of the target house as shown in Table 3.3 in Subsection 3.1.2.

The models in Case B are constructed the same as Case A, except that the indoor air-conditioning for Case B implements 24-hour cooling. For all simulations for Case B, the AC's setting temperature is 27 °C, aligning with the comfortable indoor conditions suggested by the Building Health Law in Japan [16]. The AC's relative humidity is set at 65%, the upper limit recommended by ASHRAE (ASHRAE recommended RH range: 40%–65% [17]).

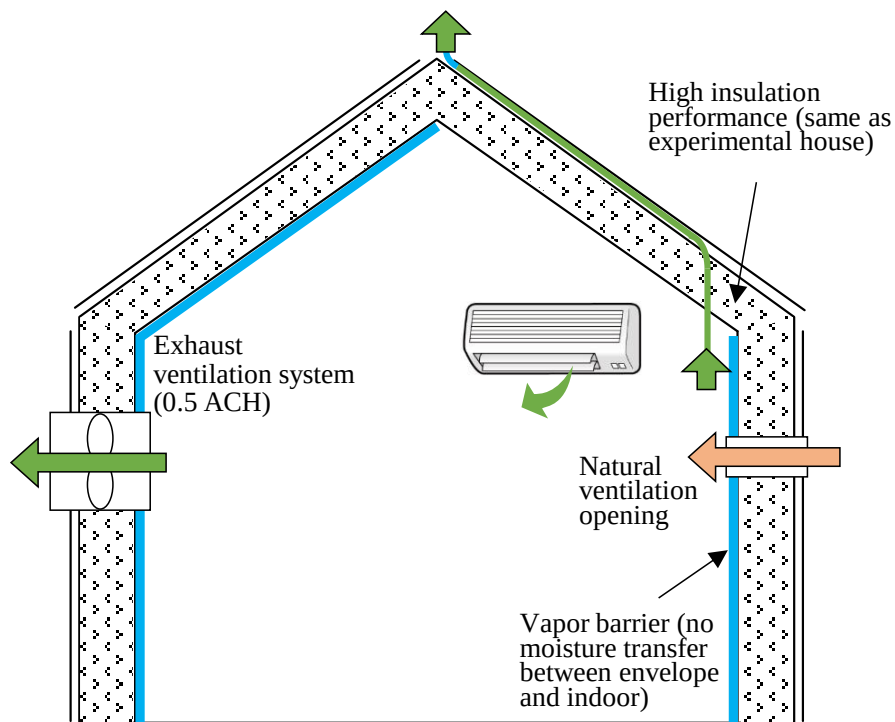
Table 4.3 Simulation cases

Cases	With AC operating	Consider the moisture adsorption and desorption function of the building envelope	The operation of independent PDSC system	The operation of independent ERV system	The operation of PSE (PDSC combined with ERV) system
Case A-1-1	No	No	-	-	-
Case A-1-2			Operation		
Case A-1-3			-	Operation	Operation
Case A-1-4				-	
Case A-2-1*		Consider	-	-	-
Case A-2-2			Operation		
Case A-2-3*			-	Operation	Operation
Case A-2-4*				-	
Case B-1-1	Operation (constant temperature = 27 °C ; Constant relative humidity = 65%)	No	-	-	-
Case B-1-2			Operation		
Case B-1-3			-	Operation	Operation
Case B-1-4				-	
Case B-2-1		Consider	-	-	-
Case B-2-2			Operation		
Case B-2-3			-	Operation	Operation
Case B-2-4				-	

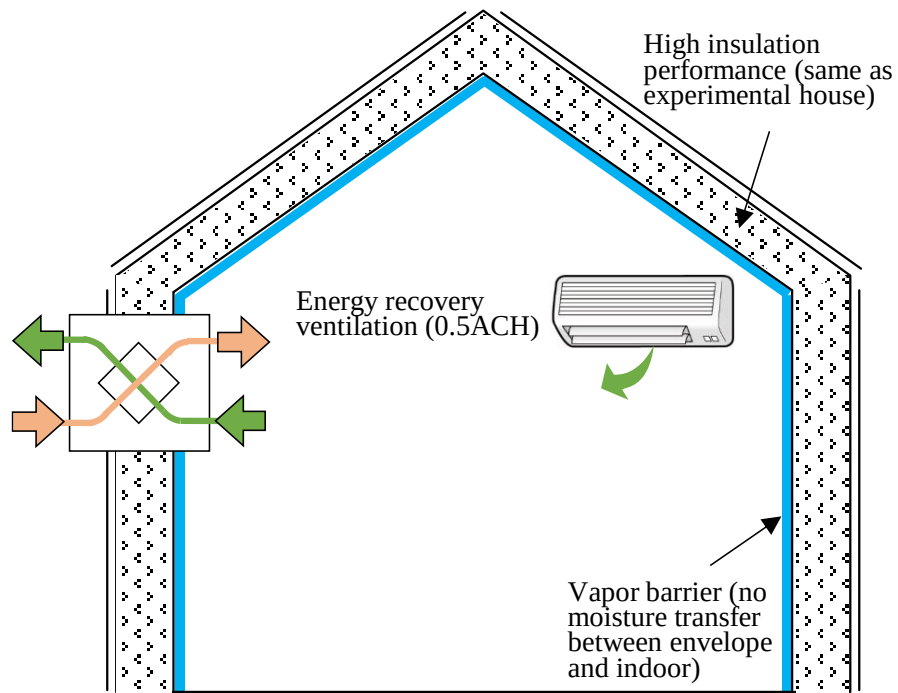
* Note: Model specifications used to validation in Section 4.1.2



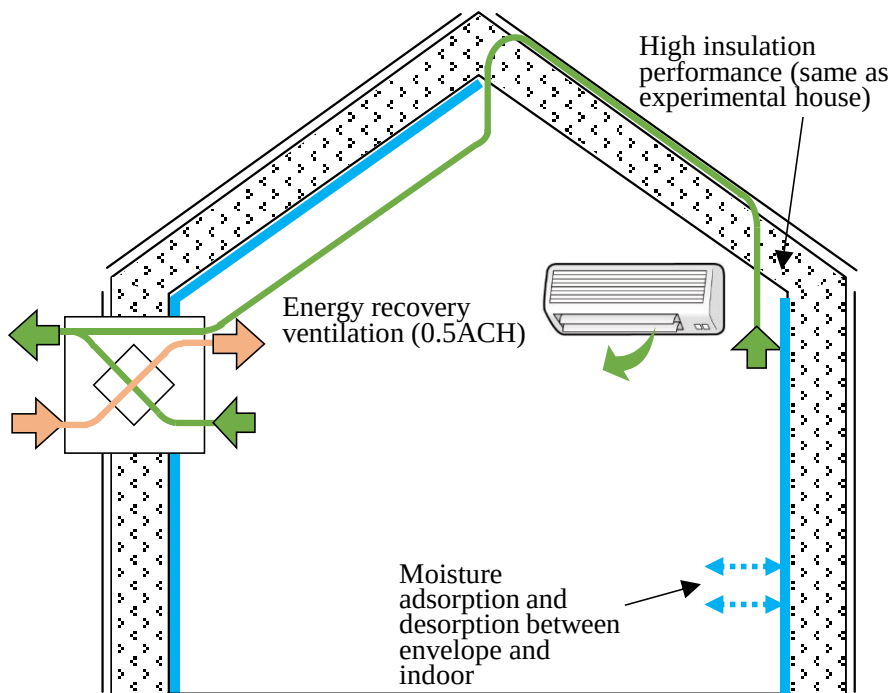
(a) Exhaust ventilation system (not considering the moisture adsorption and desorption)



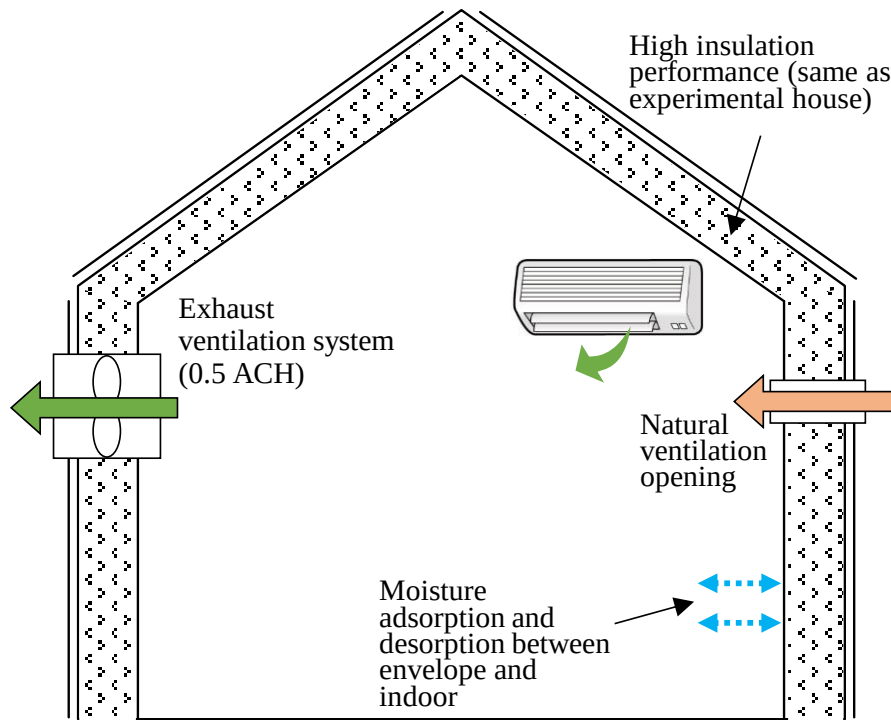
(b) PDSC system (not considering the moisture adsorption and desorption)



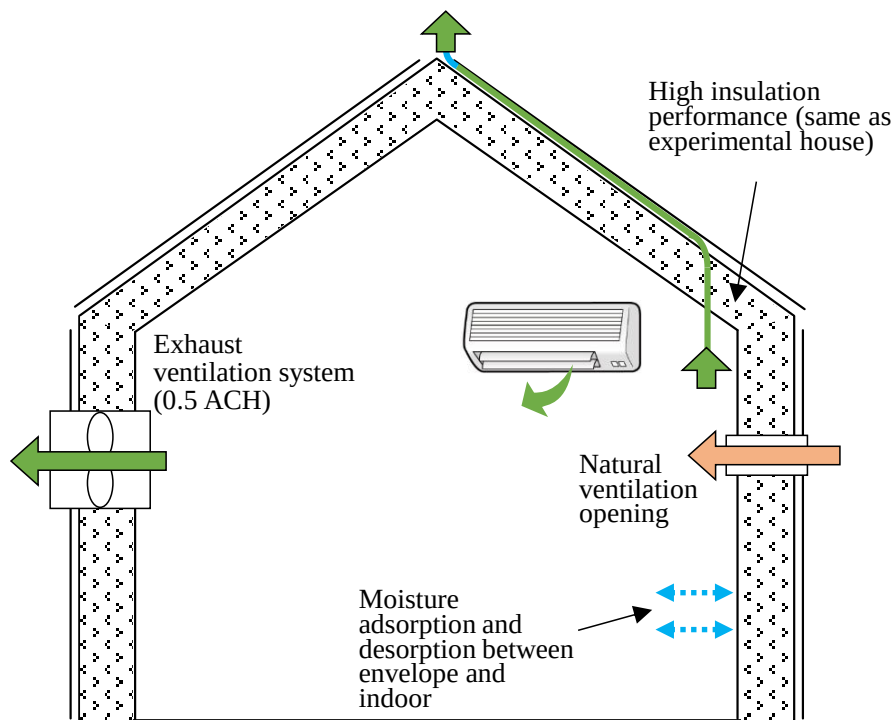
(c) ERV system (not considering the moisture adsorption and desorption)



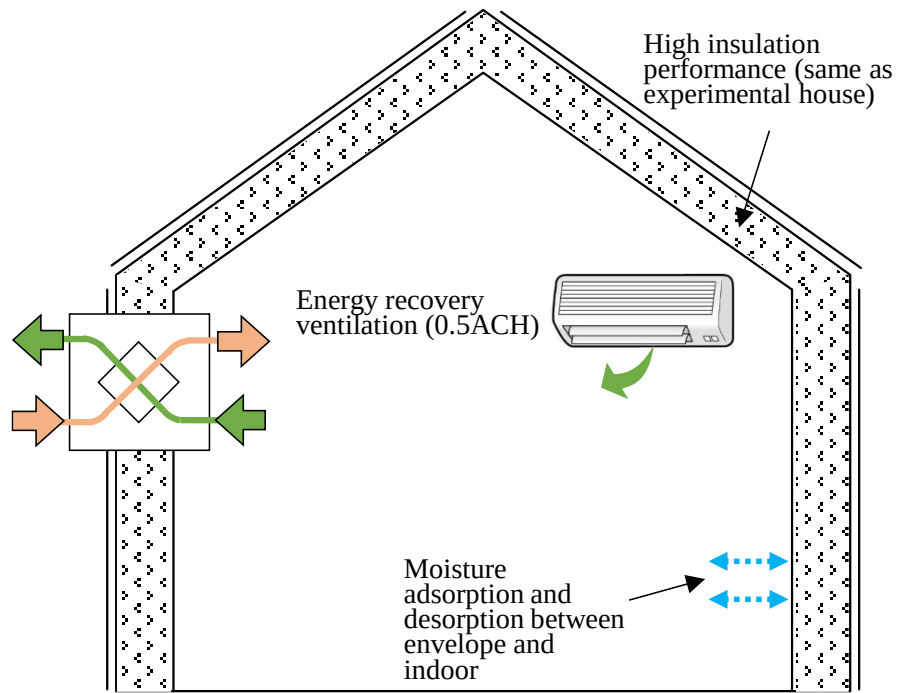
(d) PSE system (not considering the moisture adsorption and desorption)



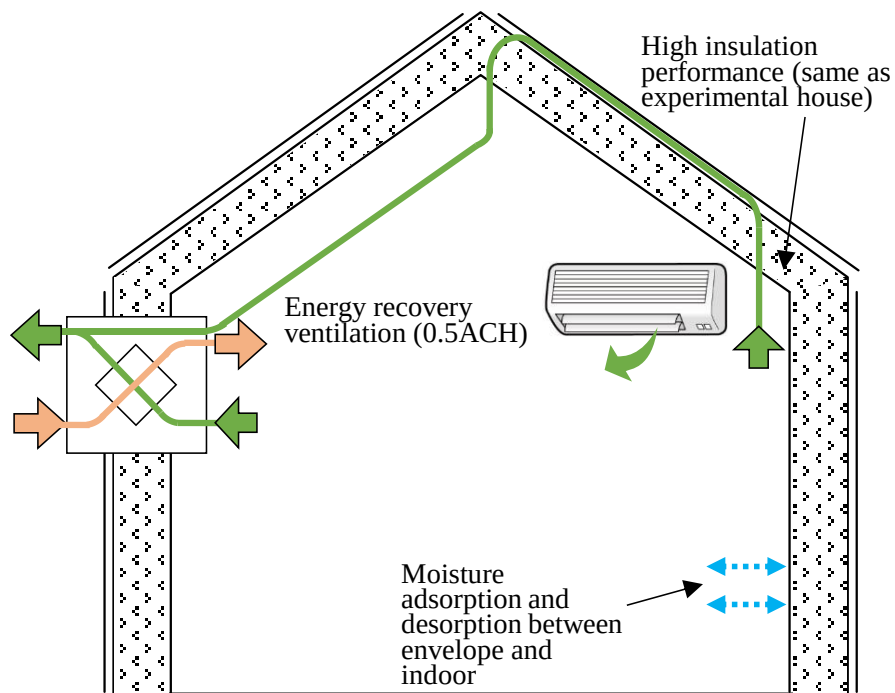
(e) Exhaust ventilation system (considering the moisture adsorption and desorption)



(f) PDSC system (considering the moisture adsorption and desorption)



(g) ERV system (considering the moisture adsorption and desorption)



(h) PSE system (considering the moisture adsorption and desorption)

Figure 4.5 Model diagrams for simulation

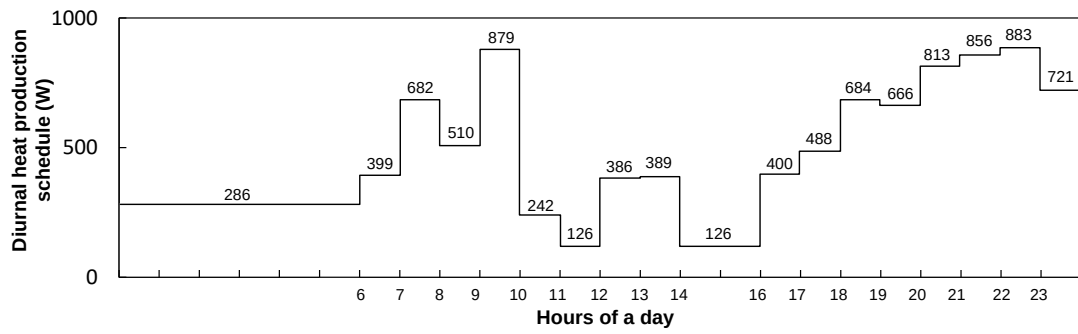
Chapter 4 Simulation analysis by using numerical simulation under several conditions

The simulation conditions for all Cases A and B are listed in Table 4.4. The weather data utilized standardized annual data from the Japan Meteorological Agency, including outside temperature, relative humidity, insolation, wind speed, and wind direction [12]. Fukuoka was selected as the target region, characterized by a humid subtropical climate with hot and humid summers [18]. To gather comprehensive simulation data, the simulation period extended throughout the summer, from June 1 to September 30. In addition, a three-month preparatory calculation was conducted to negate the influence of the initial settings on the indoor environment.

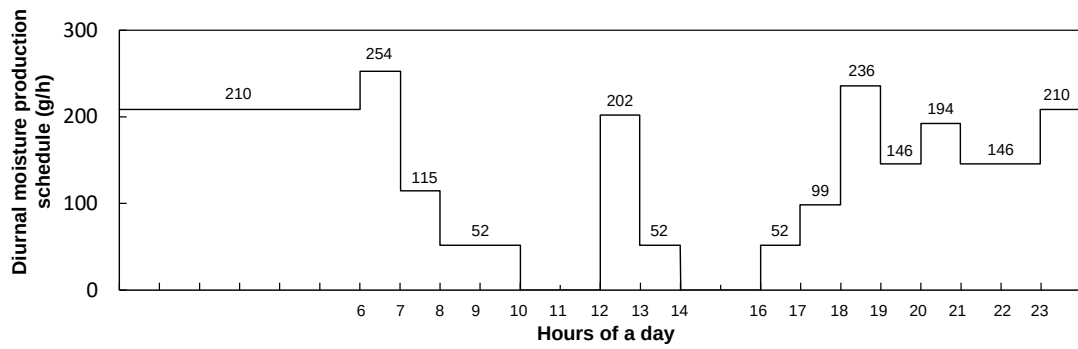
Internal moisture and heat were consistently generated. Figure 4.6 shows the daily production schedule for the entire house, adhering to a standard energy consumption calculation regulation [19] issued by Japan. Notably, the indoors were presumed to be occupied by a family of four, with water vapor and heat sources emanating from human activity, appliances, and lighting equipment. Table 4.5 shows the detailed heat and moisture generation schedule. The ventilation rate for all models was set at 0.5 ACH (approximately 170 m³/h), the minimum rate required for residences as mandated by the Basic Law of Construction in Japan [20]. In the building model equipped solely with an ERV system (Cases A-1-3, A-2-3, B-1-3, and B-2-3), the sensible and latent heat exchange efficiencies were set at 0.8 and 0.58, respectively. For the PSE system building model (Cases A-1-4, A-2-4, B-1-4, and B-2-4), the sensible and latent heat exchange efficiencies were 0.4 and 0.5 during the day, 0.8 and 0.58 at night, respectively.

Table 4.4 Simulation conditions for Cases A-1-1~A-2-4

Items	Contents
Simulation cases	Refer to Table 4.3
Simulation duration	June 1 to September 30 (trial calculation from March 1 to May 31)
Simulation interval	1 min
Weather data	Fukuoka Expanded AMeDAS weather data of Japan (standard year, 2001–2010) [12]
Building envelope materials	Refer to Table 3.3
Ventilation rate	Constant ventilation rate: 0.5 ACH (approximately 170 m ³ /h)
ERV's heat exchange effectiveness	In ERV building model: 0.8 (sensible) and 0.58 (latent) In PSE building model: 0.4 (sensible) and 0.5 (latent) during the daytime, 0.8 (sensible) and 0.58 (latent) at nighttime.
Internal moisture and heat generation	Refer to Figure 4.6.



(a) Diurnal heat production schedule of entire house



(b) Diurnal moisture production schedule of entire house

Figure 4.6 Diurnal heat and moisture production schedule of entire house for simulations—four-person family

Table 4.5 Detailed heat and moisture generation schedule

Time	Human loads		Machines		Lighting equipment		Total	
	Sensible heat (W)	Latent heat (g/h)	Sensible heat (W)	Latent heat (g/h)	Sensible heat (W)	Latent heat (g/h)	Sensible heat (W)	Latent heat (g/h)
0:00	159	210	126	0	0	-	286	210
1:00	159	210	126	0	0	-	286	210
2:00	159	210	126	0	0	-	286	210
3:00	159	210	126	0	0	-	286	210
4:00	159	210	126	0	0	-	286	210
5:00	159	210	126	0	0	-	286	210
6:00	159	210	144	44	96	-	399	254
7:00	87	115	382	0	212	-	682	115
8:00	39	52	346	0	125	-	510	52
9:00	39	52	502	0	337	-	879	52
10:00	0	0	177	0	65	-	242	0
11:00	0	0	126	0	0	-	126	0
12:00	39	52	245	150	102	-	386	202
13:00	39	52	278	0	72	-	389	52
14:00	0	0	126	0	0	-	126	0
15:00	0	0	126	0	0	-	126	0
16:00	39	52	227	0	133	-	400	52
17:00	75	99	278	0	134	-	488	99
18:00	111	146	363	90	209	-	684	236
19:00	111	146	329	0	226	-	666	146
20:00	147	194	374	0	292	-	813	194
21:00	111	146	451	0	293	-	856	146
22:00	111	146	403	0	369	-	883	146
23:00	159	210	388	0	174	-	721	210

4.1.4 Simulation results

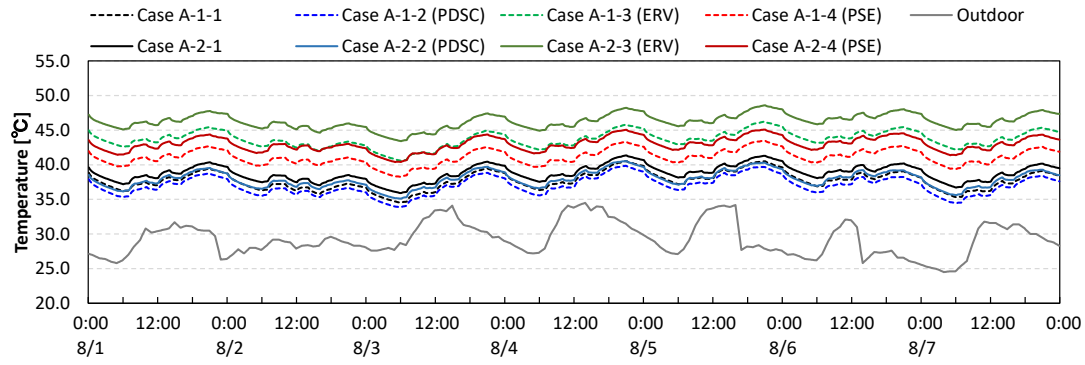
4.1.4.1 Case A: without air-conditioning

Figure 4.7 depicts the temporal variations in temperature, absolute humidity, and relative humidity for Cases A-1-1~A-2-4 from 1 to 7 August (standard year, 2001–2010), during which air-conditioning was not used. According to the temperature graph (Figure 4.7(a)), the outdoor air temperature is approximately 30 °C. Due to the building envelope's high insulation effect and internal heat generation, all cases exhibit temperatures exceeding the outdoor air temperature in the absence of air-conditioning. Notably, the indoor temperature of Case A-1-2 (PDSC system) is marginally lower than that of A-1-1 (exhaust ventilation system), owing to the absence of the building envelope's moisture adsorption and desorption capability. This difference results from the PDSC system's daytime air circulation path, which attenuates convective heat transfer inside the roof and benefits from a radiative cooling effect at night. Conversely, the temperature in A-1-3 (ERV system) is approximately 6 °C higher than in A-1-1 due to the indoor air temperature being high and the ERV's heat recovery effect, which warms the incoming fresh air by recovering heat from the outgoing indoor air. When the PDSC and ERV systems are combined, the indoor temperature of the PSE residence is approximately 3 °C lower than that of the ERV, attributed to the PDSC system's radiative cooling effect.

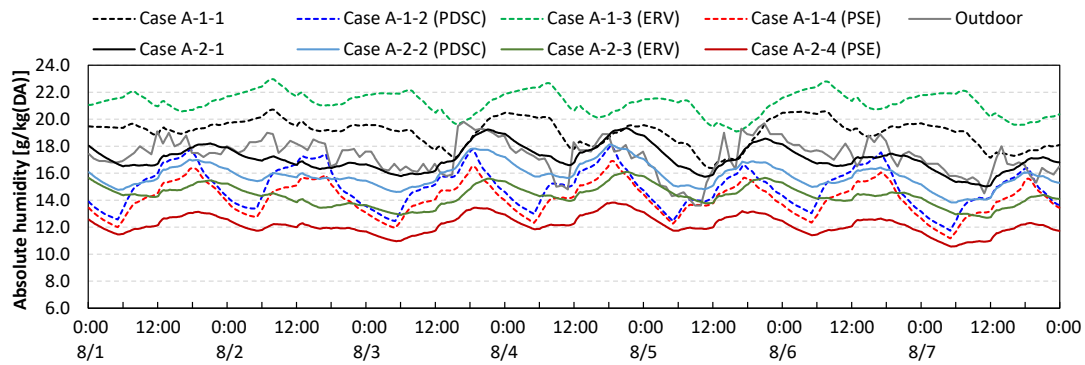
With the building envelope possessing moisture adsorption and desorption capabilities, the ranking trend of indoor temperatures across different ventilation systems mirrors that of Case A-1, with ERV recording the highest temperatures, followed by the PSE system, the exhaust ventilation system, and the PDSC system being the lowest. It is noteworthy that, overall, the indoor temperatures under conditions with a hygroscopic effect are higher than those without, e.g., Case A-2-1 exhibits higher temperatures than Case A-1-1. This increase is due to the heat generated during moisture adsorption by the building materials.

In the absolute humidity graph (Figure 4.7(b)), without moisture effects from the envelope, the indoor absolute humidity in Case A-1-1 (exhaust ventilation system) slightly exceeds outdoor levels, with a fluctuation pattern similar to that of the outdoor absolute humidity. This similarity is attributed to moisture generated indoors and ventilation. The absolute humidity in Case A-1-3 exceeds that of Case A-1-1, due to the ERV's moisture recovery effect, which humidifies incoming fresh air. When the PDSC system operates, the absolute indoor humidity of Case A-1-2 decreases, particularly at night, when the PDSC system's moisture adsorption effect is pronounced. In Case A-1-4, where the PDSC system and ERV are integrated, the absolute indoor humidity falls below outdoor levels, benefiting from the PDSC system's dehumidification effect and the ERV's utilization of the indoor-outdoor absolute humidity difference to pre-dehumidify incoming fresh air. Consequently, the PSE system demonstrates the most effective dehumidification.

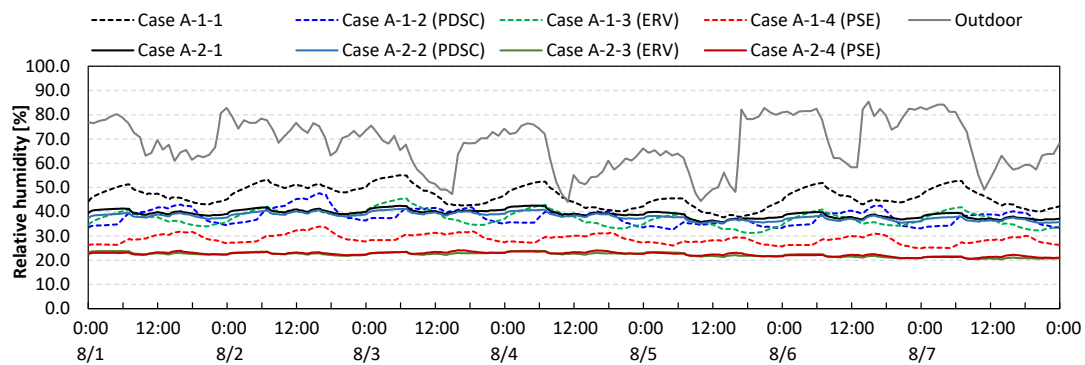
In the relative humidity graph (Figure 4.7(c)), indoor relative humidity is influenced by both temperature and absolute humidity. Indoor humidity fluctuations in buildings with hygroscopic properties are smoother than in those without. The PSE system achieves the lowest relative humidity, approximately 22%.



(a) Temperature



(b) Absolute humidity



(c) Relative humidity

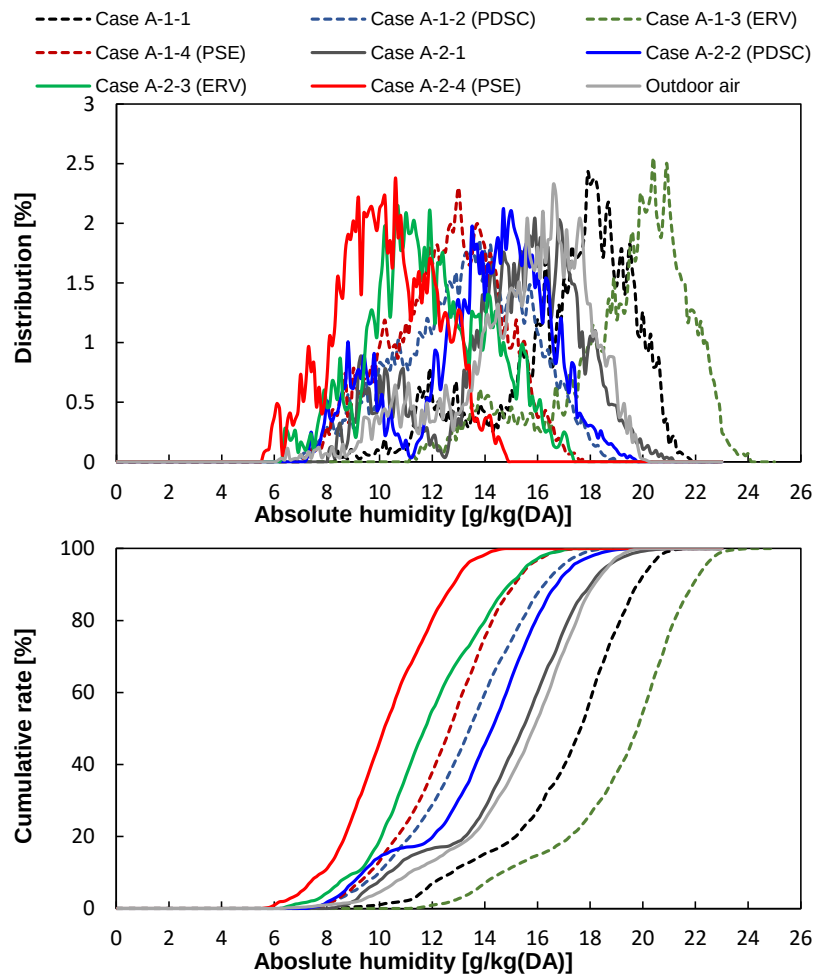
Figure 4.7 Temporal changes in temperature and humidity in LDK for Cases A-1-1~A-2-4—without air-conditioning (August 1 to 7)

Nevertheless, establishing a clear picture of the indoor humidity distribution range is challenging due to the short time period analyzed. Figure 4.8 illustrates the frequency distribution (in 0.1 g/kg (DA) intervals), cumulative rate, and boxplot of absolute humidity in the LDK for Cases A-1-1~A-2-4 from July 1 to September 30 (standard year 2001–2010), when air-conditioning was not in use. The frequency distribution graph shows the percentage (y-axis) of the number of measured values corresponding to the values on the x-axis. The cumulative rate graph depicts the percentage (y-axis) of the measured values that are lower than those corresponding to the x-axis. The numbers on the boxplot indicate average values.

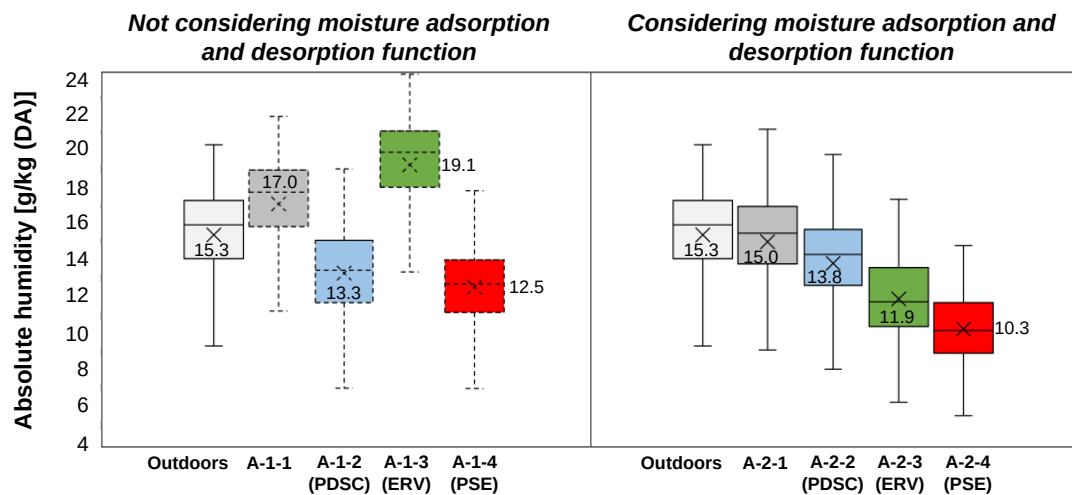
In line with the results from the temporal change chart, the PSE system proved to be more effective at dehumidifying than the exhaust ventilation, ERV, and PDSC systems during the summer months, in the absence of air-conditioning cooling and dehumidification. This effectiveness was observed regardless of the residential envelope's hygroscopic performance. The indoor/outdoor absolute humidity difference for the PSE system was 2.8 to 5.0 g/kg (DA) for buildings without and with the envelope's moisture adsorption and desorption capability, respectively. The PSE system's dehumidification was more pronounced when the envelope included moisture adsorption and desorption capabilities, as the building's hygroscopic materials adsorbed moisture from the indoor environment.

Figure 4.9 shows the frequency distribution (in 1% intervals), cumulative rate, and boxplot of relative humidity in the LDK for Cases A-1-1~A-2-4 from July 1 to September 30 (standard year, 2001–2010), without air-conditioning. Due to the high indoor temperatures in the absence of air-conditioning cooling, the indoor relative humidity remained low across the cases. Specifically, the indoor relative humidity of the PSE system reached as low as 29.2% and 21.4% in Cases A-1-4 and A-2-4, respectively.

Case A: Absolute humidity



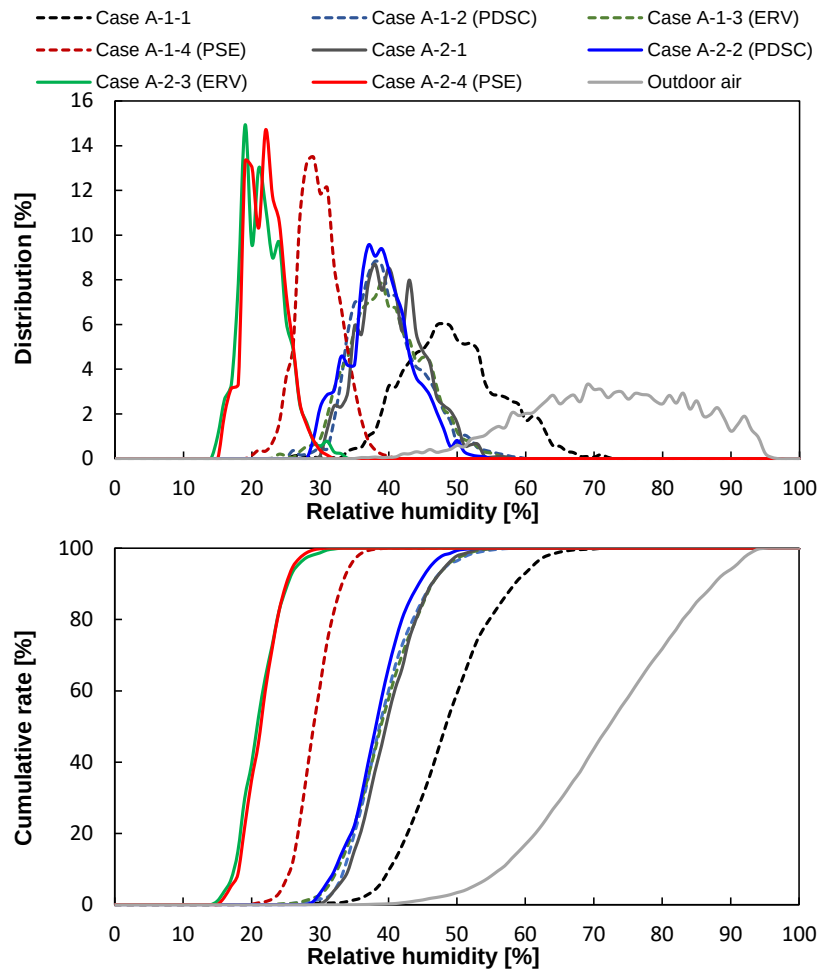
(a) Frequency distribution and cumulative rate



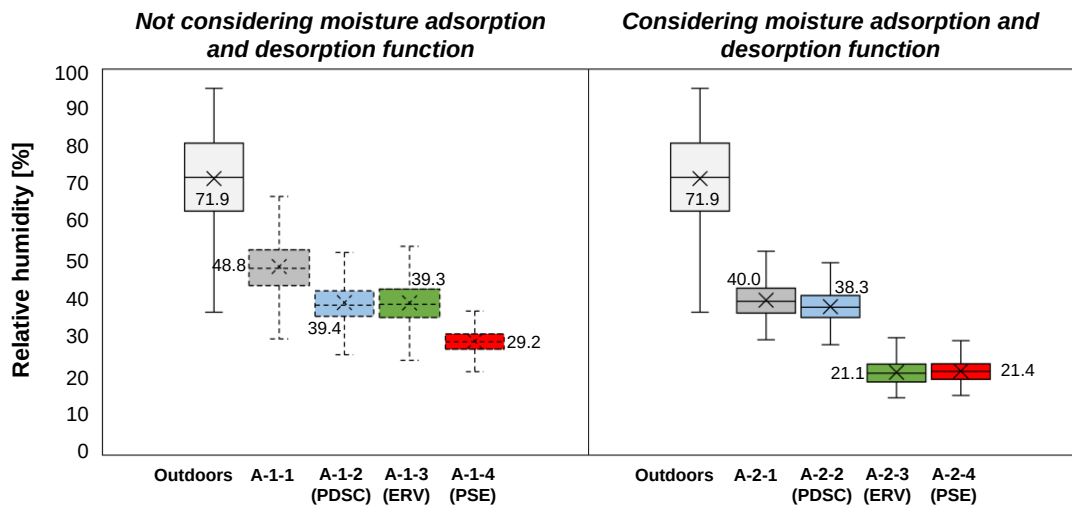
(a) Boxplot

Figure 4.8 Frequency distribution, accumulation rate, and boxplot of absolute humidity for Cases A-1-1~A-2-4 (July 1 to September 30)

Case A: Relative humidity



(a) Frequency distribution and cumulative rate



(a) Boxplot

Figure 4.9 Frequency distribution, accumulation rate, and boxplot of relative humidity for Cases A-1-1~A-2-4 (July 1 to September 30)

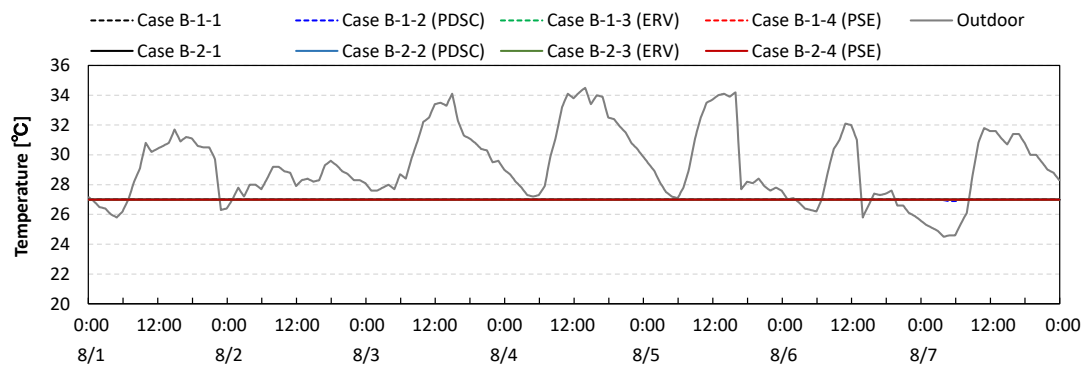
4.1.4.2 Case B: with air-conditioning of constant setting temperature and relative humidity

Figure 4.10 displays the temporal changes in temperature, absolute humidity, and relative humidity for Cases B-1-1~B-2-4 from 1 to 7 August (standard year, 2001–2010), during which the air-conditioning was set to 27 °C and 65% RH.

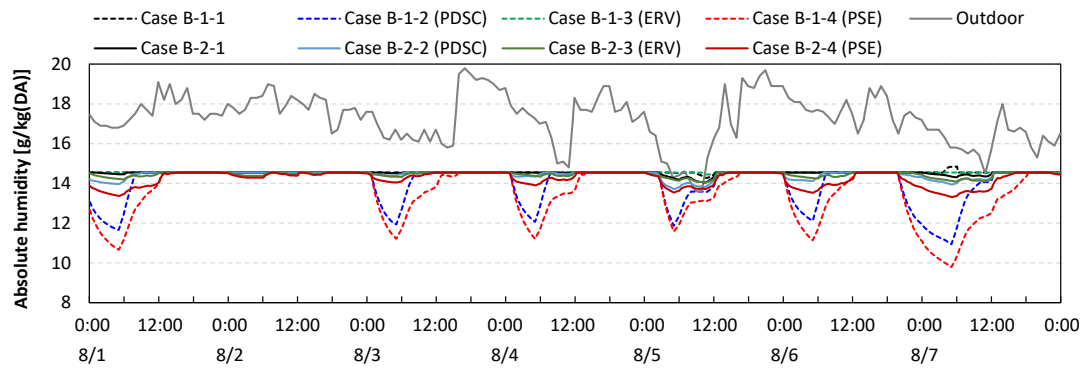
In Figure 4.10(a), due to initially high room temperatures, the air conditioner ran continuously throughout the hot month of August, maintaining the temperature in all cases at 27 °C.

Figure 4.10(b) shows that the air conditioner dehumidifies only when the indoor relative humidity exceeds 65%. When the building envelope lacks moisture adsorption and desorption properties (Cases B-1-1~B-1-4), the indoor absolute humidity levels for the exhaust ventilation (Case B-1-1) and ERV (Case B-1-3) systems are similar. However, there is a significant reduction in indoor absolute humidity with the PDSC system (Case B-1-2), which utilizes indoor air circulation to adsorb moisture at night and expels water vapor outside during the day by leveraging the water potential gradient between the indoors and the roof. The combination of the PDSC system and ERV (Case B-1-4) achieves the lowest indoor humidity levels, particularly at night, enhancing the PSE system's dehumidification effect due to the increasing absolute humidity difference between the PDSC system and the outdoors. Notably, indoor absolute humidity levels in Cases B-2-2 and B-2-4 are higher than those in Cases B-1-2 and B-1-4. This increase occurs because the dehumidification effect of both the PDSC and PSE systems overly dries the indoor environment, causing building materials to release moisture back indoors. Furthermore, the PSE system consistently exhibits the lowest indoor absolute humidity and the most effective dehumidification, regardless of the building envelope's moisture adsorption and desorption capabilities.

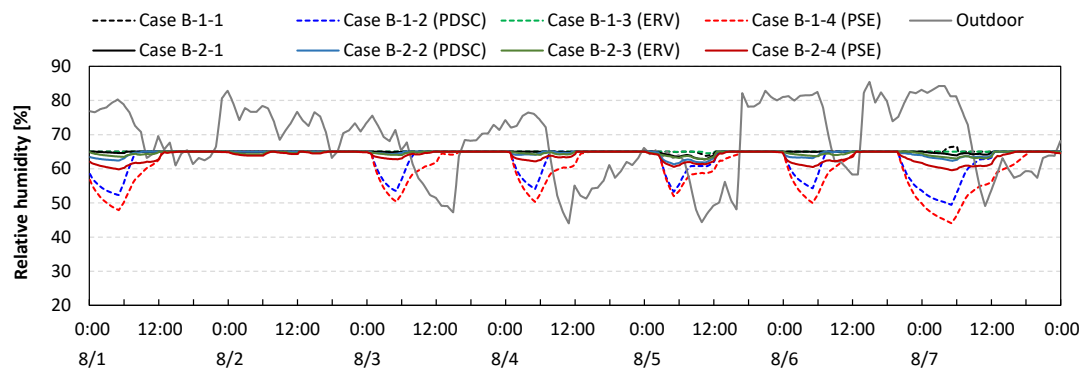
In Figure 4.10(c), the patterns observed in indoor relative humidity mirror those of absolute humidity, with greater fluctuations in Cases B-1-1~B-1-4. Conversely, in Cases B-2-1~B-2-4, indoor relative humidity fluctuations are more subdued and smoother. The PSE system maintains the lowest relative humidity, whether or not the building envelope's moisture adsorption and desorption capabilities are considered. Additionally, the relative humidity of the PSE system without considering moisture adsorption and desorption effects (Case B-1-4) is lower than that with such effects considered (Case B-2-4).



(a) Temperature



(b) Absolute humidity



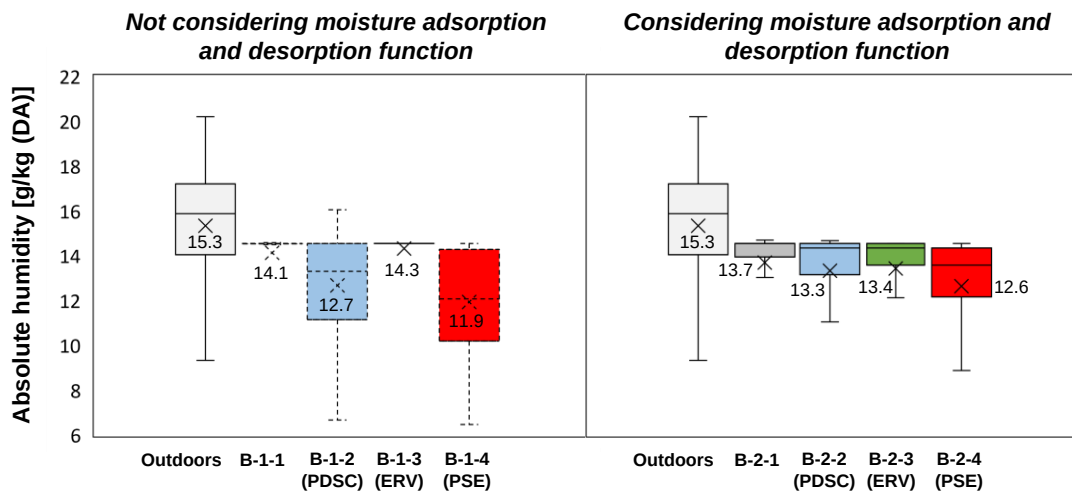
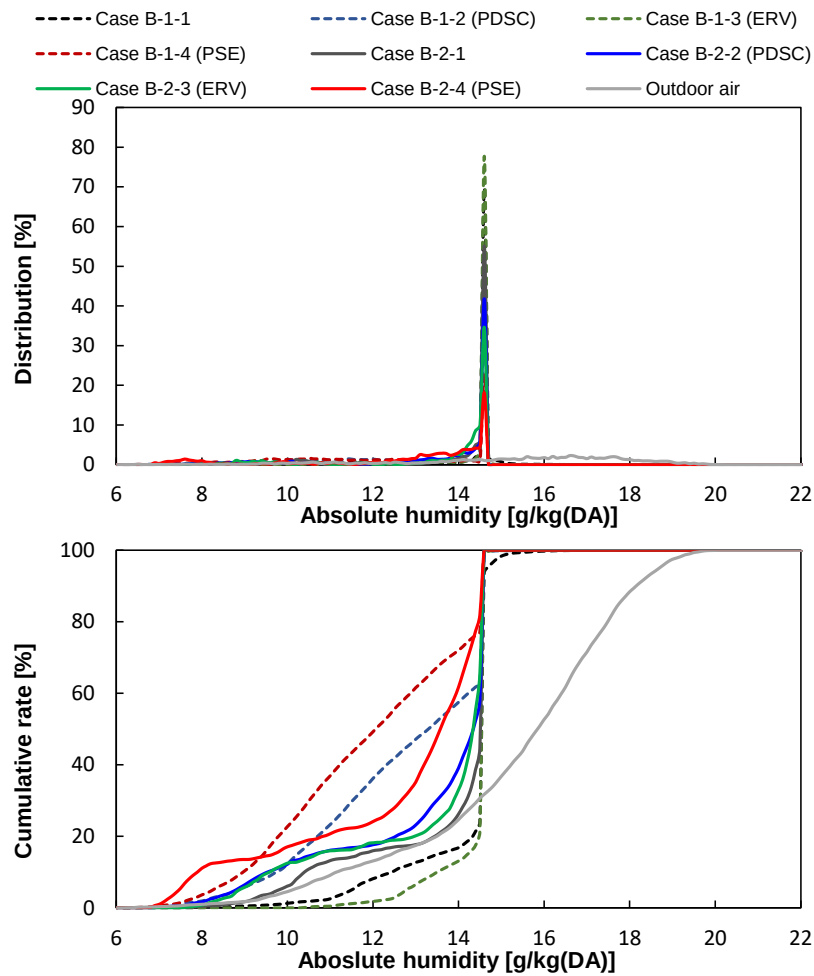
(c) Relative humidity

Figure 4.10 Temporal changes in temperature and humidity in LDK for Cases B-1-1~B-2-4—with air-conditioning (August 1 to 7)

Figure 4.11 illustrates the frequency distribution (in 0.1 g/kg (DA) intervals), cumulative rate, and boxplot of absolute humidity in the LDK for Cases B-1-1~B-2-4 from July 1 to September 30 (standard year 2001–2010), with air-conditioning operational. Utilizing an ideal air-conditioning model with constant temperature and relative humidity settings in simulations, indoor absolute humidity was maintained below 14.6 g/kg (DA) for most cases. In scenarios where the building envelope lacks moisture adsorption and desorption capabilities, combined with 24-hour continuous ventilation, the PDSC system achieves exceptional dehumidification, reducing the mean indoor absolute humidity to approximately 2.6 g/kg (DA) below the outdoor level. When the PSE system—a combination of PDSC and ERV systems—is activated, the ERV capitalizes on the humidity difference created by the PDSC to pre-dehumidify fresh air, achieving the most effective dehumidification with a mean indoor absolute humidity approximately 3.4 g/kg (DA) lower than the outside air. In Cases B-2-1~B-2-4, where the building envelope possesses moisture adsorption or desorption capabilities, the PSE system records the lowest indoor humidity, outperforming due to efficient dehumidification and a notable absolute humidity difference from outdoor air, approximately 2.7 g/kg (DA).

Figure 4.12 displays the frequency distribution (in 1% intervals), cumulative rate, and boxplot of relative humidity in the LDK for Cases B-1-1~B-2-4 spanning July 1 to September 30 (standard year 2001–2010), with air-conditioning operational. Mirroring the absolute humidity findings, the PSE system consistently shows the lowest relative humidity among the ventilation systems under identical envelope characteristics. Remarkably, buildings without moisture adsorption and desorption characteristics, utilizing the PSE system, exhibit even lower relative humidity, achieving an average indoor/outdoor relative humidity differential of approximately 18.3%.

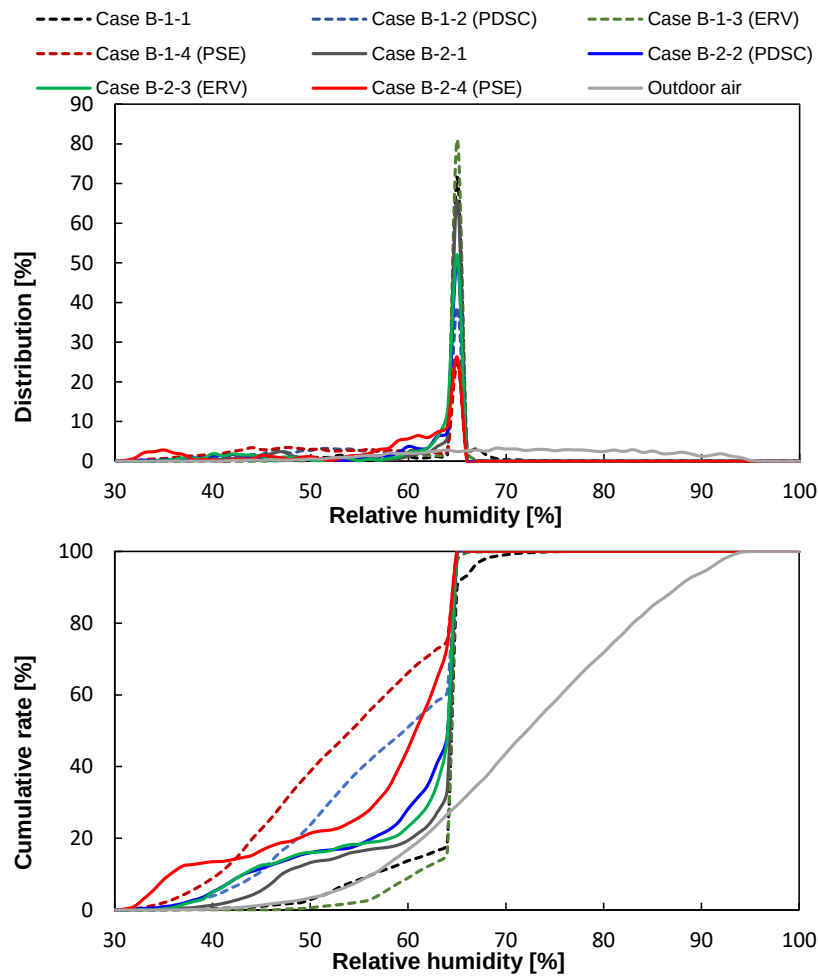
Case B: Absolute humidity



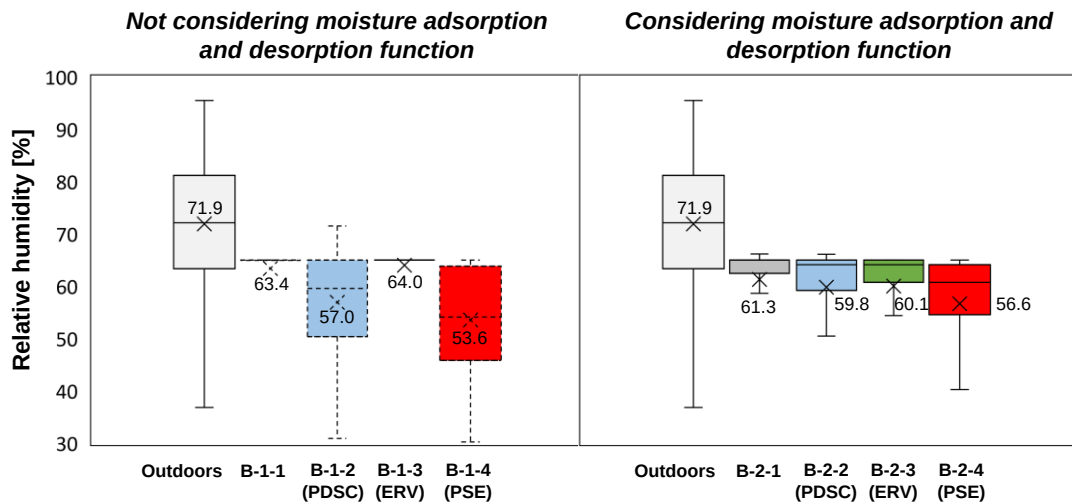
(b) Boxplot

Figure 4.11 Frequency distribution, accumulation rate, and boxplot of absolute humidity for Cases B-1-1~B-2-4 (July 1 to September 30)

Case B: Relative humidity



(a) Frequency distribution and cumulative rate



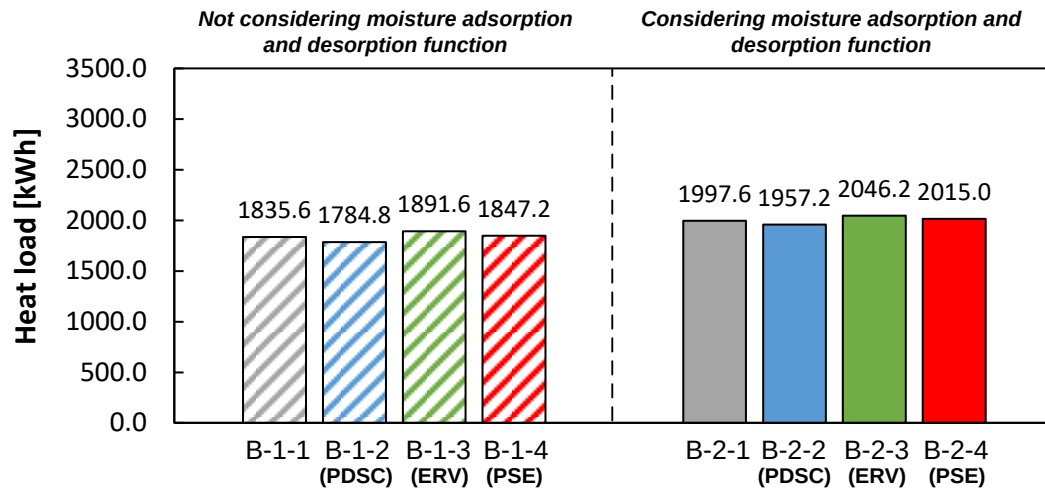
(b) Boxplot

Figure 4.12 Frequency distribution, accumulation rate, and boxplot of relative humidity for Cases B-1-1~B-2-4 (July 1 to September 30)

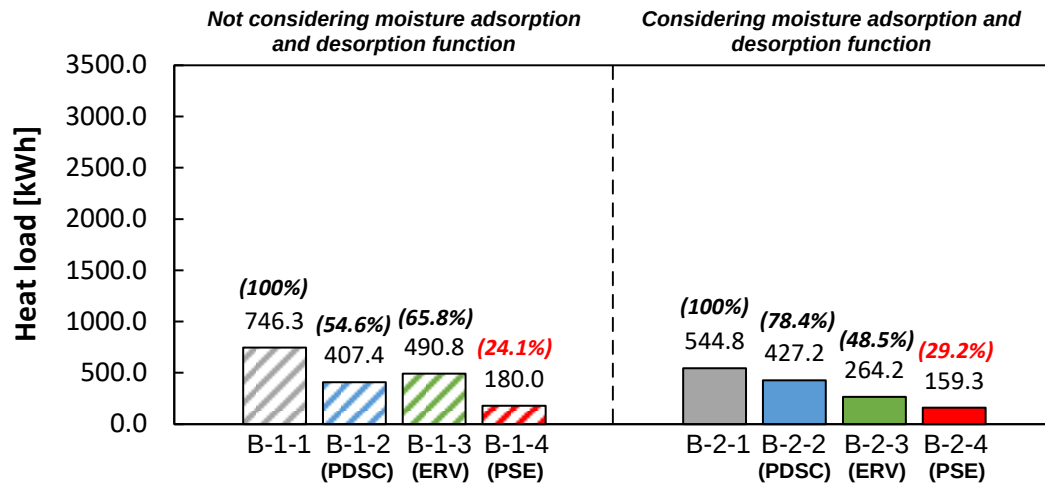
Figure 4.13 presents the simulated heat load results for Cases B-1-1~B-2-4 from July 1 to September 30. Regarding sensible heat load, as shown in Figure 4.15(a), the differences among the various ventilation systems are minor under identical building envelope characteristics. In Cases B-1-1~B-1-4, the PDSC system (Case B-1-2) showed a reduction in sensible heat load (by 50.8 kWh) compared to the exhaust ventilation system (Case B-1-1). This reduction is attributed to the suppression of temperature rise within the cavity by air circulation in the roof ventilation layer during the day and the potential for radiative cooling of indoor air at night. Conversely, operating under ERV-only conditions (Case B-1-3), the sensible heat load increased by 56.0 kWh compared to the exhaust system (Case B-1-1), as the ERV was operational for 24 h and the indoor temperature was set at 27 °C in this simulation. When outdoor air temperatures in Fukuoka fell below 27 °C at night, the ERV, while ventilating, recovered heat from the indoor air, thus increasing the cooling demand for the ERV system compared to the exhaust system. Although daytime pre-cooling by the ERV reduces the sensible heat load, the pre-heating effect at night counteracts it. Moreover, integrating the ERV with the PDSC system resulted in a slightly lower sensible heat load for the PSE system (Case B-1-4) than for the ERV (Case B-1-3), due to the PDSC system's mitigated heating and cooling effects. Furthermore, the sensible heat load for each system rose by 154.6 to 172.4 kWh when the building envelope had moisture adsorption and desorption capabilities, owing to the simulation accounting for heat released during moisture adsorption by porous building materials.

Contrary to the sensible heat load's incremental trend, a significant reduction in latent heat load is evident, as shown in Figure 4.13(b). In the Cases B-1-1~B-1-4, the PDSC system (Case B-1-3) reduced the latent heat by 338.9 kWh (45.5%) compared to exhaust ventilation (Case B-1-1), due to its effective moisture removal during the day and moisture adsorption at night. The ERV's latent heat load was 255.5 kWh (34.2%) less than the exhaust system (Case B-1-1), attributed to the ERV's pre-dehumidification effect, albeit less efficient than the PDSC-only scenario. The latent heat load reached its nadir (180.0 kWh) when the PDSC system was combined with the ERV to form the PSE system, reducing the latent heat load by 566.3 kWh, or approximately 75.9%, compared to the exhaust ventilation system. When moisture effects were incorporated in exhaust ventilation and ERV systems (Cases B-2-1 and B-2-3), their latent heat loads decreased compared to Cases B-1-1 and B-1-3, as the building's porous materials adsorbed moisture from the indoor air. However, the PDSC system's (Case B-2-2) latent heat loads increased with considered moisture effects due to its potent dehumidification effect, which resulted in low indoor humidity levels, prompting hygroscopic materials to release moisture into the room. The PSE system's latent heat loads were reduced by 385.5 kWh (70.8%) compared to the exhaust system.

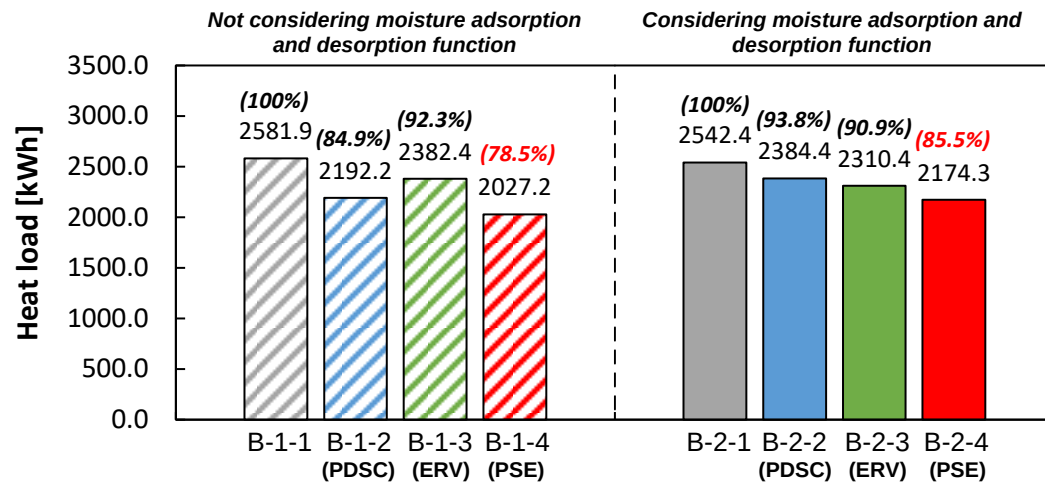
Overall, from exhaust ventilation to PSE systems, despite an increase in sensible heat load, the substantial reduction in latent heat load was more impactful. The dehumidification effects of the PSE system throughout the summer led to a total heat load reduction of 554.7 and 368.1 kWh (approximately 21.5% and 14.5%), under varying building envelope characteristics, compared to the exhaust ventilation system, as illustrated in Figure 4.15(c).



(a) Sensible heat load



(b) Latent heat load



(c) Total heat load

Figure 4.13 Heat loads of whole building for Cases B-1-1~B-2-4 (July 1 to September 30)

4.2 Simulations for evaluating PSE performance by using heat-pump characteristic model coupled with THERB for HAM

Section 4.1 estimated the PSE system's cooling load reduction effect using the THERB simulation software, which employs an ideal air-conditioning model (i.e., sets constant room temperature and relative humidity). However, this simplified air-conditioning model used in THERB for HAM has two limitations:

- 1) It does not account for the amount of dehumidification (i.e., condensate content) by the evaporator of the indoor unit in a heat-pump during cooling, nor does it calculate the electricity consumed by the heat-pump air-conditioning. This could result in an inaccurate estimation of a PSE system's dehumidification and energy savings effects under air-conditioning conditions.
- 2) Given the recent years' low dehumidification performance of air conditioners due to high refrigerant evaporation temperatures during the cooling season, which fails to satisfy indoor hygrothermal comfort, there is a need to use a detailed air-conditioning model to study the dehumidification performance of PSE systems at high refrigerant evaporation temperatures.

Therefore, this section aims to conduct a comprehensive simulation analysis of the PSE system's performance evaluation using a new detailed simulation methodology. This approach predicts the dehumidification effect of a PSE system during the cooling season. The detailed coupled simulation approach can simulate the condensation dehumidification of vapor-compression-type residential heat pumps, the energy consumption of air conditioners, and the refrigerant evaporation temperature, essential for assessing the PSE system's effectiveness under real-life conditions.

4.2.1 THERB for HAM coupled with heat-pump characteristic model

The ideal heat load model in THERB for HAM is constrained because it calculates heat load by setting constant temperature and humidity [3]. To address this, we have integrated the heat-pump characteristic model developed by the Japanese Central Research Institute of Electric Power into THERB for HAM for detailed simulation [21]. This heat-pump characteristic model is designed to calculate the ratio between theoretical and actual efficiencies of heat pumps. It utilizes manufacturer's specifications (minimum, rated, and maximum electricity consumption; minimum, rated, and maximum cooling capacities; indoor and outdoor unit airflow rates; and bypass factor) to estimate the power consumption and coefficient of performance (COP) of air conditioners. Additionally, this model can accurately predict the amount of water dehumidified by evaporator condensation, allowing for a detailed calculation of the latent cooling load. The process that couples THERB for HAM software with the heat-pump characteristic model is illustrated in Figure 4.14.

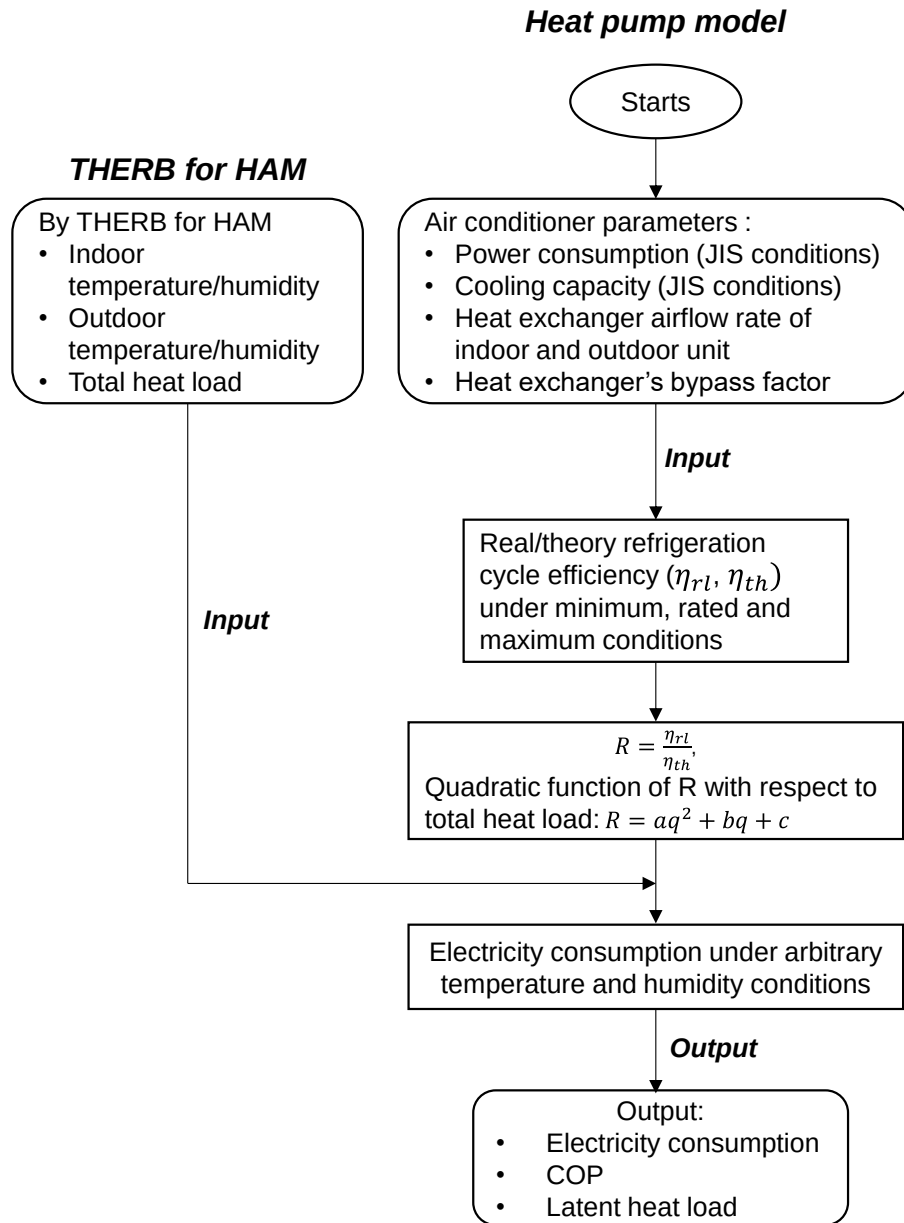


Figure 4.14 The flow chart of calculation that coupled THER for HAM software and heat-pump characteristic model

Our experimental house uses a commonly found residential air conditioner, an air-source heat-pump. Figure 4.15 provides a conceptual overview of a typical refrigeration cycle of a residential air-source heat-pump during the cooling season, where the evaporator cools indoor air and the condenser expels heat from indoors to outdoors. The cycle involves the refrigerant repeatedly compressing, exchanging heat, and expanding to transfer heat from indoors to outdoors during summer. In contrast, during winter, the refrigerant flow reverses to bring outdoor heat inside.

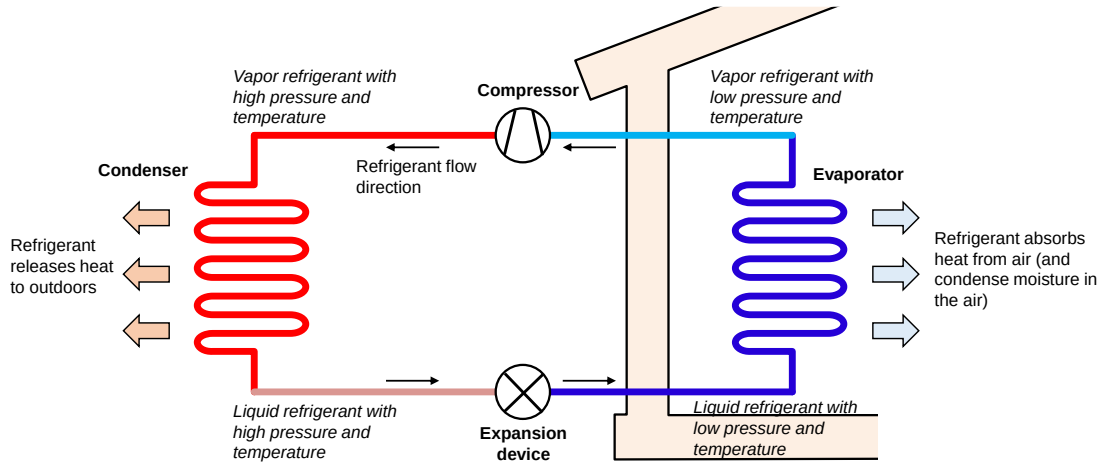


Figure 4.15 Diagram of typical refrigeration cycle of residential air-source heat-pump during cooling season

The electricity consumption for air-conditioning, P (W), is calculated as the sum of the electricity used by the refrigeration, P_r (W), and other auxiliary machines, P_{au} (W), such as fans.

$$P = P_r + P_{au} \quad (4.4)$$

The COP of an air conditioner is defined as the ratio between the required heat loads, q (W), for managing indoor temperature and humidity, and the total power input, P .

$$COP = \frac{q}{P} \quad (4.5)$$

COP is an index of overall air conditioner efficiency, while the actual efficiency of the refrigeration cycle, η_{rl} (-), is expressed as follows:

$$\eta_{rl} = \frac{q}{P_r} \quad (4.6)$$

Thus, from Eqs. (4.4) to (4.6), the real refrigeration cycle efficiency is derived as follows:

$$\eta_{rl} = \frac{COP \cdot q}{q - COP \cdot P_{au}} \quad (4.7)$$

The theoretical efficiency of a refrigeration cycle, or the inverse Carnot cycle efficiency, is based on:

$$\eta_{th} = \frac{T_{evp}}{T_{cnd} - T_{evp}} \quad (4.8)$$

where T_{evp} (°C) represent the refrigerant evaporating temperature in the evaporator and T_{cnd} (°C)

denotes the refrigerant condensing temperature in the condenser. R (-) denotes the ratio between the real and theoretical efficiencies of the refrigeration cycle.

$$R = \frac{\eta_{rl}}{\eta_{th}} \quad (4.9)$$

then, by organizing the Eqs. (4.7), (4.8), and (4.9), we obtain:

$$\frac{R \cdot T_{evp}}{T_{cnd} - T_{evp}} = \frac{COP \cdot q}{q - COP \cdot P_{au}} \quad (4.10)$$

Therefore, if the COP and q for specific operating conditions are known, and if the refrigerant evaporating temperature, T_{evp} and condensing temperature, T_{cnd} can be estimated. Eq. (4.10) provides a formula linking the unknowns R and P_{au} . Using this model, R and P_{au} can be obtained for various conditions (minimum, rated, and maximum) by applying Eq. (4.10). and following steps to ascertain the refrigerant's evaporation and condensation temperatures.

- *Refrigerant evaporation temperature:*

As depicted in Figure 4.17, during the refrigeration cycle, the refrigerant transitions from liquid to vapor in the indoor unit through evaporation. It is assumed that the evaporation temperature of the refrigerant in the evaporator matches that in the indoor heat exchanger. If the indoor heat exchanger's surface does not experience condensation, meaning all heat exchange between the air and the surface is sensible, with no change in air's water content, then the cooling load q is calculated using Eq. (9):

$$q = M_{evp} \cdot c_a \cdot (T_i - T_{evp}) \quad (4.11)$$

$$M_{evp} = (1 - BF) \cdot M_i \quad (4.12)$$

where M_i (kg/s) represents the mass flow rate of air through the indoor heat exchanger, and BF (-) (the bypass factor) is the proportion of airflow bypassing the heat exchanger surface unaffected by heat transfer. M_{evp} (kg/s) signifies the mass flow rate of air in contact with the heat exchanger. Consequently, the refrigerant temperature, T_{evp} (°C), in the evaporator can be delineated as follows:

$$T_{evp} = T_i - \frac{q}{(1 - BF) \cdot M_i \cdot C_a} \quad (4.13)$$

where C_a denotes the specific heat (kJ/kg (DA) °C) of humid air. The supply air temperature (T_{sa}) and absolute humidity (x_{sa}) when no condensation occurs can be expressed as follows:

$$T_{sa} = BF \cdot T_i + (1 - BF)T_{evp} \quad (4.14)$$

$$x_{sa} = x_i \quad (4.15)$$

In the intake air's absolute humidity, x_i , exceeds the saturation temperature determined using Eq. (7), excess moisture condenses, absorbing heat and resulting in higher supply air temperature and

lower absolute humidity. Here, the temperature and absolute humidity of the outlet air of the evaporator can be calculated as follows:

$$T_{sa} = BF \cdot T_i + (1 - BF)T_{evp,con} \quad (4.16)$$

$$x_{sa} = BF \cdot x_i + (1 - BF)x_{evp,con} \quad (4.17)$$

where $T_{evp,con}$ and $x_{evp,con}$ are the temperature and AH of air condensed in the evaporator, respectively. The dehumidification capacity of the heat-pump derives from the absolute humidity difference between inlet and outlet air at the evaporator. Thus, the air conditioner's latent heat load is expressed in Eq. (16):

$$q_l = l_w \cdot \gamma_a \cdot Q_{evp}(x_i - x_{sa}) \quad (4.18)$$

$$q = q_l + q_s \quad (4.19)$$

where l_w (kJ/kg) and γ_a (kg/m³) represent the latent heat of evaporation and specific weight of water vapor, respectively. Q_{evp} (m³/s) is the airflow rate through the indoor unit of the heat-pump. Summing the latent q_l , and sensible q_s heat loads from earlier moments gives the subsequent cooling demand, which is substituted into Eq. (9).

- *Refrigerant condensation temperature:*

During condensation in the outdoor unit, the refrigerant changes from vapor to liquid. Since outdoor unit heating does not lower air temperature below the dew point, condensation temperature is calculated similarly to the no-condensation scenario for the indoor unit. The heat released by the heat-pump's high-temperature heat source is expressed as follows:

$$q + P = M_{cnd} \cdot c_a \cdot (T_{cnd} - T_o) \quad (4.20)$$

$$M_{cnd} = (1 - BF) \cdot M_o \quad (4.21)$$

where M_o (kg/s) is the mass flow rate of air through the outdoor heat exchanger, enabling determination of temperature in the condenser using Eq. (4.22).

$$T_{cnd} = T_o + \frac{q + P}{(1 - BF) \cdot M_o \cdot c_a} \quad (4.22)$$

Airflow rates M_i and M_o for indoor and outdoor units are adjusted by the air conditioner manufacturer's airflow rate multiplied by humid air density. The bypass factor BF, reflecting factors such as airflow rate, is typically between 0.1-0.2 [21].

Air conditioner specifications, including cooling capacity and power consumption under minimum, rated, and maximum conditions, as shown in Table 4.6. These values were measured following the methods specified in the Japanese Industrial Standards (JIS C9612, JIS B8615-1). These specs, measured under specified temperature and humidity conditions listed in Table 4.7, enable determination of R , and P_c through a specific procedure:

Chapter 4 Simulation analysis by using numerical simulation under several conditions

- 1) Determine R and P_c for minimum and rated conditions using maximum and rated condition values.
- 2) Assume P_c from Step 1) is consistent at maximum conditions to determine R for these conditions.
- 3) Use a quadratic equation to approximate the relationship between cooling capacity (kW) and R across minimum, rated, and maximum conditions, making R a function of cooling capacity, with P_c considered a constant independent of cooling capacity.

This method allows calculation of the refrigerant evaporating and condensing temperatures, theoretical and actual efficiencies for various conditions, and estimation of COP and power consumption using specific parameters R (related to refrigeration capacity) and P_c (a constant value).

Table 4.6 Detailed information and configurations of heat-pump used in PSE residence

Items	Contents
Type	Air-source heat pumps both for cooling and heating
Dimension	Indoor unit: 340×1096×940, outdoor unit: 675×825×300
Power consumption	0.255, 1.090, and 1.640 kW (minimum, rated, and maximum)
Cooling capacity	1.0, 4.0, and 5.0 kW (minimum, rated, and maximum)
Heat exchanger airflow rate	Indoor unit: 400~900 m ³ /h Outdoor unit: 1818 m ³ /h
Heat exchanger's bypass factor	0.2
Rate coefficient of performance in cooling season	3.67

Table 4.7 JIS specification temperature and humidity conditions under cooling capacity test condition

Items	Contents
Indoor dry bulb temperature (°C)	27
Indoor wet bulb temperature (°C)	19
Outdoor dry bulb temperature (°C)	35
Outdoor wet bulb temperature (°C)	24

4.2.2 Model validations

4.2.2.1 Validation conditions

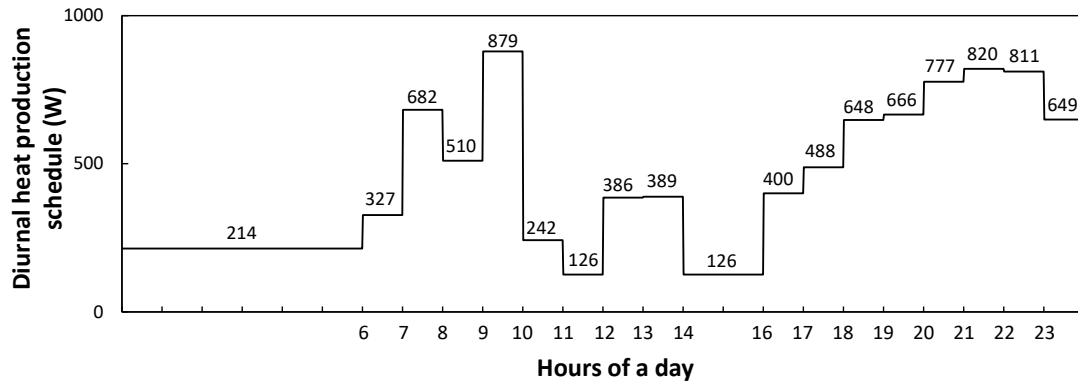
The PSE system model validation conditions under air-conditioning conditions, using THERB for HAM coupled with a heat-pump characteristic model, are listed in Table 4.8.

Chapter 4 Simulation analysis by using numerical simulation under several conditions

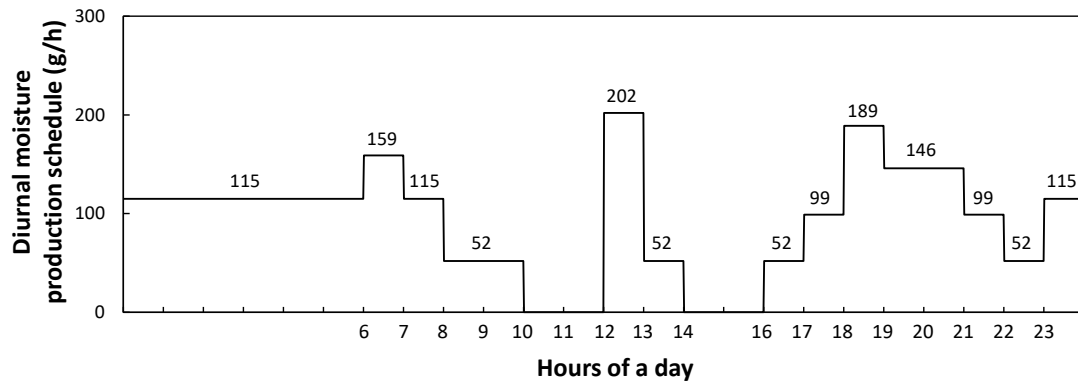
Throughout the measurement period, the PSE system operated continuously, with the air conditioner set to a constant temperature of 27 °C, as recommended by the Building Sanitation Law of Japan [16]. The ERV system provided ventilation at approximately 0.5 air changes per hour (ACH) (170 m³/h). The indoor space accommodated a couple, generating sensible heat from human metabolic activity, equipment, and lighting. Latent heat originated from human respiration, transpiration, cooking, bathing, and other daily activities. The daily internal gain schedule was based on the occupants' routine and followed the standard energy consumption calculation regulation proposed in Japan [19], as depicted in Figure 4.16.

Table 4.8 PSE system model validation conditions under air-conditioning conditions by using THERB for HAM coupled with heat-pump characteristic model

Items	Contents
Operated system	The PSE system
Measurement period	2023.7.30-8.6
Weather data	Measured meteorological data from Japan Meteorological Agency (July 30 to August 6, 2023) [12]
Indoor condition	Air-conditioning 27 °C
Internal moisture and heat gain	With internal moisture and heat generation by couple, refer to Figure 4.16
Ventilation rate	Approximately 170 m ³ /h
Heat exchange efficiency	Sensible heat exchange efficiency: 0.40 during the daytime, 0.80 at nighttime. Latent heat exchange efficiency: 0.50 during the daytime, 0.58 at nighttime.
Velocity of airflow through walls' cavity	Calculated non-stationarily by THERB for HAM.
Velocity of airflow through roofs' cavity	In the PDSC roof: daytime = 0.05 m/s and nighttime = 0.10 m/s. In the other roofs: calculated non-stationarily by THERB for HAM.



(a) Diurnal heat production schedule of entire house



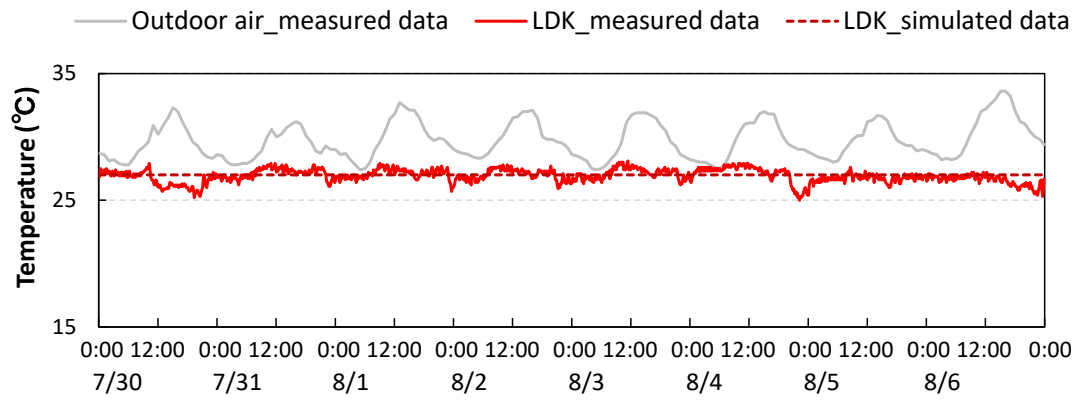
(b) Diurnal moisture production schedule of entire house

Figure 4.16 Diurnal heat and moisture production schedule of entire house for model validation—two-person family

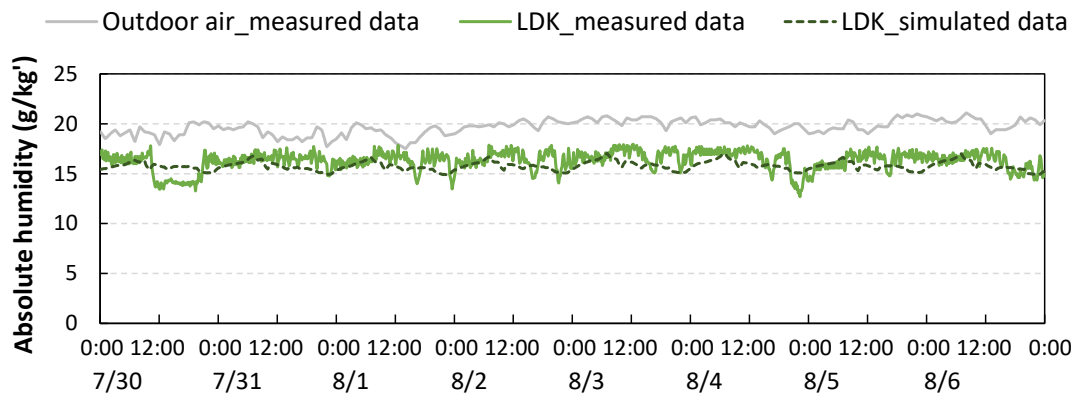
4.2.2.2 Validation results

A detailed PSE model, established using the coupling method, was predicated on consistent daily heat and moisture production by indoor occupants, as illustrated in Figure 4.16. It assumed that the hygrothermal properties of each building envelope material remained unchanged by the surrounding conditions, disregarding the hysteresis effect of porous hygroscopic materials.

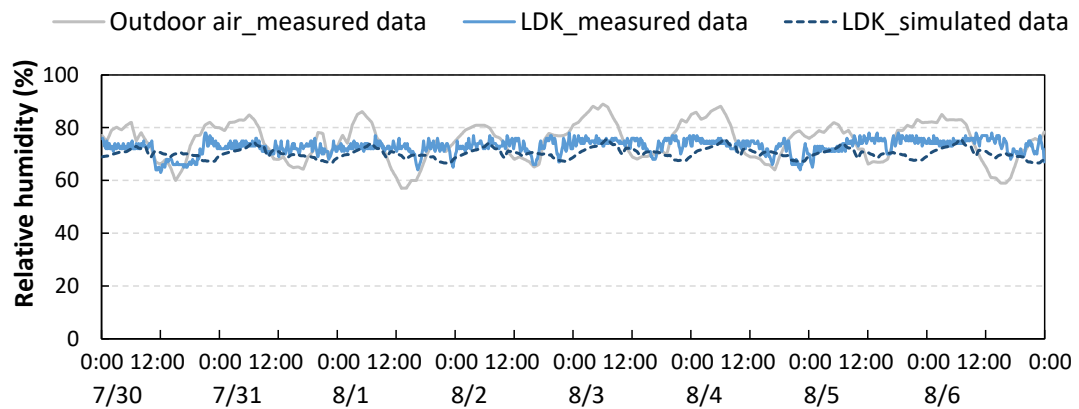
The accuracy of this model was evaluated by comparing predicted and actual data for temperature, absolute humidity, and relative humidity in the indoor area (LDK) and the roof ventilation layer, as shown in Figures 4.17 and 4.18. The MAPE between predicted and measured values is detailed in Table 4.9. With mean MAPE values of 3.84% for the LDK and 8.75% for the roof ventilation layer, the detailed PSE model is validated as highly accurate according to the Lewis standard [15].



(a) Temperature

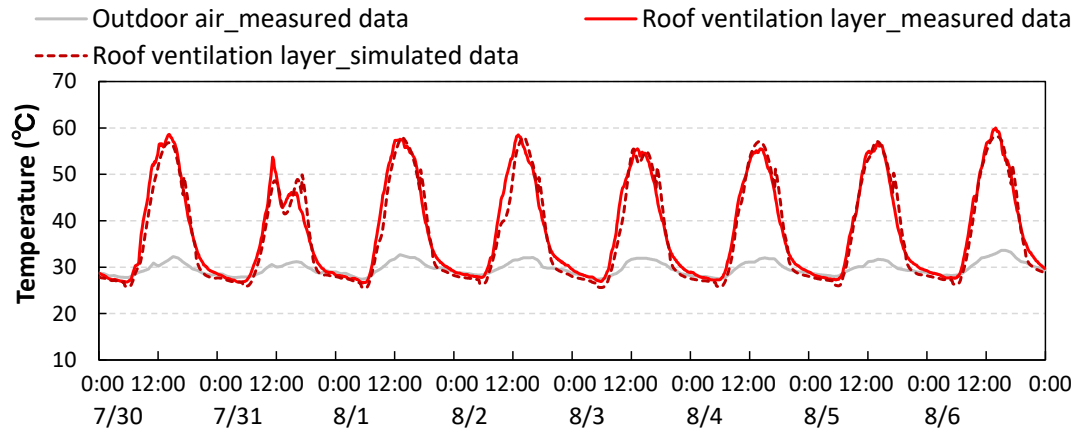


(b) Absolute humidity

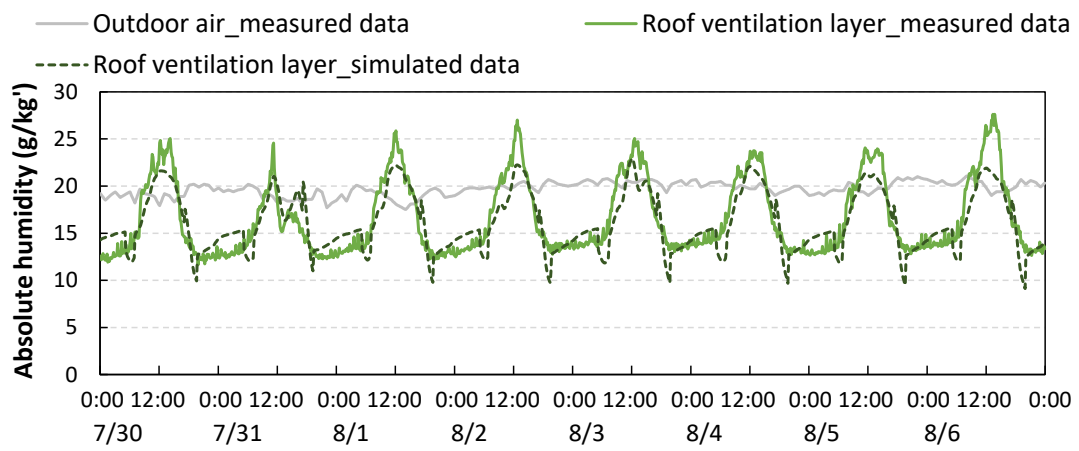


(c) Relative humidity

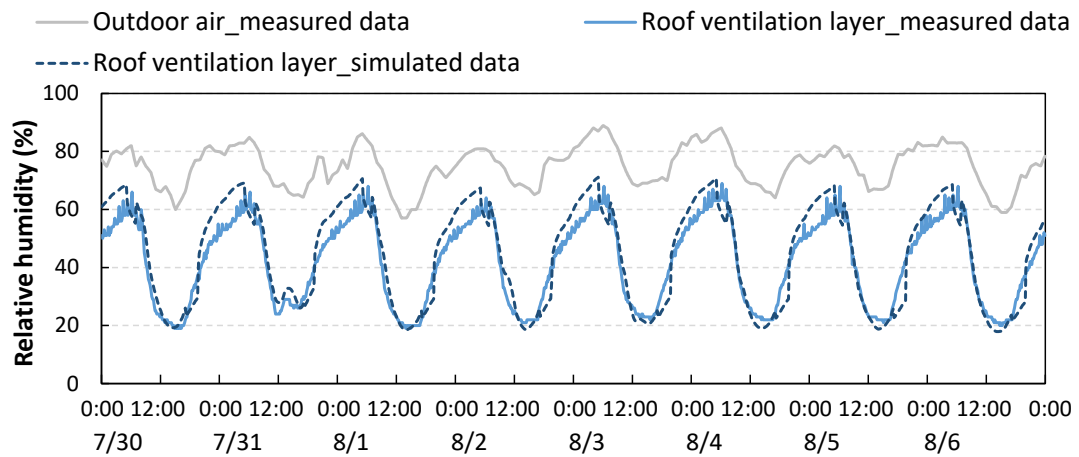
Figure 4.17 Comparisons between measured values and simulated results for LDK's temperature, absolute humidity, and relative humidity of PSE system



(a) Temperature



(b) Absolute humidity



(c) Relative humidity

Figure 4.18 Comparisons between measured values and simulated results for roof ventilation layer's temperature, absolute humidity, and relative humidity of PSE system

Table 4.9 Prediction accuracy of PSE model

	MAPE of temperature (%)	MAPE of absolute humidity (%)	MAPE of relative humidity (%)	Average (%)
LDK	1.56	5.57	4.39	3.84
Roof ventilation layer	4.69	9.85	11.71	8.75

4.2.3 Case C: PSE system performance evaluation compared with other systems

4.2.3.1 Simulation conditions

This subsection delves into the indoor temperature and humidity variations, heat loads, and air conditioner electricity consumption of the PSE system, comparing it with other ventilation systems (exhaust ventilation, PDSC, and ERV systems) across different building envelopes. The simulation scenarios, described in Table 4.10, were split into two groups based on the building models' envelope consideration for moisture adsorption and desorption: Cases C-1-1~C-1-4, which do not consider it, and Cases C-2-1~C-2-4, which do. Each series includes four distinct ventilation systems: exhaust ventilation, PDSC, ERV, and the PSE systems.

Table 4.10 Simulation cases

Cases	Consider the moisture adsorption and desorption function of the building envelope	Air conditioner	The operation of independent PDSC system	The operation of independent ERV system	The operation of PSE (PDSC combined with ERV) system
Case C-1-1	No	AC-1	-	-	-
Case C-1-2			Operation		
Case C-1-3			-	Operation	
Case C-1-4				-	Operation
Case C-2-1	Consider		-	-	-
Case C-2-2			Operation		
Case C-2-3			-	Operation	
Case C-2-4*				-	Operation

* Note: Model specifications used to validation in Section 4.2.2

Chapter 4 Simulation analysis by using numerical simulation under several conditions

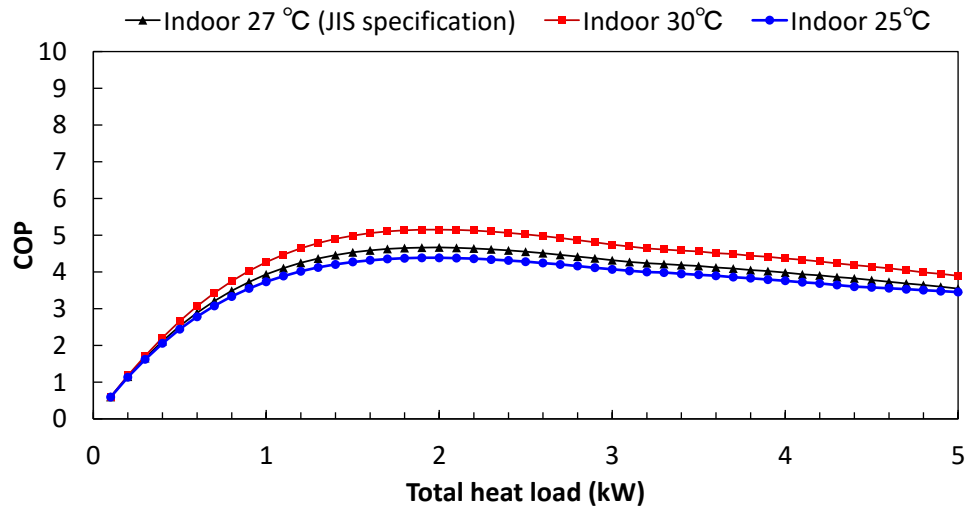
The simulation conditions for all Cases C are detailed in Table 4.11. While certain parameters such as the simulation duration, interval, weather data, building envelope materials, ventilation rate, and ERV's heat exchange effectiveness remain consistent with Cases A and B, Case C employs a detailed calculation method incorporating the heat-pump characteristic model, thereby setting the air conditioner to maintain a temperature of 27 °C. As a result, indoor humidity is influenced by a mix of human-generated moisture, moisture adsorption and desorption from building materials, and ventilated air. Notably, the air-conditioning parameters in the real PSE experimental house (AC-1) are meticulously defined for use as simulation inputs. Contrary to the model validation scenario in Subsection 4.2.2.1, which was based on a two-person household, Case C adopts the heat and moisture generation schedule recommended for a standard four-person family by the Japanese standard energy consumption calculation regulation, consistent with the approach used in simulations for Cases A and B.

Table 4.11 Simulation conditions for Cases C-1-1~C-2-4

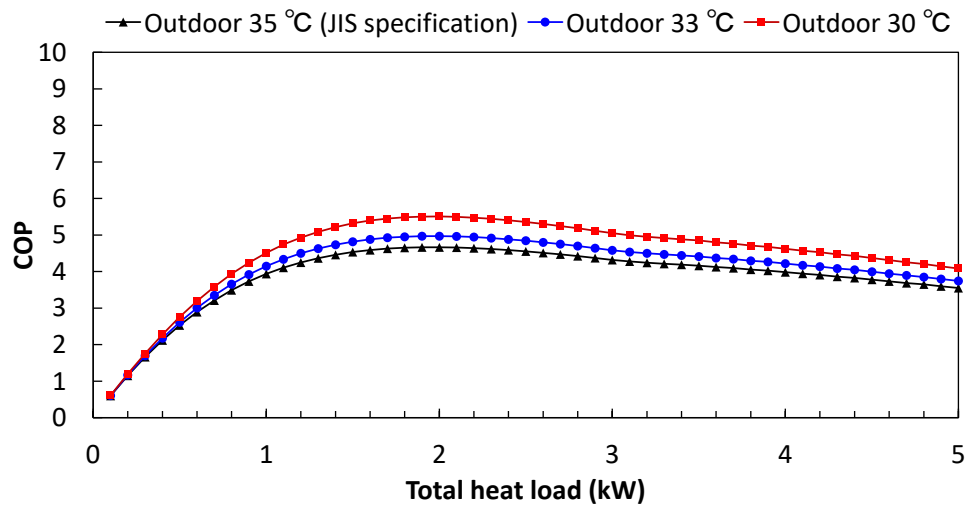
Items	Contents
Simulation cases	Refer to Table 4.6
Simulation duration	June 1 to September 30 (trial calculation from March 1 to May 31)
Simulation interval	1 min
Weather data	Fukuoka (standard year, 2001–2010) Expanded AMeDAS Weather Data in Japan [12]
Building envelope materials	Refer to Table 3.3
Air-conditioning	Constant temperature = 27 °C - <i>Specifications of AC-1 (the AC used in experiment house):</i> Power consumption (minimum, rated, and maximum): 0.255, 1.090, and 1.640 kW. Cooling capacity (minimum, rated, and maximum): 1.0, 4.0, and 5.0 kW. Indoor heat exchanger's airflow rate: 400~900 m ³ /h Outdoor heat exchanger's airflow rate: 1818 m ³ /h Indoor heat exchanger's bypass factor: 0.2
Ventilation rate	Constant ventilation rate: 0.5 ACH
ERV's heat exchange effectiveness	In ERV building model: 0.8 (sensible) and 0.58 (latent) In PSE building model: 0.4 (sensible) and 0.5 (latent) during the daytime, 0.8 (sensible) and 0.58 (latent) at nighttime.
Internal moisture and heat gain	Refer to Figure 4.6

Prior to the simulation, a basic performance analysis of the air conditioner (AC-1) used in the simulation is essential. Figure 4.19 illustrates the relationship between cooling loads (up to a maximum of 5.0 kW, as referenced in Table 4.11) and the COP for AC-1, utilizing indoor and outdoor temperature parameters and following JIS specifications as listed in Table 4.8.

In Figure 4.19(a), with indoor humidity and outdoor temperature and humidity aligned with JIS specifications, the COP of AC-1 is determined by varying the indoor temperature under different cooling loads. It is observed that the COP increases with higher indoor temperatures under a fixed cooling load, attributable to the rise in the refrigerant evaporation temperature (Eq. (4.11)). Similarly, Figure 4.19(b) calculates the COP of AC-1 by varying the outdoor temperature under different cooling loads, demonstrating that a lower outdoor temperature under the same cooling load results in a higher COP. This effect is explained by Eq. (4.20), where a lower outdoor temperature reduces the refrigerant's evaporating temperature, thereby increasing the COP [21]. The COP of AC-1 peaks at approximately 4.7 with an indoor cooling load of approximately 2.0 kW under JIS specifications.



(a) Indoor temperature as parameter



(b) Outdoor temperature as parameter

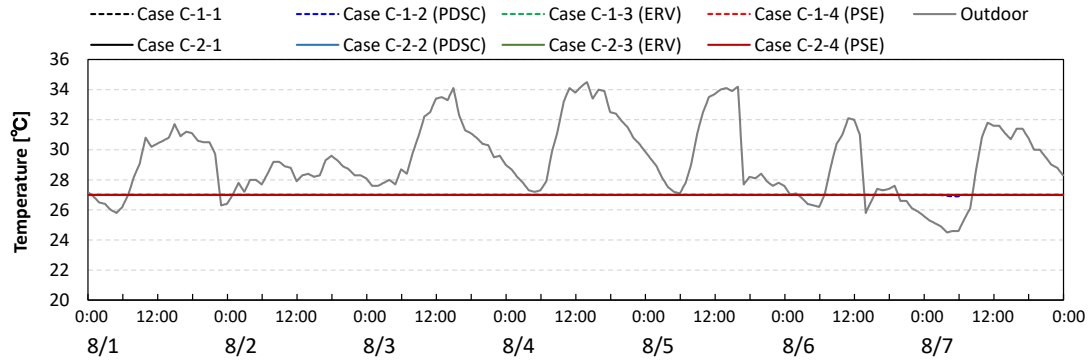
Figure 4.19 Relationship between total heat load and COP for AC 1

4.2.3.2 Simulation results

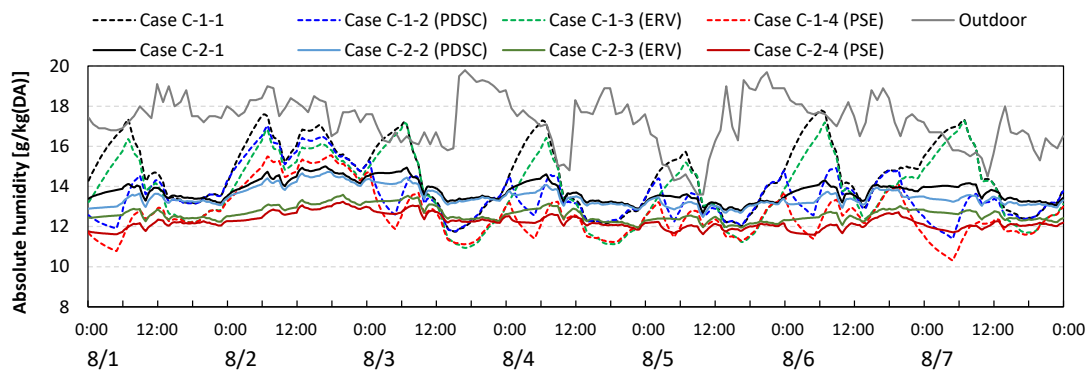
Figure 4.20 depicts the temporal changes in temperature, absolute humidity, and relative humidity for Cases C-1-1~C-2-4 from 1 to 7 August (standard year, 2001–2010), with air-conditioning set at 27 °C using the coupled detailed simulation method.

Despite the indoor temperatures being constant across simulations, as dictated by THERB's constant temperature-setting algorithm, indoor absolute humidity outcomes, as seen in Figure 4.20(b), vary significantly. The indoor humidity for all cases exhibits natural variations due to the combined effects of the air conditioner's moisture condensation, the building envelope's moisture adsorption and desorption, ventilation, and indoor moisture generation. When the building envelope does not incorporate moisture adsorption or desorption, indoor humidity varies widely. Notably, Case B-1-2 (PDSC system) shows slightly lower daytime indoor absolute humidity than Case B-1-1 (exhaust ventilation system) and substantially lower at night due to the PDSC system's dehumidification effect. Especially at night, the roof moisture adsorption effect of the PDSC system causes the absolute humidity in the room to drop significantly. Case B-1-4 (PSE system) achieves the lowest absolute indoor humidity and the most effective dehumidification by integrating the PDSC and ERV systems. When considering moisture adsorption and desorption effects in building materials, Cases B-2-1~B-2-4 show relatively stable indoor absolute humidity variations, with Case B-2-4 achieving the most effective dehumidification.

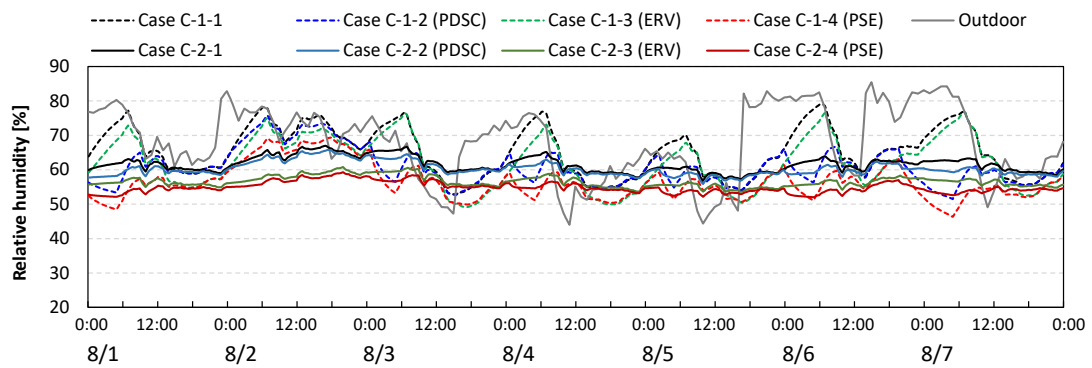
The relative humidity trends in Figure 4.20(c) mirror those of the absolute humidity, showcasing the PSE system's lower relative humidity compared to other systems, irrespective of the building envelope's consideration for moisture adsorption and desorption effects.



(a) Temperature



(b) Absolute humidity



(c) Relative humidity

Figure 4.20 Temporal changes in temperature and humidity in LDK for Cases C-1-1~C-2-4—with air-conditioning (August 1 to 7)

Figure 4.21 illustrates the frequency distribution (0.1 g/kg (DA) intervals), cumulative rate, and boxplot of absolute humidity in the LDK for Cases C-1-1~C-2-4 from July 1 to September 30 (standard year 2001–2010), during which the air-conditioning was set at 27 °C using the detailed simulation method. This method revealed that the indoor absolute humidity during summer was lowest in buildings equipped with the PSE system, regardless of building envelope characteristics. Additionally, the presence or absence of the moisture effect in the building envelope had minimal impact on the dehumidification effectiveness of the PSE system, with the mean absolute humidity difference between indoor and outdoor air being 3.6 g/kg (DA) for both Cases C-1-4 and C-2-4. However, the dehumidification effectiveness of the PDSC system was reduced in the presence of hygroscopic building envelope characteristics (Case C-2-2) compared to its absence (Case C-1-2), as previously explained: the PDSC system's dehumidification effect is so effective that it leads to moisture being released into the room from the building materials in Case C-2-2.

Figure 4.22 displays the frequency distribution (1% intervals), cumulative rate, and boxplot of relative humidity in the LDK for Cases C-1-1~C-2-4 from July 1 to September 30 (standard year 2001–2010), with air-conditioning maintained at 27 °C using the detailed simulation method. The mean indoor relative humidity for Cases C-1-1 and C-2-1 are 64.9% and 59.8%, respectively, indicating that the dehumidification effect of the air conditioner alone was insufficient to achieve thermal comfort. The lowest mean indoor relative humidity, found in Cases C-1-4 and C-2-4 at 52.4%, which utilized the PSE system, was lower than the mean outdoor relative humidity by 19.5%. Although the mean indoor relative humidity is the same for different building envelopes within the PSE system, the relative humidity in Case C-1-4, without considering the moisture adsorption and desorption effect, fluctuates significantly, occasionally exceeding 65%, leading to reduced hygrothermal comfort. Therefore, a PSE system combined with a hygroscopic building envelope provides more effective dehumidification and a more comfortable indoor environment.

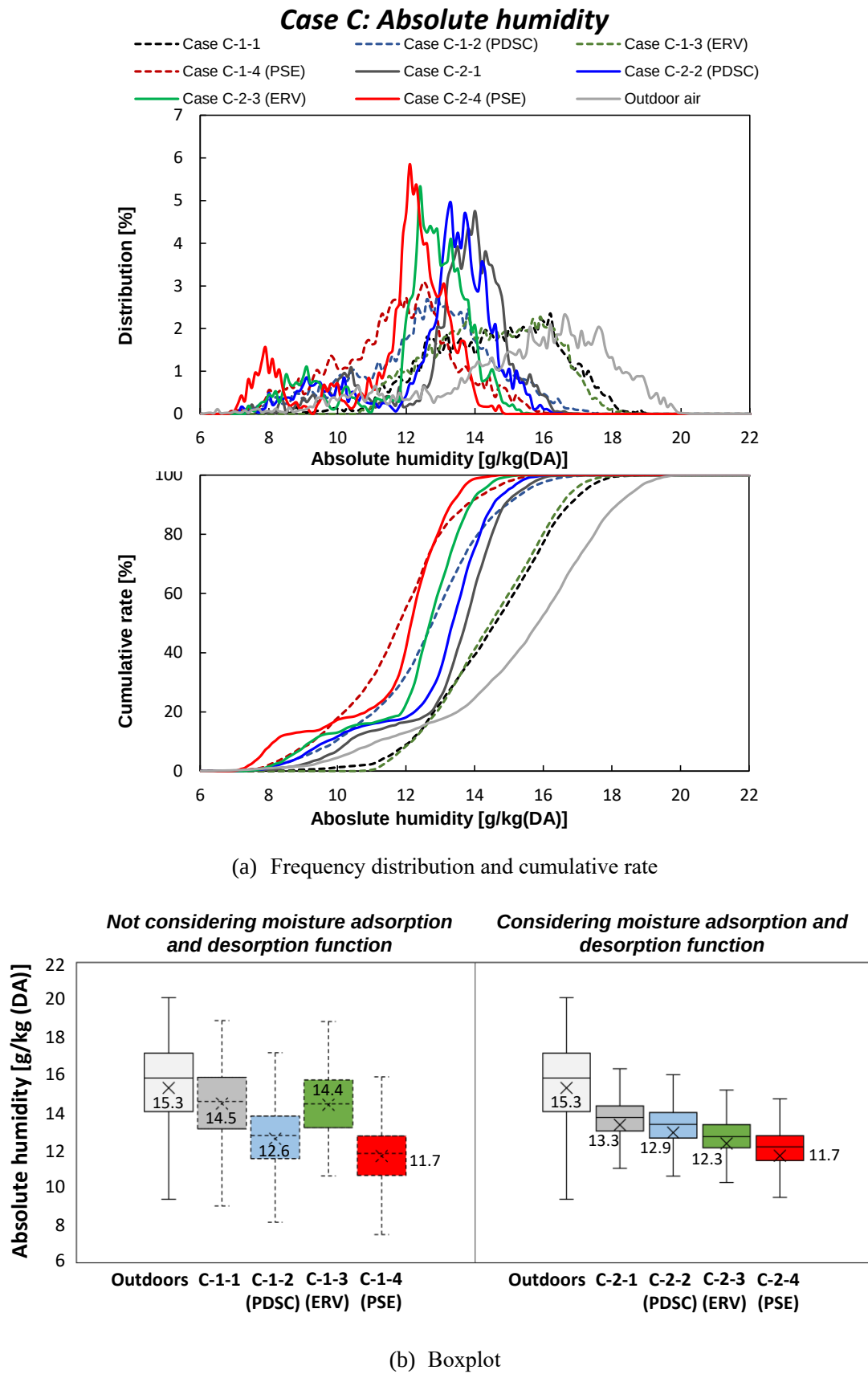


Figure 4.21 Frequency distribution, accumulation rate, and boxplot of absolute humidity for Cases C-1-1~C-2-4 (July 1 to September 30)

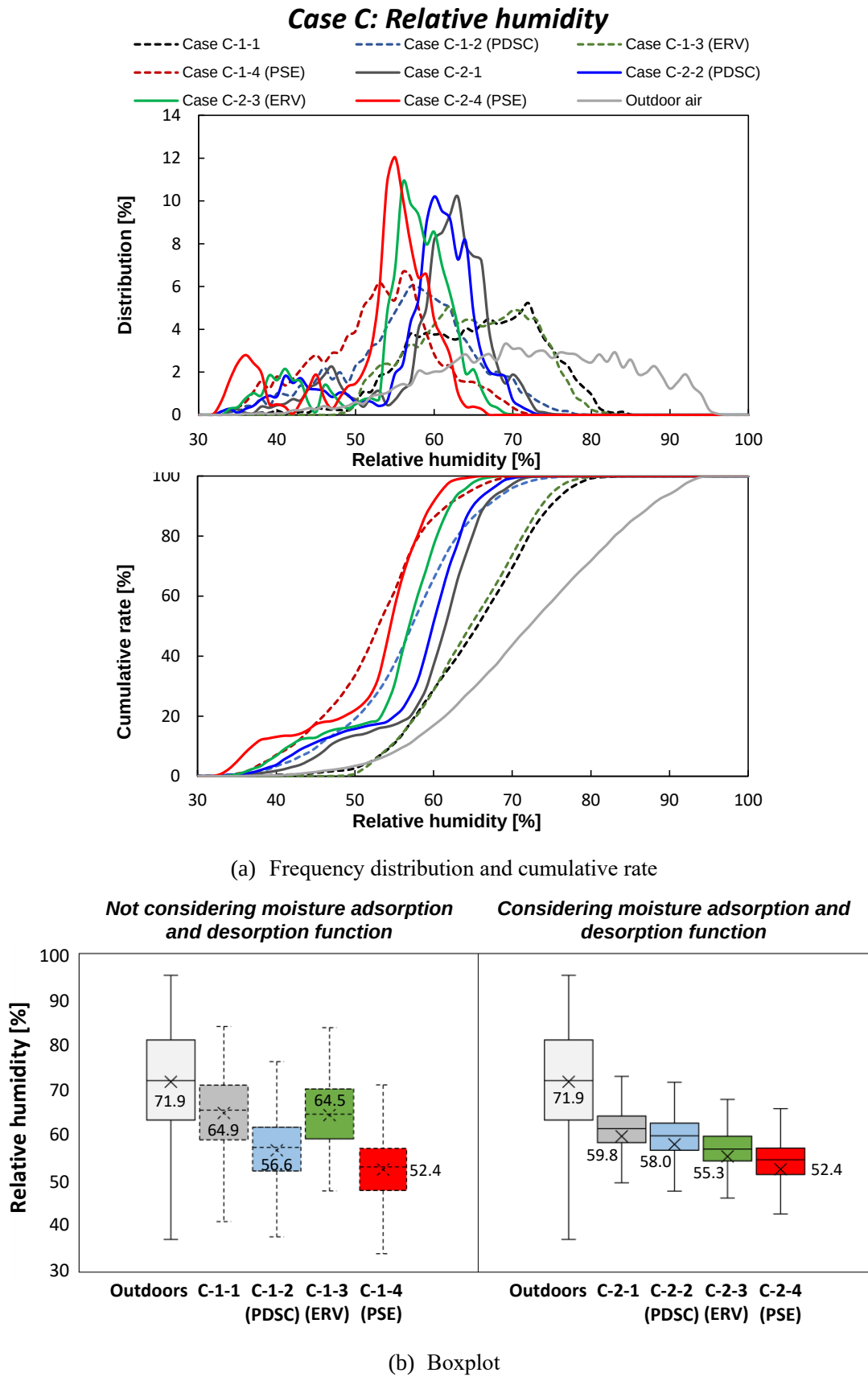


Figure 4.22 Frequency distribution, accumulation rate, and boxplot of relative humidity for Cases C-1-1~C-2-4 (July 1 to September 30)

Figures 4.23 and 4.24 present the refrigerant evaporation temperature and instantaneous latent heat load for Cases C-1-1~C-2-4, respectively. According to Figure 4.23, the refrigerant evaporates at lower temperatures during the day and higher temperatures at night, attributed to high solar radiation and cooling loads during the day, necessitating lower temperature operation to manage indoor temperature. The demand for cooling is lower at night, allowing for higher refrigerant evaporation temperatures.

Regarding latent heat load, as depicted in Figure 4.24, there are periodic changes in the daily curve, reflecting the schedule of indoor moisture generation. Latent heat load decreases to nearly zero at night and increases during the day across all cases, correlating with the refrigerant evaporation temperature of Figure 4.23. The lower daytime evaporation temperatures facilitate air conditioner evaporator condensation, while the higher nighttime temperatures reduce or eliminate condensed moisture. The PSE system demonstrates the lowest latent heat load, benefiting from its superior dehumidification effect, which reduces the moisture content in indoor air and consequently, the amount of moisture available for condensation.

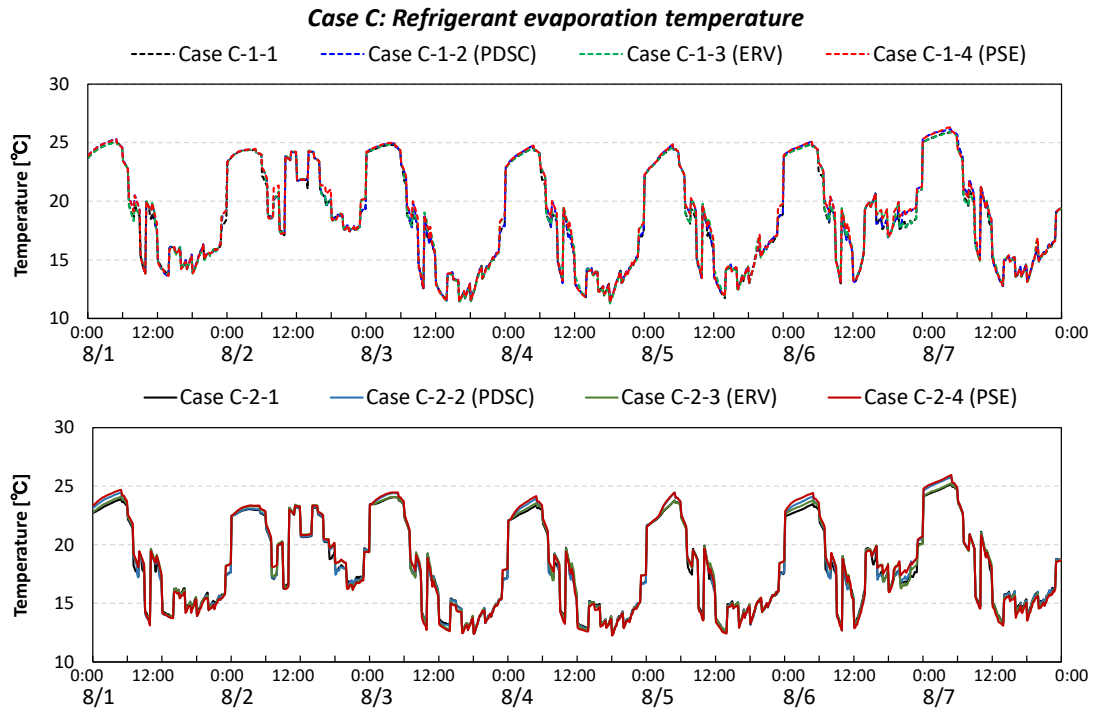


Figure 4.23 Refrigerant evaporation temperature for Cases C-1-1~C-2-4 (July 1 to September 30)

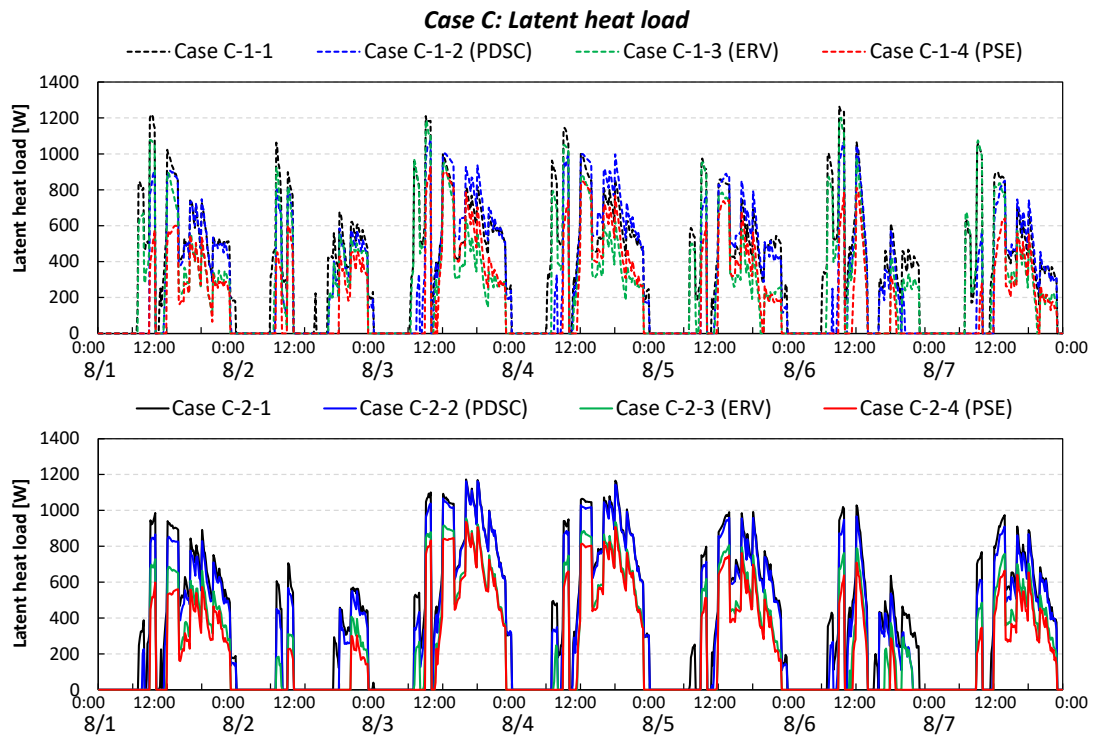
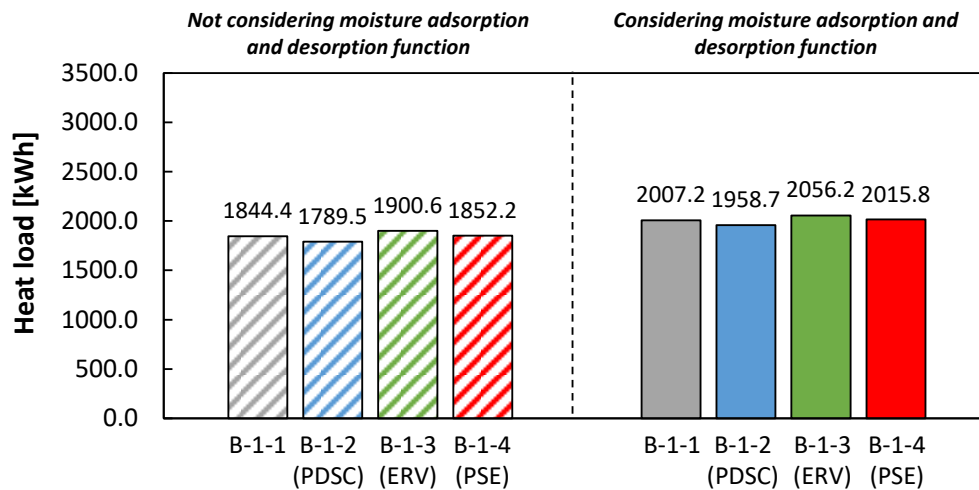


Figure 4.24 Instantaneous latent heat load for Cases C-1-1~C-2-4 (July 1 to September 30)

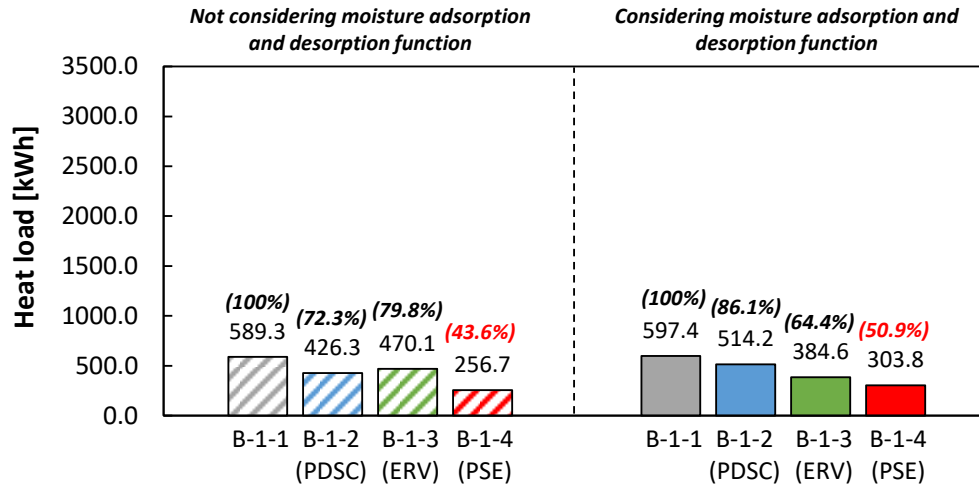
Figure 4.25 presents the simulated heat load and electricity consumption results for Cases C-1-1~C-2-4 from July 1 to September 30. In terms of sensible heat load (Figure 4.25(a)), consistent with Case B's findings, the ERV within the PSE system operates round-the-clock. Given that outdoor temperatures are cooler than indoor temperatures at night, the ERV's sensible heat recovery inadvertently warms the incoming fresh air, thereby raising the sensible heat load. Nonetheless, due to the PSE system's efficient passive dehumidification, it registers lower latent and overall heat loads compared to alternative systems. Specifically, the latent and total heat loads (Figure 4.25(b) and 4.25(c)) for the PSE system (Cases C-1-4) show reductions of 56.4% and 13.3%, respectively, against Case C-1-1. When accounting for the building envelope's moisture adsorption and desorption capabilities, the latent and overall heat loads in the PSE system (Cases C-2-4) see decreases of 49.1% and 10.9%, respectively, in comparison to Case C-2-1.

Regarding air-conditioning energy consumption (Figure 4.25(d)), the PSE system demonstrates modest reductions of 1.9% and 2.8%. This limited impact is attributed to the PSE system's role in diminishing the latent heat load, which accounts for a relatively minor portion of the total heat load. Consequently, this translates to a slight reduction in the total heat load and, by extension, minimal savings on energy consumption.

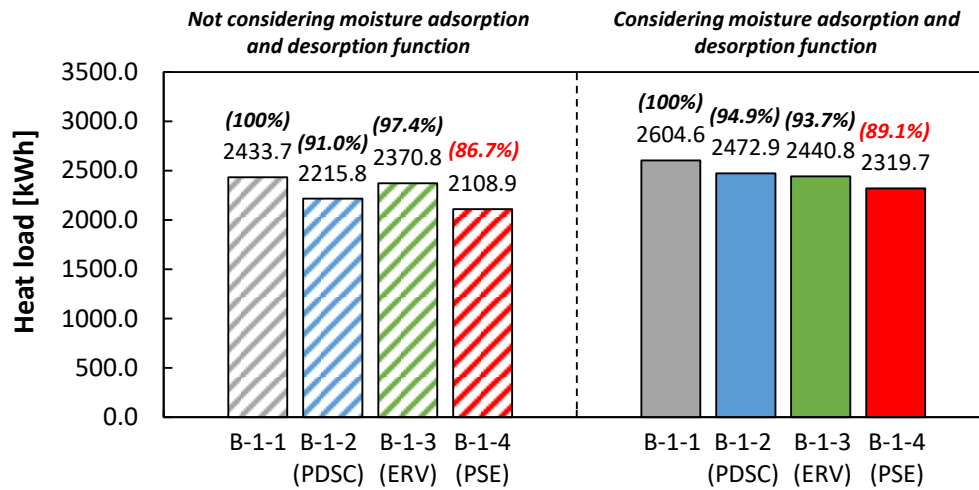
Despite the marginal influence of the PSE system on energy savings, its remarkable dehumidification capability effectively compensates for the reduced dehumidification efficiency of the air conditioner during condensation. Furthermore, when compared to other systems, the PSE system stands out for its ability to lower both the absolute and relative humidity levels indoors, thereby enhancing thermal comfort within the indoor environment without significantly increasing electricity consumption.



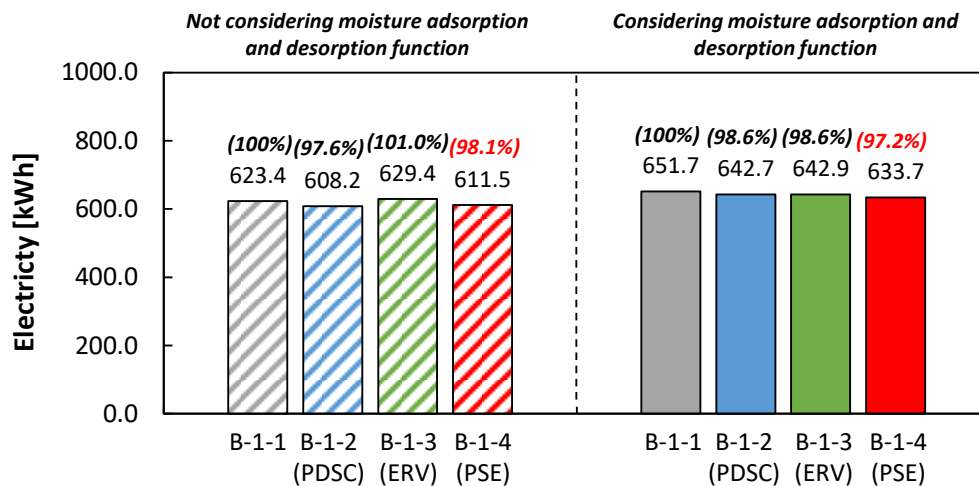
(a) Sensible heat load



(b) Latent heat load



(c) Total heat load



(d) Electricity consumption

Figure 4.25 Heat loads and electricity consumption of whole building for Cases C-1-1~C-2-4 (July 1 to September 30)

4.2.4 Case D: PSE system performance evaluation under various air conditioners' airflow rate

Condensation dehumidification by the air conditioner's evaporator is the principal method for removing moisture in residential buildings. However, there has been a trend toward increasing the refrigerant evaporation temperature in air conditioners to enhance the COP for energy-saving benefits, such as lowering power consumption by increasing the airflow rate from the indoor unit. Despite these efforts, a higher refrigerant evaporation temperature may diminish the air conditioner's dehumidification capability, potentially compromising the indoor hygrothermal environment during the hot and humid summer months. This subsection undertakes simulations to assess the PSE system's dehumidification performance across various air conditioner airflow rates.

4.2.4.1 Simulation conditions

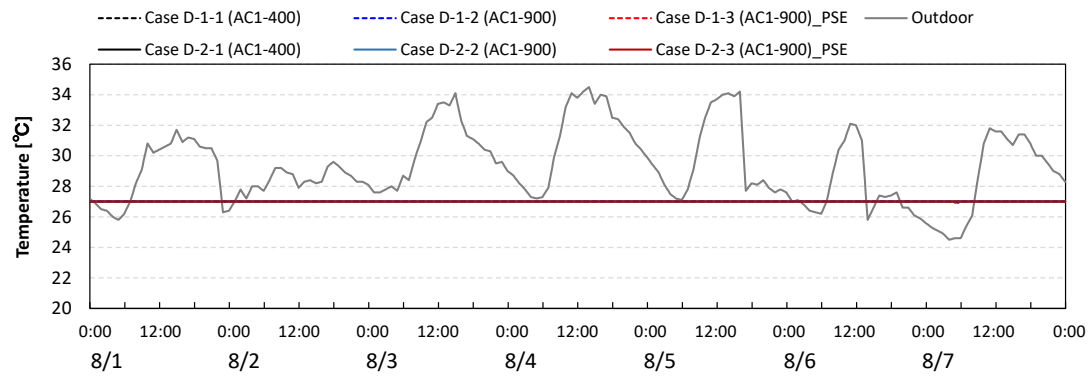
The simulation scenarios are detailed in Table 4.12. Cases D-1-1~D-1-3 do not account for the building envelope's moisture adsorption and desorption capabilities. Case D-1-1 involves an air conditioner, AC-1, operating at a low airflow rate (400 m³/h) combined with an exhaust ventilation system. Case D-1-2 parallels Case D-1-1, but with the air conditioner set to a high-airflow rate (900 m³/h). Case D-1-3 mirrors Case D-1-2, substituting the exhaust ventilation with the PSE system. Cases D-2-1~D-2-3 replicate the D-1 series, with the additions of the building envelope's moisture adsorption and desorption functions. The remaining simulation parameters for Cases D-1-1~D-2-3 align with those specified in Cases C, as referenced in Table 4.11.

Table 4.12 Simulation cases

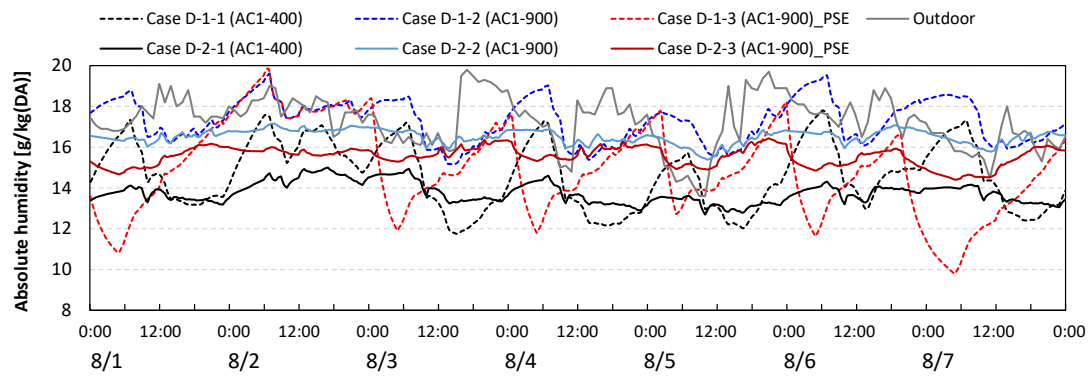
Cases	Consider the moisture adsorption and desorption function of the building envelope	Operation of PSE system	Air conditioner	The air flow rate of air conditioner (m ³ /h)
Case D-1-1	No	-	AC-1	400
Case D-1-2				900
Case D-1-3		Operation		900
Case D-2-1	Consider	-		400
Case D-2-2				900
Case D-2-3		Operation		900

4.2.4.2 Simulation results

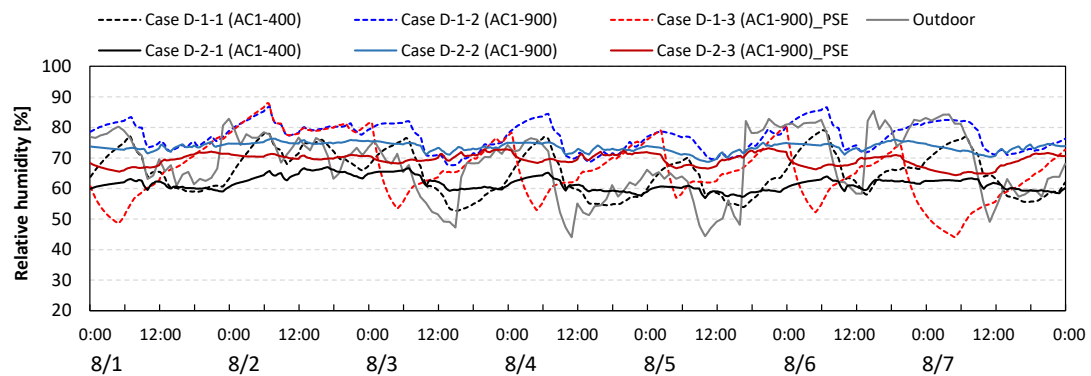
Figure 4.26 illustrates the temperature, absolute humidity, and relative humidity for Cases D-1-1~D-2-3 between August 1 and 7 (standard year, 2001–2010), with air-conditioning set to 27 degrees, employing the detailed coupled simulation methodology. As depicted in Figure 4.26(a), the air-conditioning maintains a steady room temperature of 27 degrees. The absolute humidity chart (Figure 4.26(b)) reveals that in Case D-1-2, which utilizes exhaust ventilation, indoor absolute humidity surpasses that of Case D-1-1 due to the increased air conditioner airflow rate in Case D-1-2. This higher airflow reduces the temperature differential between the indoor environment and the evaporator's refrigerant in Case D-1-2 (refer to Eq. (4.11)), raising the refrigerant evaporation temperature and diminishing condensate formation. Engaging the PSE system alongside a high-airflow air conditioner, Case D-1-3 demonstrates a reduction in indoor absolute humidity relative to D-1-2, attributed to the PSE system's passive dehumidification effect. Additionally, Case D-2-3 shows lower absolute humidity than Case D-2-2 when considering the building envelope's moisture adsorption and desorption properties. The relative humidity outcomes (Figure 4.26(c)) mirror the absolute humidity findings, indicating that combining high-airflow air-conditioning with the PSE system can further mitigate indoor relative humidity, thus enhancing indoor hygrothermal comfort.



(a) Temperature



(b) Absolute humidity

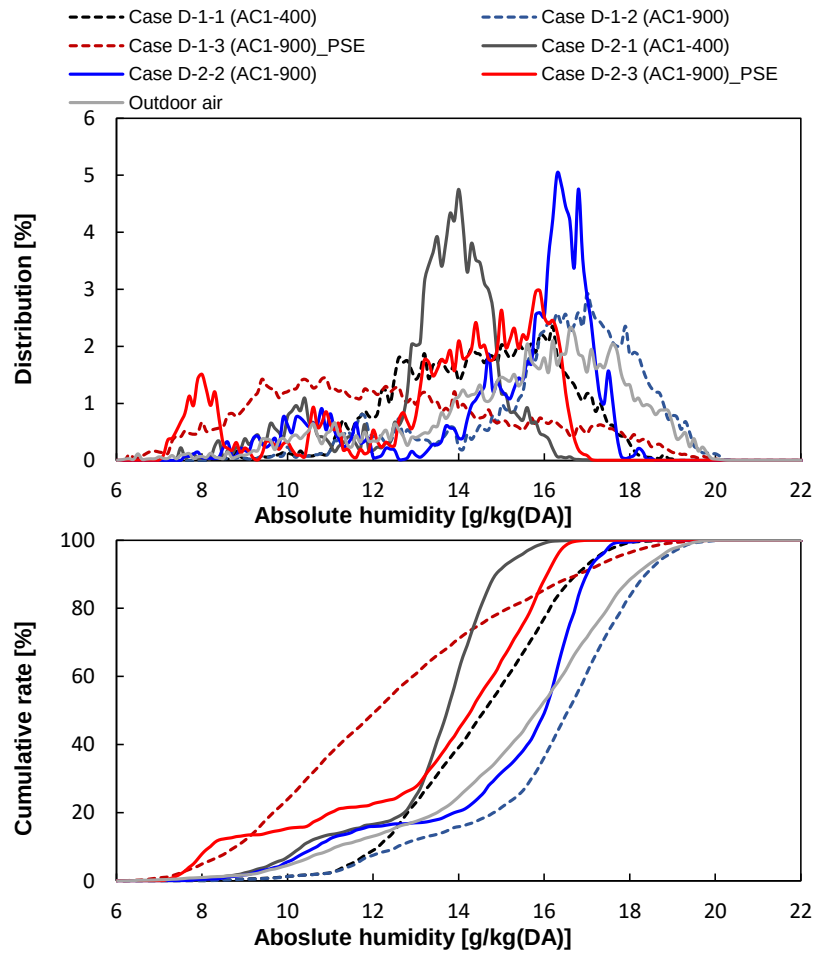


(c) Relative humidity

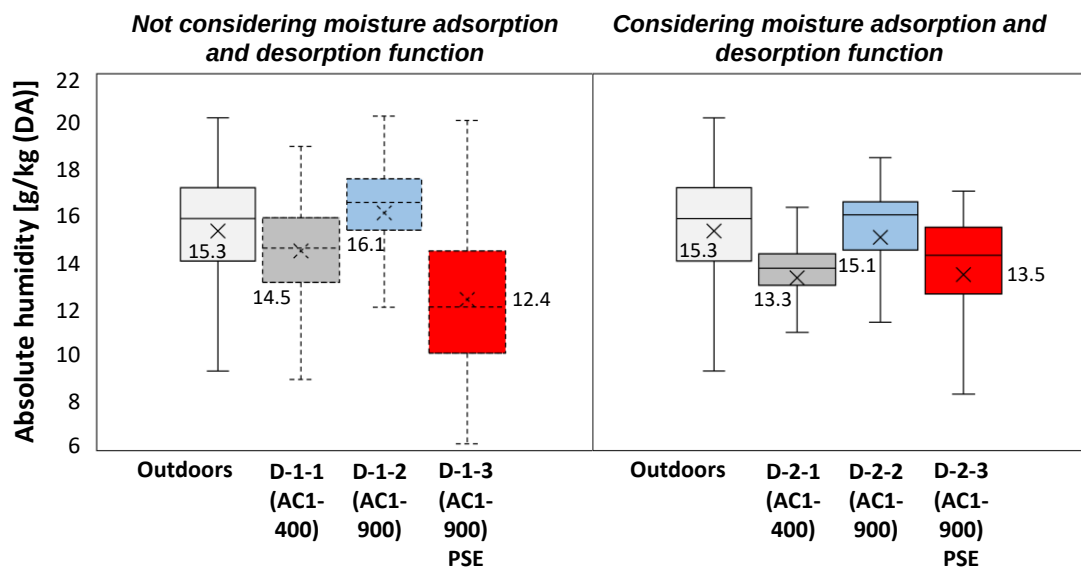
Figure 4.26 Temporal changes in temperature and humidity in LDK for Cases D-1-1~D-2-3—with air-conditioning (August 1 to 7)

Figures 4.27 and 4.28 present the frequency distribution, cumulative rate, and boxplot for absolute and relative humidity in the LDK for Cases D-1-1~D-2-3, spanning July 1 to September 30 (standard year, 2001–2010), during which the air-conditioning was set to 27 degrees using the detailed simulation method. Across the summer, air-conditioning at low airflow rates (Cases D-1-1 and D-2-1) exhibited effective dehumidification, with average indoor absolute humidity lower than outdoor by 0.8 and 2.0 g/kg (DA), and average relative humidity lower than outdoor air by 7% and 12.1%, respectively, for cases without and with building moisture effects considered. Increasing air-conditioning airflow diminished condensation moisture, leading to higher average indoor absolute and relative humidity in Case D-1-2 compared to outdoor air by 0.8 g/kg (DA) and 0.2%, respectively, when building moisture effects were not considered. This suggests that high-airflow rates in air conditioners are insufficient for managing indoor moisture, potentially degrading hygrothermal comfort. The PSE system offset this limitation through its superior dehumidification capability, lowering mean indoor absolute humidity for Cases D-1-3 and D-2-3 by 2.9 and 1.8 g/kg (DA), respectively, and mean indoor relative humidity by 16.3% and 11.7% compared to outdoor air.

Case D: Absolute humidity



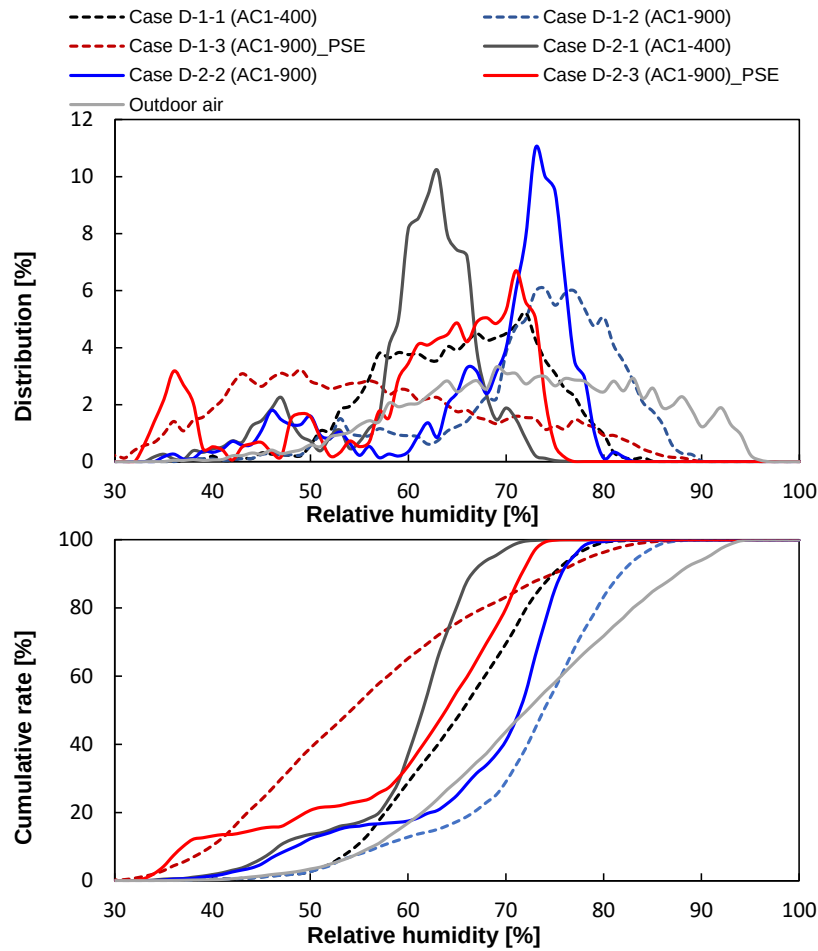
(a) Frequency distribution and cumulative rate



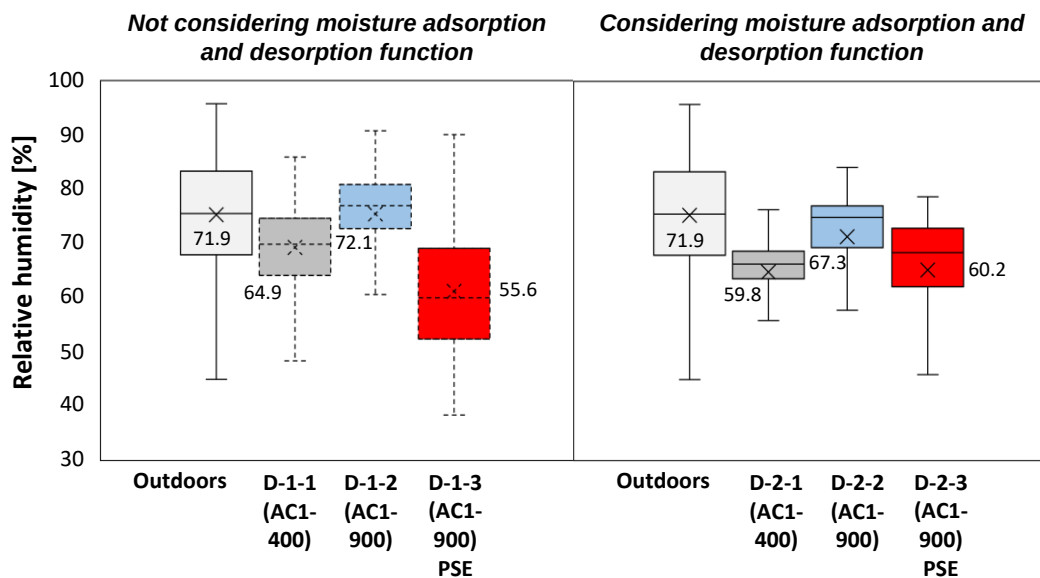
(b) Boxplot

Figure 4.27 Frequency distribution, accumulation rate, and boxplot of absolute humidity for Cases D-1-1~D-2-3 (July 1 to September 30)

Case D: Relative humidity



(a) Frequency distribution and cumulative rate



(b) Boxplot

Figure 4.28 Frequency distribution, accumulation rate, and boxplot of relative humidity for Cases D-1-1~D-2-3 (July 1 to September 30)

Figures 4.29 and 4.30 illustrate the refrigerant evaporation temperature and instantaneous latent heat load for Cases D-1-1~D-2-3. In Cases D-1-1 and D-1-2 (exhaust ventilation system), with low air conditioner airflow, the refrigerant evaporates at 12 to 15 degrees during the day. Figure 4.30(a) shows reduced latent heat load in Case D-1-2 due to decreased evaporator condensation moisture. Case D-1-3, operating under high-airflow conditions with the PSE system, exhibits lower instantaneous latent heat load due to higher refrigerant evaporation temperature and the PSE system's passive indoor dehumidification, reducing indoor moisture content. The pattern for Cases D-2-1~D-2-3 mirrors that of Cases D-1-1~D-1-3, with the PSE system achieving the highest refrigerant evaporation temperature and lowest instantaneous latent heat load.

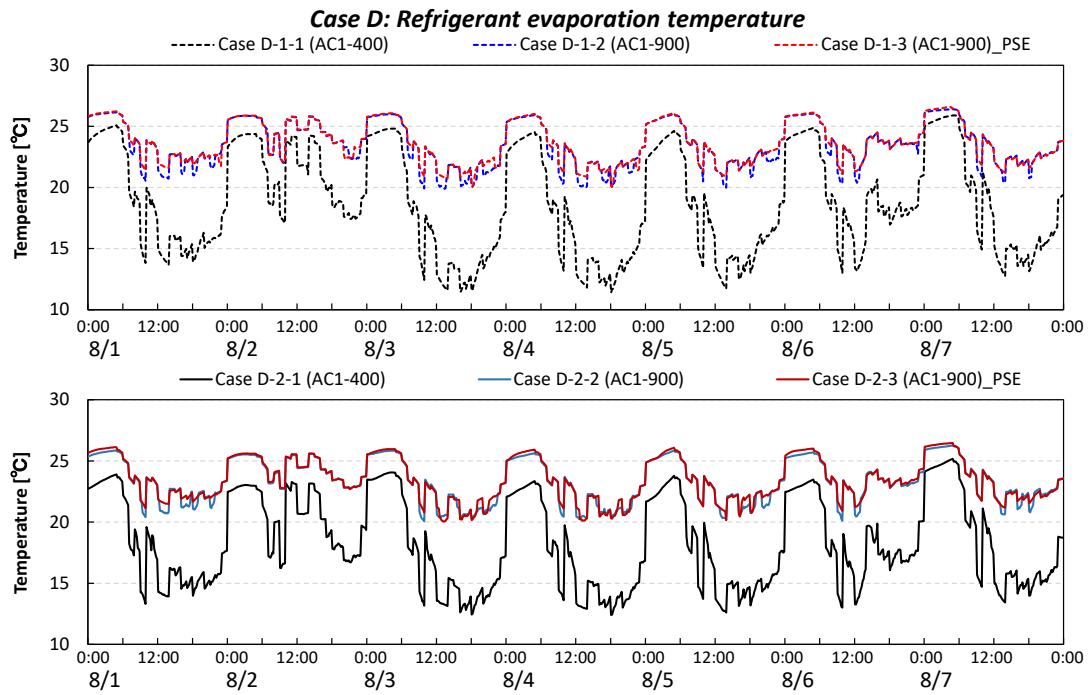


Figure 4.29 Refrigerant evaporation temperature for Cases D-1-1~D-2-3 (July 1 to September 30)

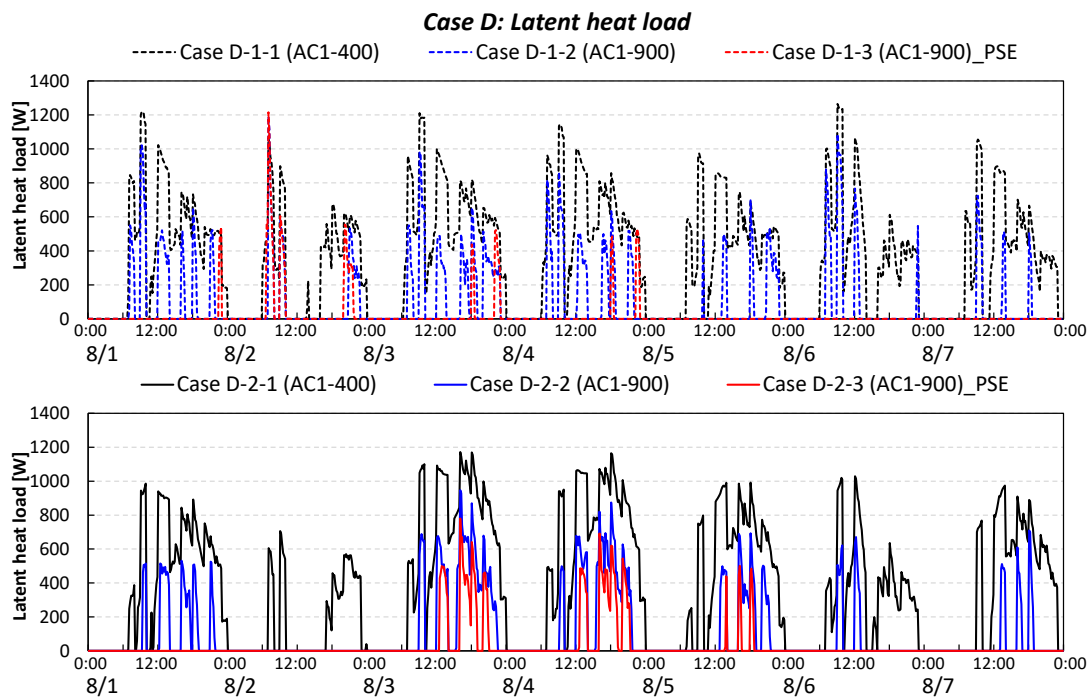
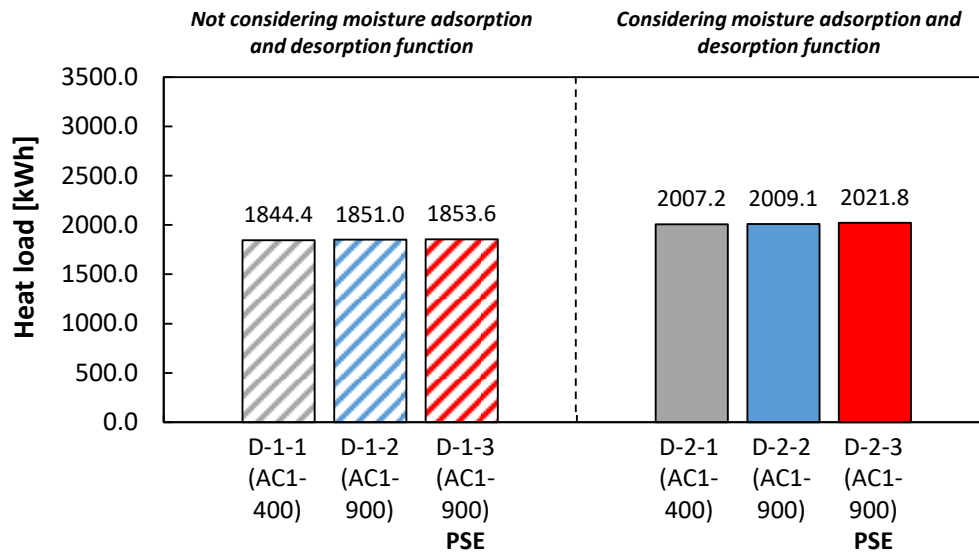


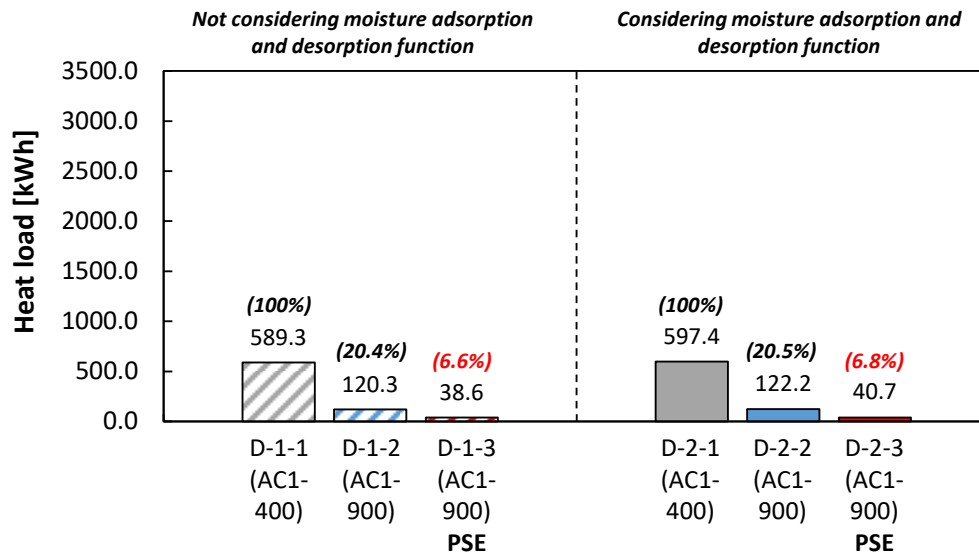
Figure 4.30 Instantaneous latent heat load for Cases D-1-1~D-2-3 (July 1 to September 30)

Figure 4.31 presents simulated heat load and electricity consumption for Cases D-1-1~D-2-3 from July 1 to September 30. Figure 4.31(a) shows negligible differences in sensible heat loads across cases, with higher loads in cases considering the building envelope's moisture adsorption and desorption properties due to adsorption heat increasing cooling demand. Figure 4.31(b) demonstrates that Cases D-1-2 and D-2-2, operating with high-airflow air conditioners, reduced latent heat loads by 79.6% and 79.5%, respectively, though this also increased indoor humidity. Conversely, the PSE system in Cases D-1-3 and D-2-3 achieved the lowest latent heat loads, reduced by 93.4% and 93.2% compared to Cases D-1-1 and D-2-1, thus minimizing total heat load by 22.3% and 20.8%, respectively, as shown in Figure 4.30(c).

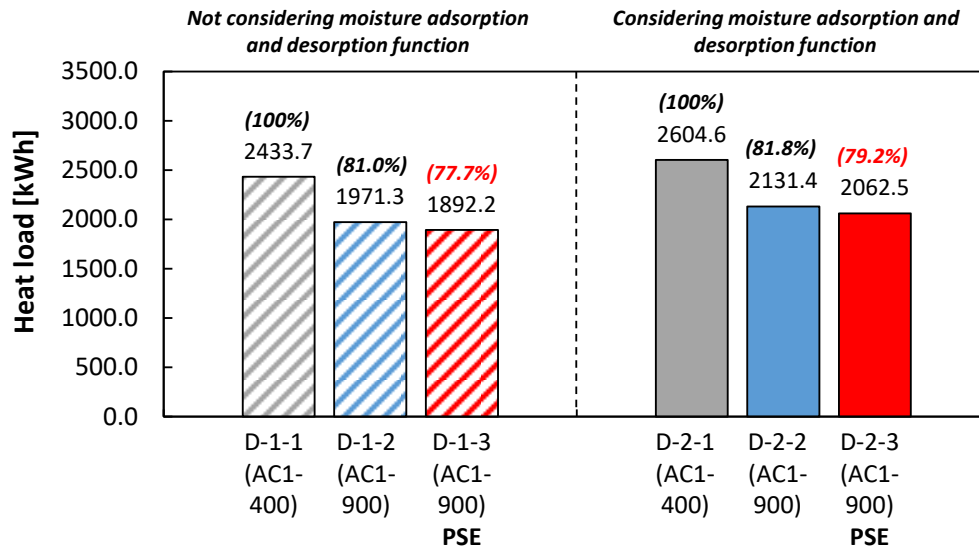
Regarding electricity consumption (Figure 4.30(d)), high airflow in air conditioners lowered consumption by 15.2% and 14.0% in Cases D-1-2 and D-2-2, respectively. Operating with the PSE system under high-airflow conditions, Cases D-1-3 and D-2-3 further reduced electricity consumption by 14.2% and 15.1%, respectively, compared to Cases D-1-1 and D-2-1. While the PSE system's impact on electricity consumption was modest, its dehumidification significantly enhanced indoor thermal comfort.



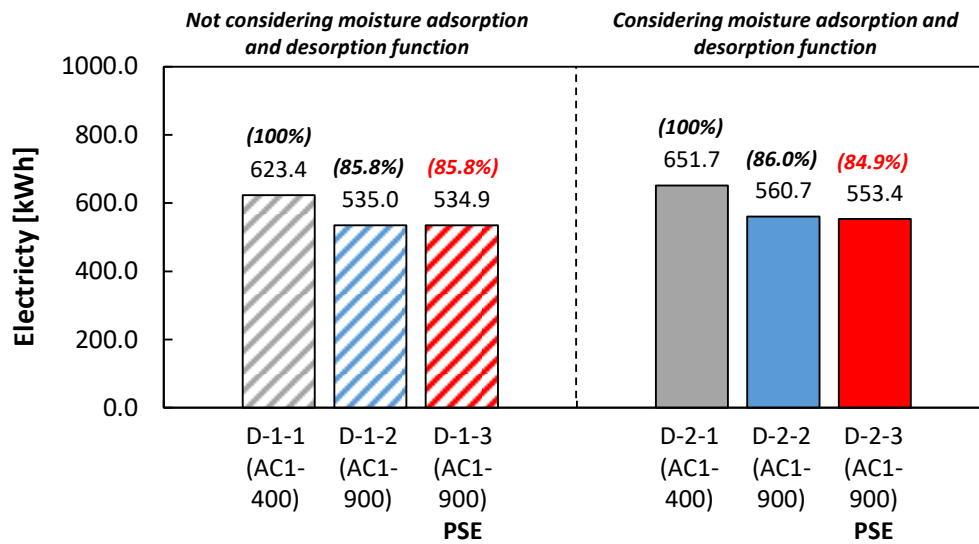
(a) Sensible heat load



(b) Latent heat load



(c) Total heat load



(d) Electricity consumption

Figure 4.31 Heat loads and electricity consumption of whole building for Cases D-1-1~D-2-3 (July 1 to September 30)

4.2.5 Case E: PSE performance evaluation under various air conditioners' performance

It is widely recognized that the power consumption of an air conditioner is influenced by its COP. With a constant cooling load, a higher COP results in lower power consumption. The COP is determined by the refrigerant evaporation temperature during the cooling process; a higher evaporation temperature leads to a higher COP. Consequently, there is an increasing trend among manufacturers to develop high-efficiency air conditioners characterized by higher refrigerant evaporation temperatures. This, however, may reduce the dehumidification effect due to decreased moisture condensation.

In this subsection, parametric simulation analysis was conducted to compare the dehumidification effect of the PSE system at high refrigerant evaporation temperatures using an additional air conditioner with higher efficiency.

4.2.5.1 Simulation conditions

Table 4.13 outlines the simulation cases, comprising eight scenarios all utilizing the PSE system. Cases E-1-1 and E-1-2 employ AC-1, installed in the experimental houses, operating at low (400 m³/h) and high (900 m³/h) airflow rates, respectively. AC-2, used in Cases E-1-3 and E-1-4, showcases improved performance, with operation at low (900 m³/h) and high (1600 m³/h) airflow rates, respectively. Notably, AC-2's low airflow matches AC-1's high-airflow rate. Cases E-2-1~E-2-4 mirror E-1-1~E-1-4 but incorporate the building envelope's moisture adsorption and desorption properties. For AC-1 specifications, please see Table 4.6; AC-2 specifications are detailed in Table 4.14.

Table 4.13 Simulation cases

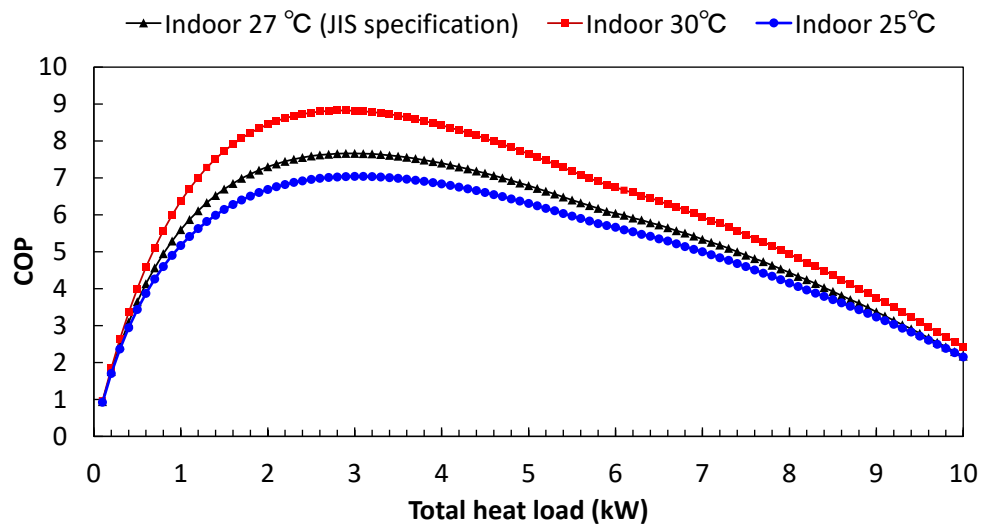
Cases	Consider the moisture adsorption and desorption function of the building envelope	Operation of PSE system	Air conditioner	The air flow rate of air conditioner (m ³ /h)
Case E-1-1 (same with Case C-1-4)	No	Operation	AC-1	400
Case E-1-2 (same with Case D-1-3)				900
Case E-1-3			AC-2	900
Case E-1-4				1600
Case E-2-1 (same with Case C-2-4)	Consider		AC-1	400
Case E-2-2 (same with Case D-2-3)				900
Case E-2-3			AC-2	900
Case E-2-4				1600

Table 4.14 Specifications of AC-2 for simulation input

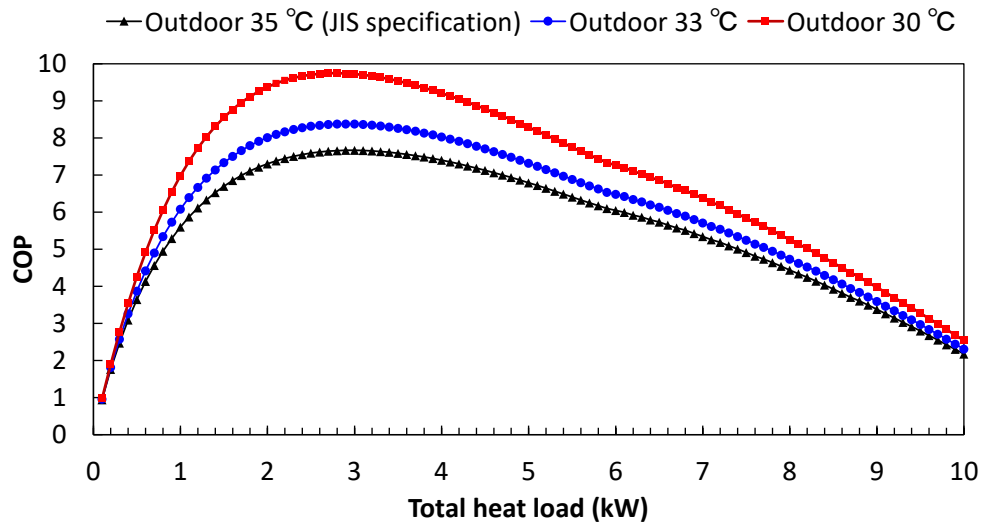
Items	Contents
AC	AC-2
Power consumption	0.145, 3.000, and 3.150 kW (minimum, rated, and maximum)
Cooling capacity	0.6, 9.0, and 9.1 kW (minimum, rated, and maximum)
Heat exchanger airflow rate	Indoor unit: ~1674 m ³ /h Outdoor unit: 3834 m ³ /h
Heat exchanger's bypass factor	0.2

Figure 4.31 illustrates the relationship between cooling loads (up to 10.0 kW, as detailed in Table 4.14) and COP for AC-2, using indoor and outdoor temperature parameters per JIS specifications in Table 4.7.

In Figure 4.31(a), following JIS specifications for indoor humidity and outdoor temperature and humidity, AC-2's COP is calculated across various cooling loads by adjusting the indoor temperature. Similar to the observations in Subsection 4.2.3, a higher indoor temperature under identical cooling loads results in a higher COP. Figure 4.31(b) calculates AC-2's COP by varying the outdoor temperature across different cooling loads, demonstrating that a lower outdoor temperature under the same cooling load yields a higher COP. Moreover, AC-2 achieves a peak COP of approximately 7.7 at an indoor cooling load of approximately 2.9 kW, according to JIS standards.



(a) Indoor temperature as parameter



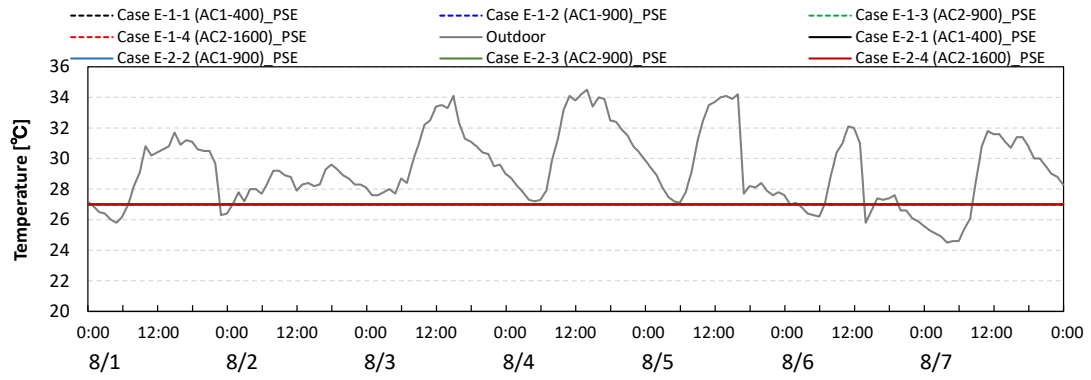
(b) Outdoor temperature as parameter

Figure 4.32 Relationship between total heat load and COP for AC 1

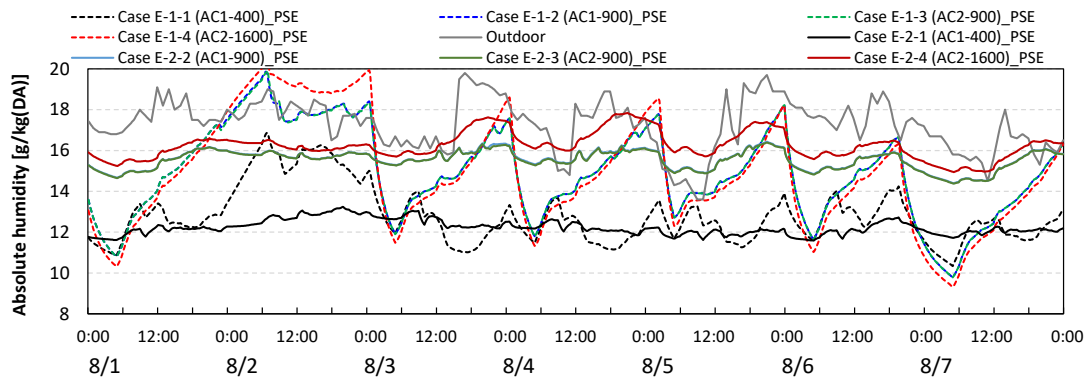
4.2.5.2 Simulation results

Figure 4.33 illustrates the temporal changes in temperature, absolute humidity, and relative humidity for Cases E-1-1~E-2-4 from August 1 to 7 (standard year, 2001–2010), during which the air-conditioning was set to operate at 27 degrees, as per the detailed coupled simulation method. Throughout the summer, the air-conditioning maintains a constant temperature of 27 degrees across all cases, as depicted in Figure 4.33(a). Figure 4.33(b) demonstrates that for both AC-1 and AC-2, increasing the airflow results in a decreased dehumidifying effect, leading to higher indoor absolute humidity. The indoor absolute humidity levels in Cases E-1-2 and E-1-3 are nearly identical, as are those in Cases E-2-2 and E-2-3, attributed to both air conditioners operating at the same airflow rate, resulting in similar evaporative temperatures of the refrigerant and condensation moisture content in the evaporator. Furthermore, considering the moisture adsorption and desorption characteristics in the building envelope stabilizes both absolute and relative humidity fluctuations, as observed in previous findings.

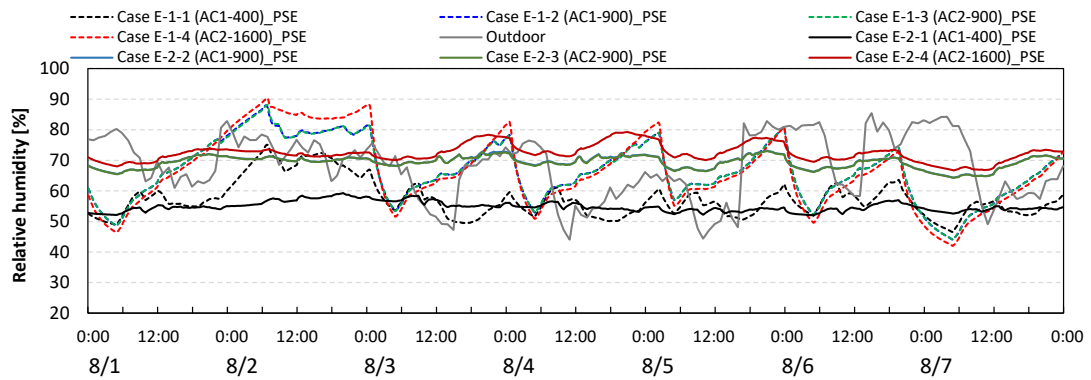
Chapter 4 Simulation analysis by using numerical simulation under several conditions



(a) Temperature



(b) Absolute humidity



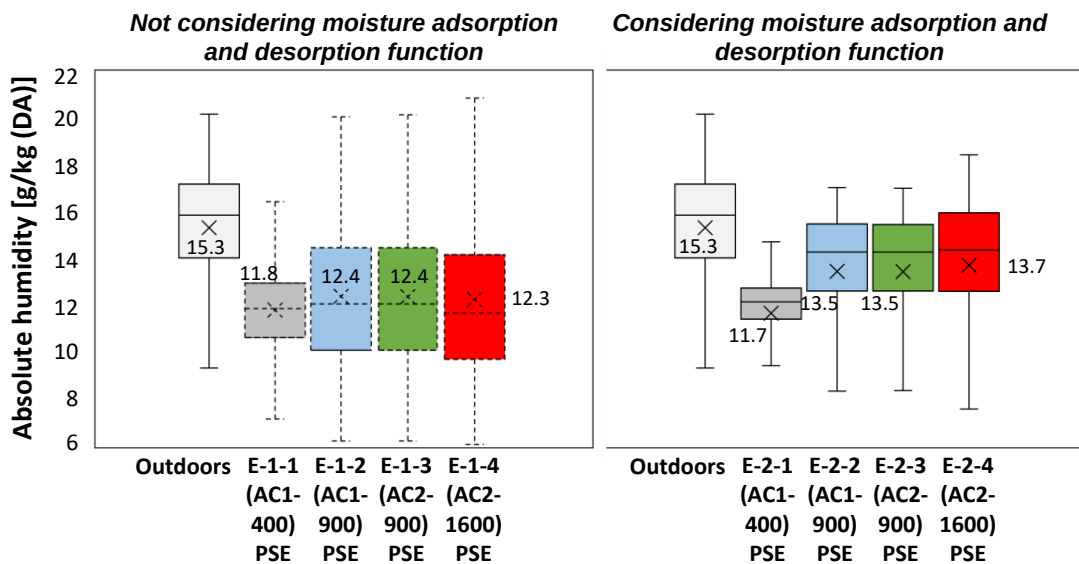
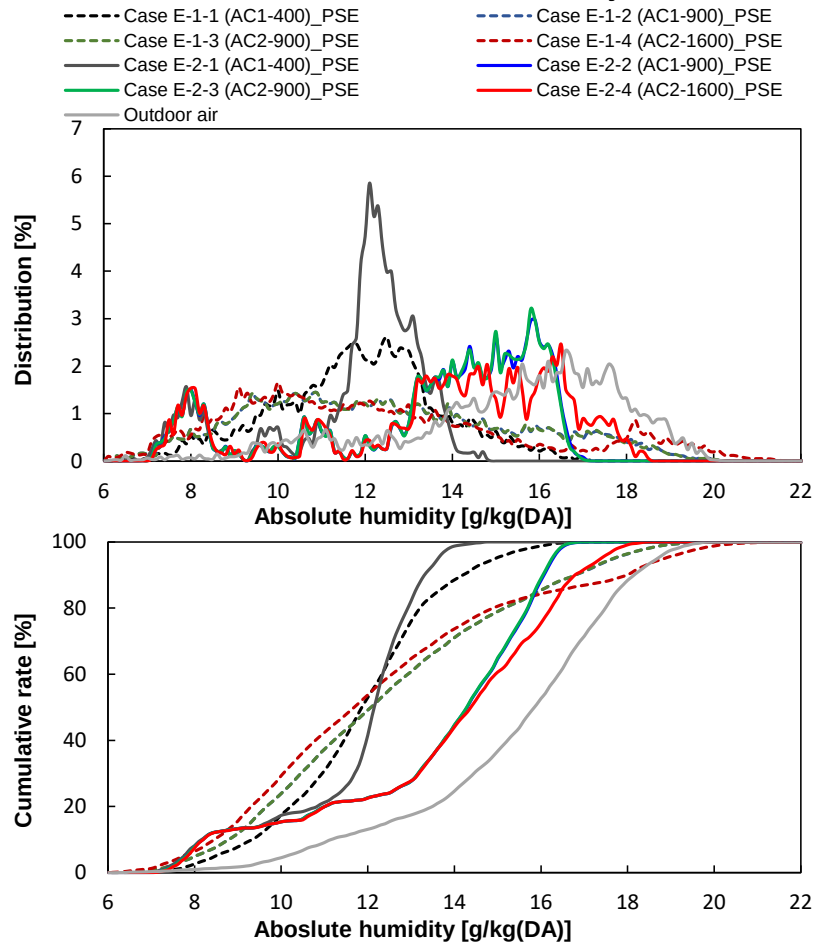
(c) Relative humidity

Figure 4.33 Temporal changes in temperature and humidity in LDK for Cases E-1-1~E-2-4—with air-conditioning (August 1 to 7)

Figures 4.34 and 4.35 present the frequency distribution, cumulative rate, and boxplot analysis of absolute and relative humidity within the LDK for Cases E-1-1~E-2-4, spanning from July 1 to September 30 (standard year, 2001–2010), with the air-conditioning operational at 27 degrees. Additionally, Figures 4.36 and 4.37 detail the refrigerant evaporation temperature and the instantaneous latent heat load for Cases E-1-1~E-2-4.

During summer, increasing AC-1's airflow raises the refrigerant's evaporative temperature (Figure 4.36), thereby diminishing the air-conditioning's condensation dehumidification capacity and reducing the air conditioner's latent heat load (Figure 4.37). Consequently, Cases E-1-2 and E-2-2 exhibit higher indoor absolute and relative humidity levels than Cases E-1-1 and E-2-1. At this time, the indoor dehumidification is mainly accomplished by air-conditioning condensation and the PSE system's passive dehumidification and moisture adsorption. Notably, AC-2's higher efficiency results in a significantly higher refrigerant evaporation temperature (Figure 4.36), and further increasing its airflow renders the air conditioner's condensation dehumidification nearly ineffective, leading to almost zero latent heat load in Cases E-1-4 and E-2-4. As a result, no significant differences in the average indoor absolute and relative humidity levels are observed between Cases E-1-3 and E-1-4, and between Cases E-2-3 and E-2-4, indicating that the only PSE system controls indoor humidity, due to the higher refrigerant evaporation temperature effectively negating the air conditioner's dehumidification capability by condensation.

Case E: Absolute humidity



(b) Boxplot

Figure 4.34 Frequency distribution, accumulation rate, and boxplot of absolute humidity for Cases E-1-1~E-2-4 (July 1 to September 30)

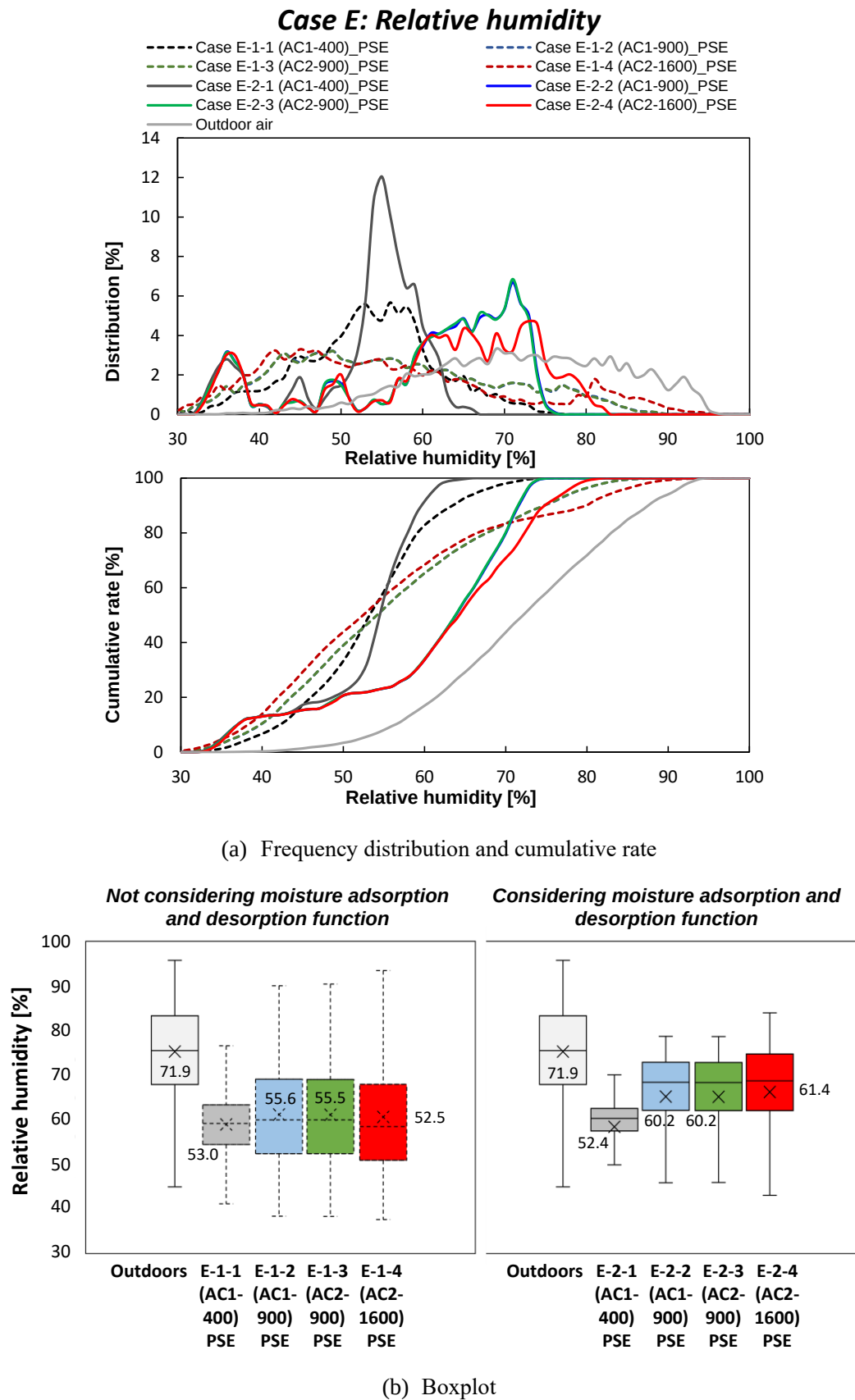


Figure 4.35 Frequency distribution, accumulation rate, and boxplot of relative humidity for Cases E-1-1~E-2-4 (July 1 to September 30)

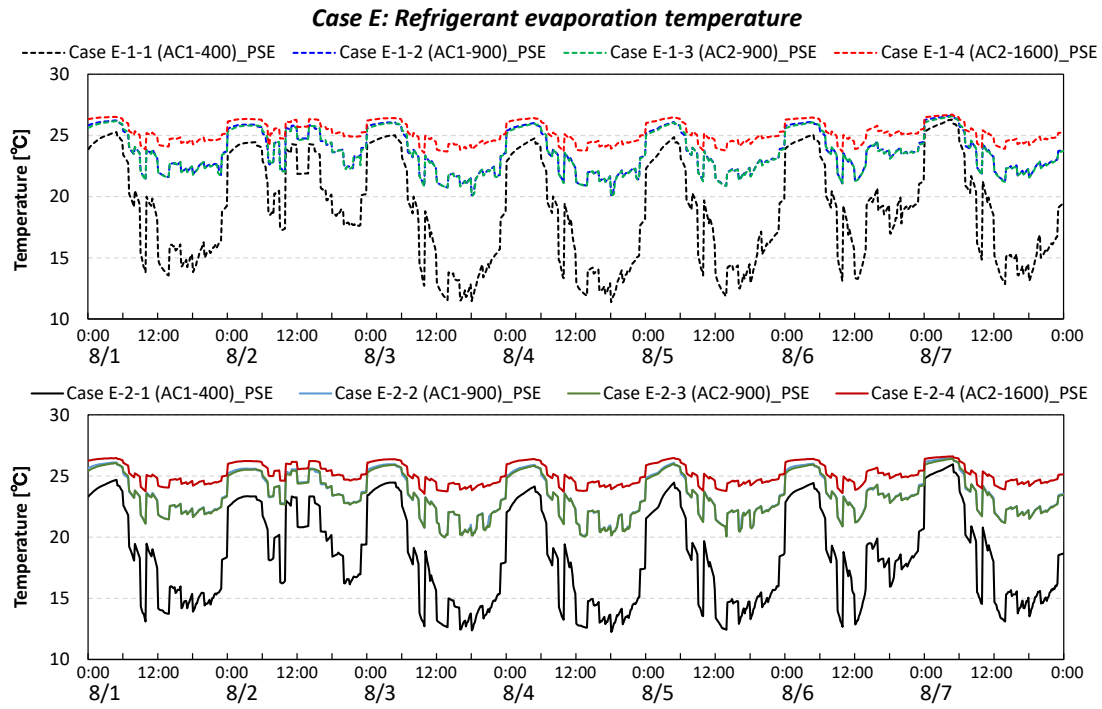


Figure 4.36 Refrigerant evaporation temperature for Cases E-1-1~E-2-4 (July 1 to September 30)

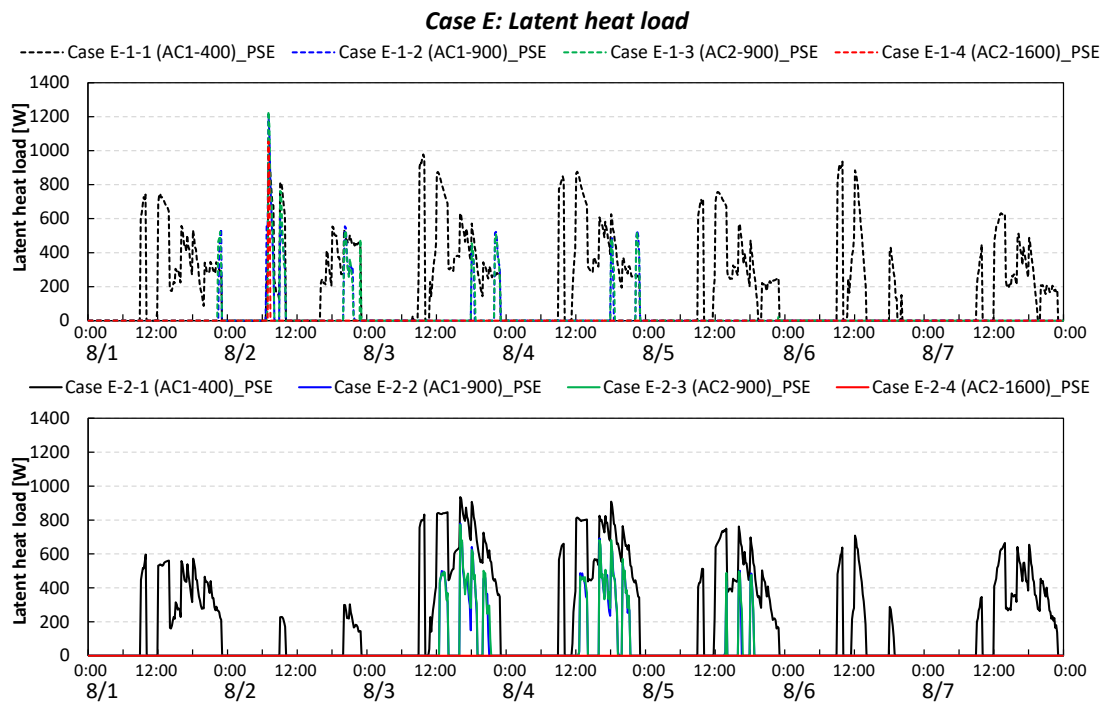
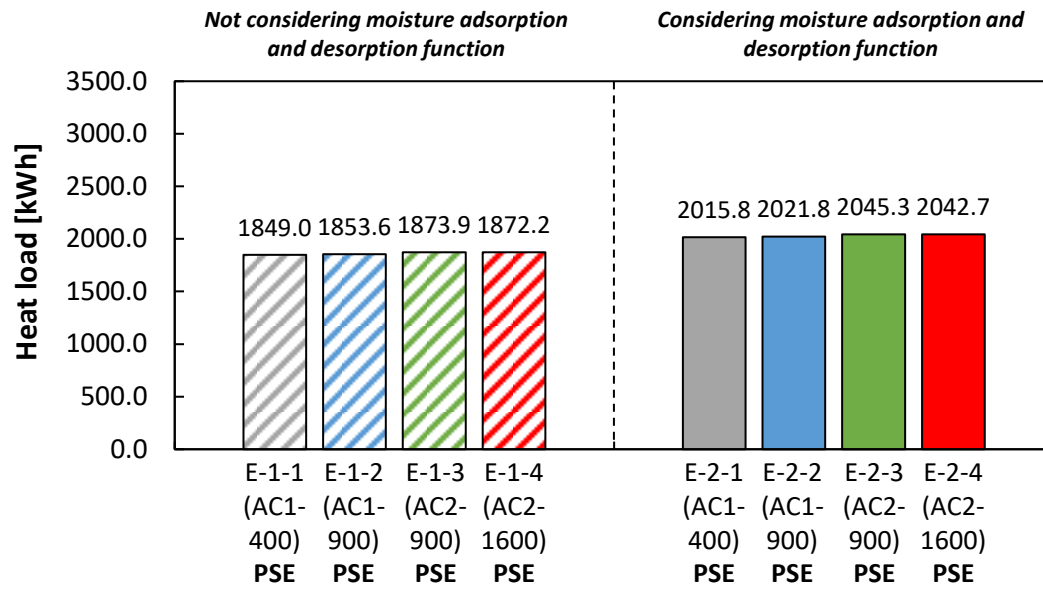


Figure 4.37 Instantaneous latent heat load for Cases E-1-1~E-2-4 (July 1 to September 30)

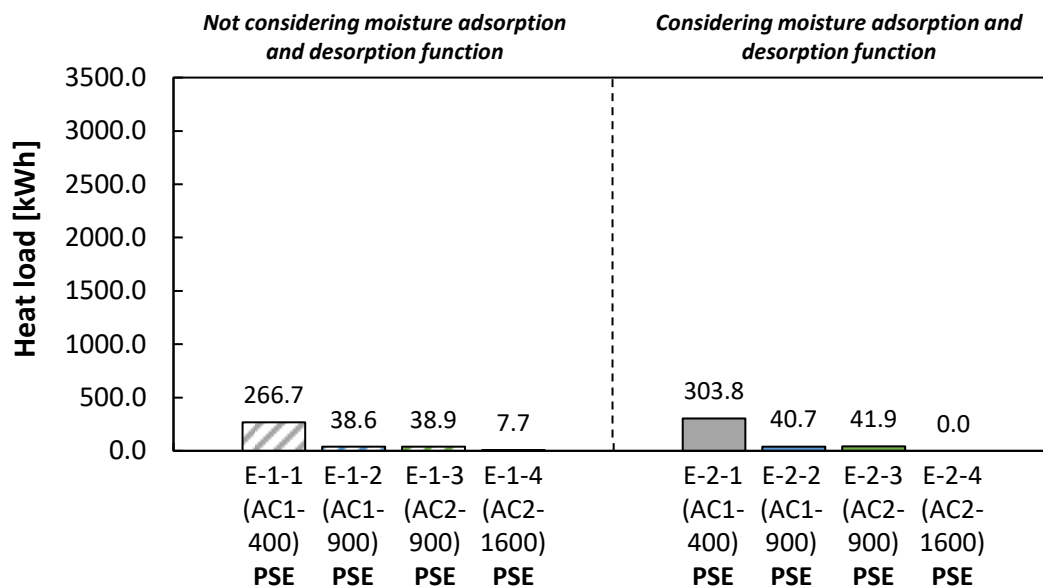
Figure 4.38 displays the simulated heat load and electricity consumption for Cases E-1-1~E-2-4 from July 1 to September 30. With respect to the sensible heat load results (Figure 4.38(a)), cases considering the moisture adsorption and desorption properties of the building envelope exhibited higher sensible heat loads than those that did not, due to the heat generated by the moisture-buffering effect of building materials. For the latent heat load results (Figure 4.38(b)), the indoor latent heat load under high-airflow conditions was lower than under low airflow conditions when using AC-1, showing a reduction of 38.6~40.7% throughout the summer, indicating minimal dehumidification by the air conditioner. With AC-2, the latent heat load due to air-conditioning dehumidification was almost identical at the same airflow rate (900 m³/h) as AC-1. Increasing AC-2's airflow resulted in a latent heat load of zero. Concerning the total heat load results (Figure 4.38(c)), AC-1 at low airflow conditions exhibited the highest total heat load, while the latent heat loads for Cases E-1-2~E-1-4 and E-2-2~E-2-4 were low to nearly zero, leading to similar total heat loads among these cases.

Regarding electricity consumption, compared to Cases E-1-2 and E-2-2, using the high-performance air conditioner AC-2, under conditions achieving the same indoor dehumidification effect, resulted in lower electricity consumption for Cases E-1-3 and E-2-3, with reductions of 32.2% and 32.7%, respectively. This reduction is attributed to AC-2's higher COP compared to AC-1 for the same cooling capacity. Moreover, increasing AC-2's airflow decreased electricity consumption, but the indoor humidity levels remained constant. This outcome suggests that in Cases E-1-4 and E-2-4, a separate control for sensible and latent heat was achieved, with the PSE system managing indoor humidity.

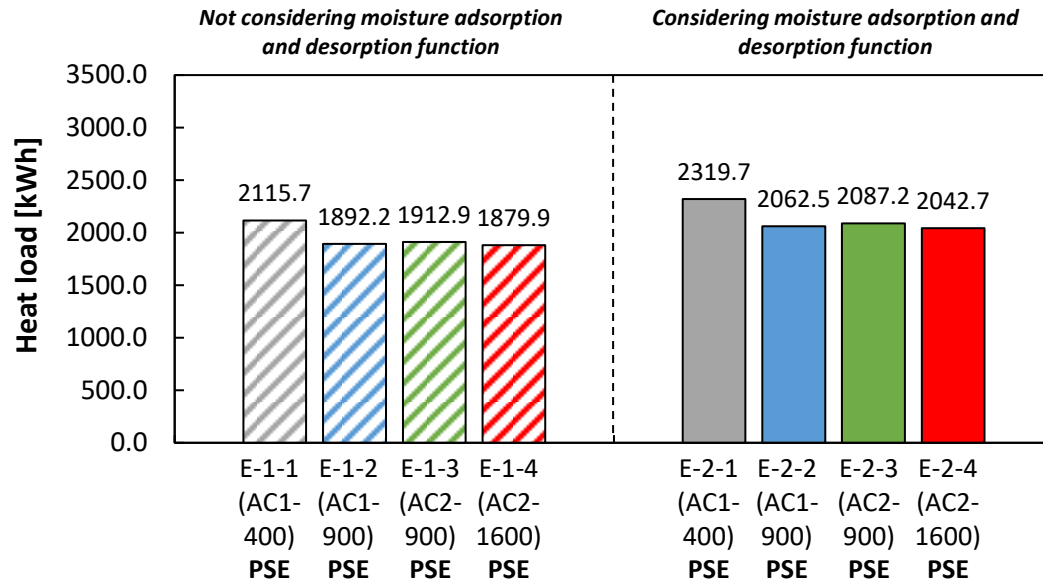
In summary, using high-efficiency air conditioners with higher refrigerant evaporation temperatures results in lower electricity consumption, yet such air conditioners struggle to dehumidify the room effectively. Conversely, integrating the PSE system enables room dehumidification while maintaining a hygrothermally comfortable indoor environment.



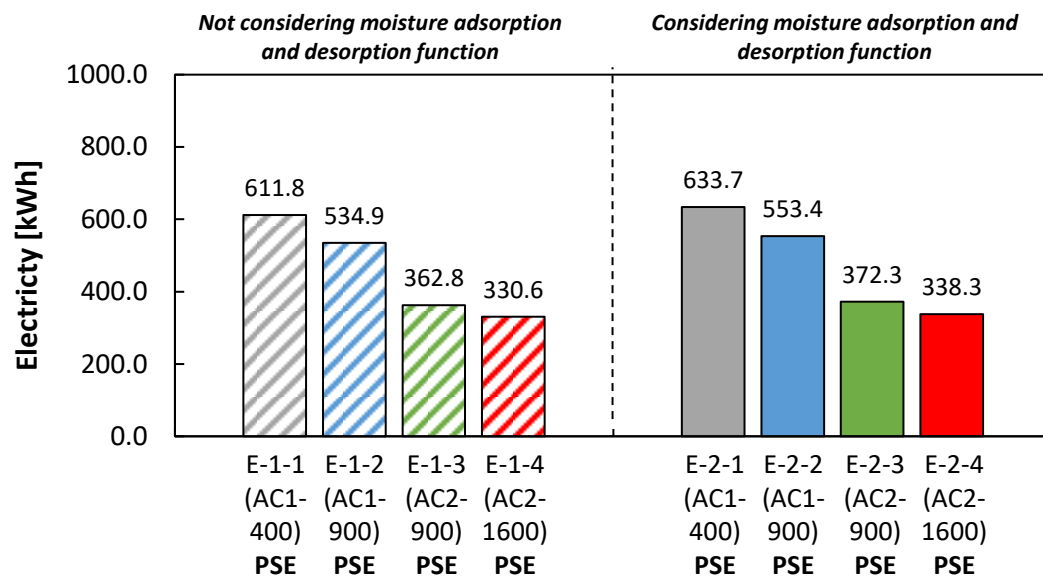
(a) Sensible heat load



(b) Latent heat load



(c) Total heat load



(d) Electricity consumption

Figure 4.38 Heat loads and electricity consumption of whole building for Cases E-1-1~E-2-4 (July 1 to September 30)

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Chapter 5 Conclusions and discussions

5.1 Conclusions

We proposed a passive dehumidification and mechanical ventilation system that integrates energy recovery ventilation (ERV) into a passive dehumidification and solar collection (PDSC) system, referred to as the PSE (PDSC and ERV) system. The PSE system can control indoor humidity and save energy for air conditioning systems in residential buildings. Experiments and numerical simulations were conducted to examine the summer dehumidification and latent heat load reduction effect of the PSE system.

Chapter 3 detailed a series of controlled measurement experiments compared to standard ventilation systems (exhaust ventilation and ERV systems). The findings indicated that:

- 1) Under conditions without air conditioning, the PSE system exhibited the most substantial dehumidification effect among the tested systems over a prolonged measurement period. The PSE system maintained an indoor/outdoor absolute humidity difference of approximately 3.4 g/kg (DA), with indoor relative humidity consistently below 56%, aligning with ASHRAE's recommended comfort humidity level.
- 2) With air conditioning operational, the PSE system achieved differences in indoor and outdoor absolute humidity and relative humidity of 4.9 g/kg (DA) and 16.1%, respectively, also presenting the lowest ventilation latent heat load.

Chapter 4 utilized the high-precision hygrothermal simulation software THERB for HAM to evaluate the PSE system's performance under various conditions. The simulation cases were as follows:

- 1) In Case A, the indoor temperature and humidity environment of the PSE system and other systems (exhaust ventilation, ERV, PSE systems) were predicted using THERB for HAM under un-air-conditioned conditions, revealing:
 - The PSE system demonstrated the most effective dehumidification, with an indoor/outdoor absolute humidity difference ranging from 2.8 to 5.0 g/kg (DA).
 - The PSE system's dehumidification was enhanced when the building envelope possessed moisture adsorption and desorption capabilities, due to the hygroscopic materials adsorbing moisture.
 - The indoor relative humidity remained low across cases due to high temperatures without cooling, with the PSE system's indoor relative humidity as low as 29.2% to 21.4%.
- 2) In Case B, the indoor temperature and humidity and loads of the PSE system and other systems (exhaust ventilation, ERV, PSE systems) were studied using THERB for HAM under air-conditioned conditions with constant temperature and relative humidity settings, the PSE system:
 - Showed the best dehumidification effect, lowering absolute humidity by 2.7 g/kg (DA) to 3.4 g/kg (DA) and relative humidity by 15.3% to 18.3% compared to outdoor air.

- Achieved the greatest reduction in latent and total heat load, with reductions of 70.8% to 75.9% and 14.5% to 21.5%, respectively.
- 3) In Case C, a detailed simulation method coupled with THERB for HAM software and a heat pump model was used to investigate the performance of the PSE system compared with other systems. Simulations were under constant temperature settings and found that the PSE system:
- Reduced indoor absolute humidity by 3.6 g/kg (DA) and relative humidity by 19.5%, and latent and total heat load by 49.1% to 56.4% and 10.9% to 13.3%, respectively, while saving 1.9% to 2.8% in electricity consumption.
 - Provided a smoother relative humidity fluctuation and a more comfortable indoor hygrothermal environment when implemented in a residence with building materials featuring moisture adsorption and desorption properties.
 - Offered an exceptional dehumidification effect that compensated for the lower efficiency of the air conditioner's condensation dehumidification, reducing the indoor environment's absolute and relative humidity while enhancing thermal comfort with almost the same electricity consumption as other systems.
- 4) In Case D, we explored the PSE system's dehumidification effect under high airflow air conditioning conditions through detailed simulation. These simulations were conducted under constant temperature settings. Achieving thermal comfort in conventional residential buildings solely with a typical heat pump for temperature and humidity control presents challenges. Recently, there has been a trend to increase the refrigerator evaporation temperature (such as by increasing the air conditioner's indoor airflow rate) to reduce electricity consumption. However, this approach diminishes the air conditioner's dehumidification capacity, potentially leading to moisture accumulation indoors, especially when indoor moisture is generated. The PSE system addresses this by providing separate sensible and latent heat air conditioning effects, with temperature control managed by the air conditioner and humidity control by the PSE system. This system compensates for the diminished dehumidification performance of energy-saving air conditioners. Under high airflow air conditioning settings, the PSE system was able to reduce indoor absolute humidity by 1.8-2.9 g/kg (DA) and relative humidity by 11.7-16.3%, while also achieving a 14.2% to 15.1% savings in electricity consumption.
- 5) In Case E, the impact of the PSE system under different air conditioner performances was examined using detailed simulation. Using high-efficiency air conditioners with higher refrigerant evaporation temperatures reduces electricity consumption but significantly limits the units' ability to dehumidify indoor air. Conversely, integrating the PSE system enables room dehumidification and maintains a comfortable indoor hygrothermal environment.

5.2 Discussions

The simulation cases in this study highlight differences in building envelopes, specifically those without and with moisture adsorption and desorption properties. This distinction is crucial as most traditional Japanese residential building envelopes do not incorporate moisture adsorption and desorption characteristics. When the building envelope is hygroscopic, the low humidity environment created by the dehumidification capabilities of air conditioning (with low refrigerant temperatures) and PSE systems can cause the building materials to release moisture back into the indoor environment, as observed in Cases B. However, when air conditioning performance is high (resulting in a diminished dehumidification effect), this issue is mitigated, as demonstrated in Case E.

In the measurement experiments during cooling, the indoor temperature was low. Therefore, the temperature of the roof ventilation layer often failed to drop below the set temperature at night, resulting in a reduction of the operation time for the PDSC system at night. This issue will be improved in future measurement experiments by modifying the temperature determination conditions for nighttime air circulation.

The dehumidification effect of the PSE system under different air conditioning airflow rate conditions is discussed in Chapter 4. However, the electricity consumed due to the fan speed change is ignored, which needs to be examined in future studies.

Furthermore, while the PSE system is adaptable for use in both winter and summer, this study focuses exclusively on evaluating its passive dehumidification performance during the summer months. Future research should delve into the PSE system's heat collection and humidity control performance during winter, providing a comprehensive analysis of its year-round efficacy.

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