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Comparison of the environmental, energy, and thermal comfort performance of air and radiant cooling systems in a zero-energy office building in Singapore

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Abstract

In an experimental study set in Singapore's tropical climate, we evaluated the thermal environmental performance, energy consumption, and thermal comfort of air and radiant cooling systems, operating at an operative and air temperature of 26 °C. 78 participants across five groups answered thermal comfort surveys in a crossover study design. Environmental performance metrics indicated that both systems produced similar conditions, with a noticeable difference in air velocity. The mean radiant temperature to air temperature difference was less than 0.5 °C in both systems. The radiant system exhibited a 33% higher heat flux extraction than the air system and required less electrical power for the transportation of the cooling medium and ventilation air. Overall, the radiant system used 4% less energy than the air system when controlled at 26 °C and 34% when operated at 23 °C. Results show that radiant and air systems provided equal thermal comfort in cooling, with over 60% of participants expressing satisfaction and ~20% voted neutral thermal satisfaction. ~40% of participants preferred cooler conditions, and ~30% desired increased air movement.

Keywords

radiant cooling; mean radiant temperature; energy efficiency; thermal comfort; thermal environment performance

1. Introduction

Radiant cooling systems have the potential to be a sustainable and energy-efficient alternative to traditional air conditioning methods [1,2]. Noteworthy implementations of this technology can be seen in several landmark buildings worldwide, such as the Bullitt Center in Seattle, the California Academy of Sciences in San Francisco, the Pearl River Tower in Guangzhou, the Diamond Building in Kuala Lumpur, and the Crystal (City Hall) in London. Despite implementation in some landmark buildings, questions about energy efficiency, environmental performance, and associated thermal comfort of occupants remain [3]. Radiant cooling technology consists of a diverse array of hydronic systems that employ water as the primary medium for cooling, circulate it through surfaces (e.g., floor, slab, metal panels mounted on the ceiling), reduce surface temperature, and are designed to exchange heat with occupants by radiation. Based on the placement of the piping within the building structure, they can be classified as embedded surface systems, thermally active building systems (TABS), capillary surface systems, and radiant panels [4]. From the thermal time response perspective, the radiant cooling systems can be divided into fast-response systems (radiant ceiling panels and thin floor overlays), moderate-response systems (capillary mats and embedded surface systems in light construction), and slow-response systems (TABS and embedded systems in thick slabs) [5]. Each of the types has advantages and some challenges.

Rhee et al. [6] reviewed literature on radiant heating and cooling, summarizing that, compared to air systems, radiant systems save energy by (i) reducing the energy required to transport cooling medium, (ii) having higher chiller efficiency due to the ability to utilize higher chilled water temperature as a primary cooling source, and (iii) potential to provide equal thermal comfort with higher air temperature.

Most of the comparisons between the systems are done via simulation or comparison of different buildings. A direct comparison of radiant and air systems performance in the same building is very rare. One of the real buildings that enable such comparison is the Infosys building in Hyderabad, India. The building has two identical wings, one cooled with a variable air volume (VAV) system and the other with a radiant cooling system. The comparison of measured energy use was 34% lower on the radiant side [7]. An energy efficiency analysis showed that the radiant system outperformed a conventional all-air system by 17.5% [8]. Simulation of identical buildings in various climate zones in India showed a maximum energy saving of 27% in temperate climates and a minimum of 11% in warm and humid climates [9]. In spaces highly affected by solar radiation, radiant cooling systems can achieve ~12% energy savings compared to air-based cooling systems [10]. Kwong et al. [11] reported that the radiant slab cooling, compared to the conventional VAV system, reduced energy consumption by 34%. TABS can reduce peak thermal load by thermal load shifting [12].

Besides effective and energy-saving radiant cooling systems, examples of underperforming systems also exist. A critical review by Zhang et al. [13] states that most research studies concluded that the energy-saving potential of radiant cooling technology in China still needs to meet the anticipated goals. An earlier review by Hu and Niu [14] made a similar conclusion about energy savings for radiant cooling implemented in different climate zones in China. A similar result was also found by Tian and Love in Canada [15]. Radiant systems have some challenges compared to air systems. With the equivalent heat gains, a radiant cooling system must extract more heat than the air system [16,17]. This research led to initiatives to revisit design loads for radiant cooling systems [18,19]. An example of this is an increase of extracted heat flux from 32 to 110 W/m^2 for radiant floor direct solar exposure [20].

A literature review of a comparison between all air and radiant cooling systems by Karmann et al. [21] concluded that there is no definitive answer to whether buildings using radiant cooling perform better than all air cooling from the thermal comfort perspective. Theoretical arguments supporting radiant cooling as the more advanced thermal comfort cooling system include reduced air movement, active control over mean radiant temperature (MRT), and the ability to create a homogeneous indoor environment. Based on a survey conducted by Sastry and Rumsey [7] in the Infosys Software Development Block 1 in Hyderabad, India, it was concluded that an increase in thermal comfort satisfaction from 45% in the VAV wing to 63% in the radiant-cooled wing. Karmann et al. [3] analyzed indoor environmental quality survey results from 3892 respondents in 60 office buildings located in North America; 34 of which used air systems and 26 of which used radiant systems as the primary conditioning system. This is the largest dataset used in a comparison of occupant satisfaction in radiant buildings. They found that radiant and air spaces have equal indoor environmental quality, including acoustic satisfaction, with a tendency towards improved temperature satisfaction in radiant buildings.

The amount of information about radiant systems in the tropics is limited. Shakya et al. [22] used computational fluid dynamics (CFD) modeling to demonstrate that high-temperature radiant cooling could significantly improve comfort in Singapore's tropical climate compared to natural ventilation. Saber et al. [23], using the predicted mean vote (PMV) model in an environment cooled with a dedicated outdoor air system (DOAS) and radiant panels in Singapore, showed that occupants can maintain partial thermal comfort during the day with a setpoint air temperature of 26 °C. Studies based on environmental simulation and thermal comfort modeling provide support for radiant cooling as a more effective way to provide comfort for the occupants; however, there are some questions related to the accuracy of thermal comfort modeling. Several thermal comfort field studies have identified discrepancies between the PMV index and the actual thermal sensations of occupants [24–26]. Schellen et al. [27] showed that participants felt colder than the PMV index predicted. Cheung et al. [28], using the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Global Thermal Comfort Database II [29], showed the low accuracy of PMV models in predicting thermal comfort in the field using ~57,000 thermal sensation votes collected in buildings. Considering the low accuracy of the PMV index, we need to consider the results from the modeling studies with caution.

Thermal comfort evaluation of an outdoor radiant infrared membrane cooling in Singapore with surface temperatures below the dew point showed a "satisfactory" thermal sensation of 79% of participants experiencing air temperature of 29.6 °C, MRT of 23.9 °C at RH 66.5%, with air movement of 0.26 m/s and [30–33]. This type of radiant cooling enables a ~6 °C difference between air temperature and MRT, suggesting that cooling is dominated by radiant heat exchange. Using measurements from 48 office buildings in the ASHRAE Global Thermal Comfort Database II, five field studies in radiant and all-air buildings, and five test conditions from a laboratory experiment, Dawe et al. [34] showed that measurement of the air temperature is sufficient to estimate the MRT under typical office conditions, and that separate measurement of MRT, or a further calculation of operative temperature (the average of the mean radiant and air temperatures, weighted by their respective heat transfer coefficients), is not likely to have practical benefits to thermal comfort in most cases. They concluded that this is especially true for buildings with radiant systems. Black globes and smaller ping-pong size black globe sensors are the most frequently used instruments for the measurement of MRT. A systematic measurement error exists in these MRT measurement methods [35]. Several publications showed that standard correction readings in low-temperature radiant cooling systems could be up to several degrees Celsius higher or lower than the ground truth readings [35–37]. Since the difference between air temperature and MRT is critical, measuring it accurately and understanding its value and impact on comfort is very important.

Energy savings achieved by setting the operative temperature to 26 °C depend on maintaining a comfortable environment for building occupants. Traditional thermal comfort models, like PMV, do not accurately reflect occupants' true thermal comfort experience. Alongside the limited accuracy of the PMV model, there are some concerns about the accuracy of MRT measurements, one of the key factors impacting radiant heat exchange, thermal environmental performance, and subsequent comfort of occupants. As highlighted by Karmann et al. [21], there is a significant need for more research on thermal comfort within actual buildings to fully understand the relationships between energy-efficient operation setpoints, actual energy savings, and occupant comfort. The current study comprehensively evaluates environmental performance, energy savings, and delivered occupant comfort in a real high-performance building fitted with interchangeable air and radiant cooling systems. Regarding energy savings and environmental performance, this study expands the existing body of knowledge by adding information measured in a specific building in a tropical climate. The thermal comfort aspect of this study is unique and comprehensive since it is the only comparison of air and radiant cooling systems measured in the same space with both systems in a real building. This study provides a step further in understanding thermal satisfaction, perception, and preference of people exposed to both systems.

2. Methodology

The experiments were conducted at a zero-energy office building named Zero Energy Plus Building (ZEB Plus) [38] at the Building and Construction Authority (BCA) Braddell campus in Singapore, as shown in Figure 1a. ZEB Plus, one of the first retrofitted net zero energy buildings in Asia, hosts living labs that testbed various energy-efficient cooling technologies and generate energy via solar panels to compensate for its consumption. The hybrid cooling system (elevated air movement and increased temperature setpoint air-conditioning system) was tested on the first floor [39], and the fast-response radiant cooling system (this study) was tested on the second floor. To evaluate environmental performance, energy consumption, and thermal comfort between air and radiant systems, we designed three sets of experiments.

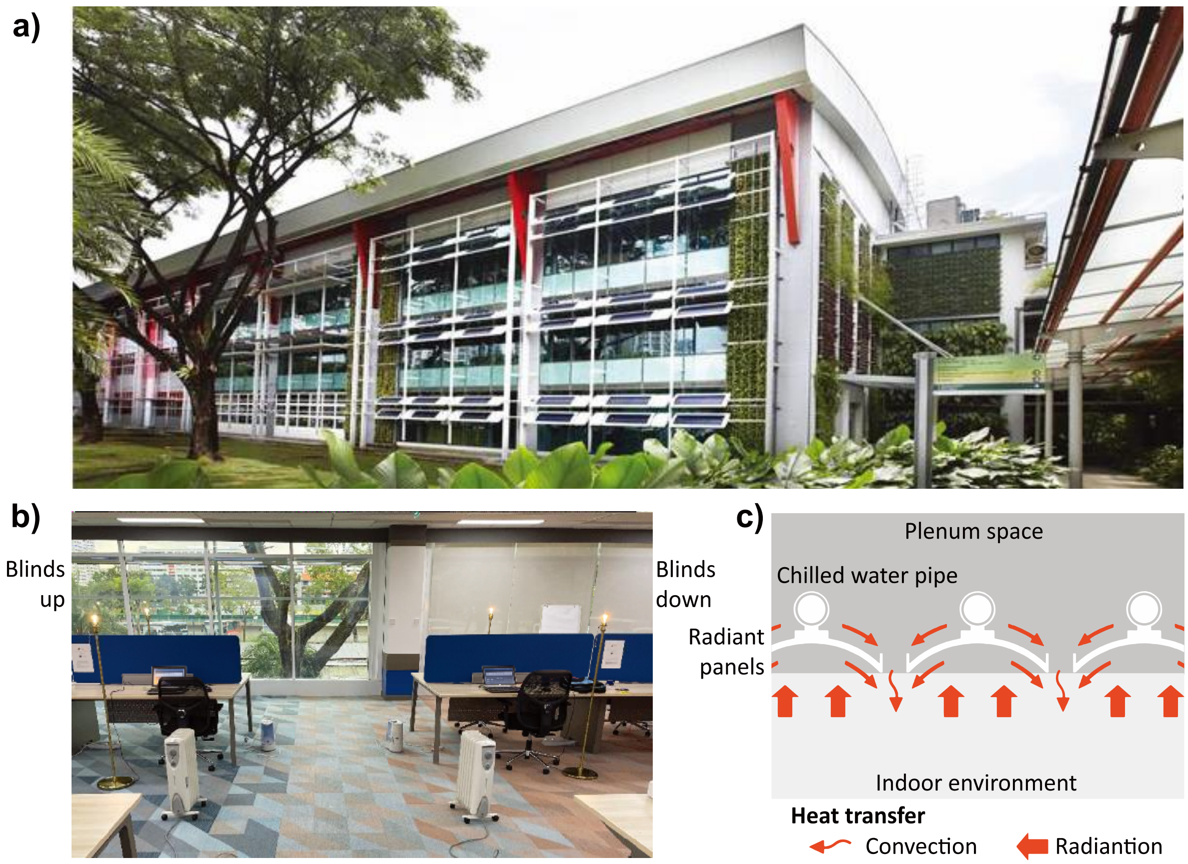

Figure 1. a) Photo of ZEB Plus. b) Photo of the window with half of the blinds up and half down. c) Schematic diagram of the curved radiant panels.

The first set of experiments was designed to evaluate the method for MRT measurement, which is necessary to understand the difference between the air and mean radiant temperatures. The building had floor-to-ceiling glass walls with blinds installed on the interior (Figure 1b). The second objective of the first set experiment was to understand the impact of blinds (outer wall with no blinds or fully covered with blinds) on the difference between mean radiant and air temperatures in the room.

The second set of experiments was designed to simulate the real environment and load fluctuations from 100% occupancy to 33% occupancy. We characterized the thermal environment and compared the energy consumption of the air system and the radiant cooling system. ASHRAE 55 [40] defined thermal comfort as: "The condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation." By ASHRAE definition, thermal comfort can be assessed using a survey of occupants, and to assess thermal comfort, we designed a set of experiments where we recruited subjects and assessed their thermal comfort using the “right now” survey of the updated ASHRAE 55 questions.

The third set of experiments is designed to evaluate the thermal comfort of occupants exposed to radiant system cooling at an operative temperature of 26 °C and benchmarked against an air cooling system with an air temperature setpoint of 26 °C.

2.1. Equipment and measurement instruments

The heating, ventilation, and air conditioning (HVAC) system conditions a ~180 m^2 floor area and is intended to accommodate 24 occupants in a desk-sharing workspace. The HVAC system offers two configurations to condition the space and provide ventilation. The first configuration consists of a DOAS combined with four fan coil units (FCUs), recirculating and cooling air in the space (Figure 2a). DOAS system is designed to provide ventilation needs and dehumidify outdoor air sufficiently to handle the latent load in the indoor environment. The configuration, which consists of DOAS and four FCUs, will be referred to as the air system. The air system is controlled based on the air temperature sensors installed on the walls. The second HVAC configuration consists of DOAS and fast-response radiant panels. The radiant panels are designed to provide sensible cooling, covering ~50% of the ceiling. The second configuration will be called the radiant system (Figure 2b). The radiant system is divided into 4 zones identical to the air system. The radiant system is operated based on the calculated operative temperature accounting for air temperature and solar temperature measured with the sensors situated near the window, ceiling, and wall surface.

DOAS (Figure 2c) has two stages. Initially, in the pre-cooling phase, outdoor air is in contact with a cooling coil that cools and dehumidifies the air. The cooling coil is supplied with chilled water at 10 °C. This precooled air then progresses to the second stage, passing through a heat and mass exchanger. During the second stage, outdoor air is cooled and dehumidified to a humidity ratio of 6.5 g/kg, before being released into the room. The outdoor air distribution is designed so that air first enters the plenum space above the radiant panels, allowing additional convective cooling as it flows between the panels' gaps entering the space (Figure 1c). Originally, the DOAS system was designed to operate based on the CO_2 signal and modulate the flow rate based on the demand, but due to the onset of the Coronavirus disease 2019 (COVID-19) pandemic in January 2020, a safety rule was adopted that the DOAS system operates at maximum capacity supplying 236 L/s (850 m^3/h, 1.8 air changes per hour) effectively converting on-demand supply to a constant air volume system.

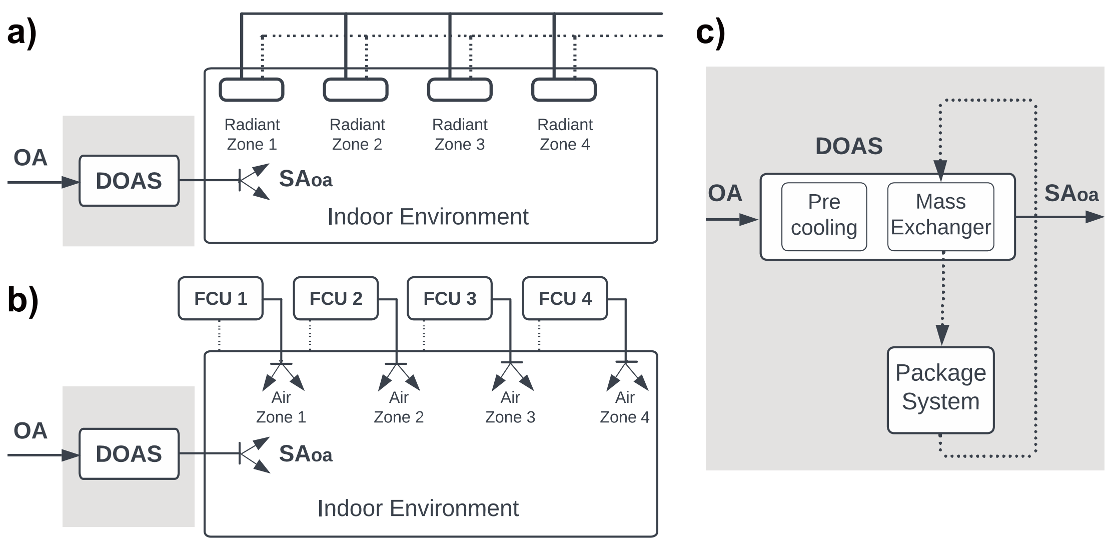

Figure 2. Schematics of a) air system, b) radiant system, and c) DOAS.

2.2. Energy Consumption Measurement Equipment

The pipe temperature sensor (Azbil, Model TY783) has a sensing accuracy of ± (0.15 + 0.002 × measured temperature) °C with a sensing range of -50 to 200 °C. It has a time constant of 50 s, which is shorter than the data collection interval of 5 min. The electromagnetic digital flow sensor (Keyence, FD-M100AYP) provides a sensing accuracy of ± 0.8% of the measured flow rate. The flow sensor can operate in the temperature ranging from 0 to 85 °C. The duct temperature sensor (Azbil, Model HTY78X3) measures temperature in the range of -20 to 60 °C with an accuracy of ± 0.3 °C and has a time constant of less than 1 min. We used the EBTRON thermal dispersion airflow measurement system (EBTRON Inc, SC, US). The measurements ensured that the upstream and downstream straight section lengths were at least 7.5 and 3 times the duct diameter, respectively.

ThermCondSys 5500 (Sensor Electronic, Poland) is a measurement system that simultaneously measures air temperature (t_a), globe temperature (t_g), airspeed (V_a), barometric pressure (p), and relative humidity (RH).

Shielded PT-100 probe is used for the air temperature with an accuracy of ± 0.1 °C with a measurement range of -10 °C to 50 °C. t_g is also measured with the PT-100 probe since it is a measurement of the air temperature inside the black sphere with a diameter of 150 mm. This measurement needs to be corrected to represent MRT ($\overline{t_r}$). We applied mixed convection correction that takes into account both free and forced convection in the heat transfer [35]. Airspeed is measured with an omnidirectional spherical probe that converts the quantity of heat convected from the probe's tip into a speed. The measurement range of the omnidirectional probe is 0.05 to 5 m/s with an accuracy of ± 0.02 m/s or ± 1% of readings, whichever is larger. The highest measurement frequency is 1 Hz. Relative Humidity is measured with a capacitive polymer sensing element and compensation bandgap temperature sensor. The measurement range is 0 to 100% RH, with the measurement uncertainty of ± 2% in the 10 to 90% range.

During the experiments, environmental measurements were taken with ThermCondSys 5500 positioned as depicted in Figure 3 with probes mounted at 4 heights: 0.1, 0.6, 1.1, and 1.7 m from the ground. Measurement frequency was 1 min.

high.RES sensor (CHAOSense, USA) measures MRT ($\overline{t_{r,high.RES}}$) by using six array-based thermopile infrared sensors to capture a ~6000 pixel panoramic image in the longwave infrared 5.5 to 50 µm every 30 seconds. Sensor accuracy of the total Radiance expressed in equivalent blackbody temperatures is ± 0.3 °C within the 22 to 45 °C range and extends to ± 0.5 °C from 0 to 50 °C, mostly irrespective of material average and spectral emissivity variances due to the nature of the thermopile detector. MRT measurements in this study were between 22 and 30 °C, therefore, we consider $\overline{t_{r,high.RES}}$ accuracy of 0.3 °C. We expect to see a difference in the radiant environment between the two cooling modes reflected through a separation of MRT and air temperature in space.

Internal sensible thermal loads were simulated using (i) light bulbs, (ii) radiators, and (iii) heating mats to emulate the heat release by a varying number of occupants through both convection and radiation. Each of the 15 light bulbs was mounted on the lamp post, releasing 40 W of heat through a combination of radiation and convection (Figure 3). Two radiators (OFRC15, Dimplex, UK) with a thermal output of 1050 W were placed in the room (Figure 3). Radiators have a surface temperature of 60 °C. Radiators are considered to release 80% of heat via radiation. Three radiant heating mats with 165 W each at the surface temperature of 55 to 70 °C (Figure 3). **Internal latent thermal loads** were simulated using ultrasonic cold mist humidifiers. Each of the humidifier emitted 0.81 kg/h of water into the air (Figure 3).

2.3. MRT measurements and impact of shading

To understand radiant systems performance, it's vital to distinguish between the air temperature and the MRT in a given space. Teitelbaum et al. [30] illustrated that these differences can be significant in certain radiant system applications. Previous research also indicates that MRT measurements using 150 mm black globes or small ping pong ball-sized black globes can yield imprecise results [37]. Given the importance of MRT measurements, we first conducted a set of experiments to verify the performance of different sensors. We compared the MRT measurement of the high.RES sensor and ThermCondSys 5500 with 150 mm black globes. Due to the narrow calibration, wide spectral response engendering an insensitivity to spectral emissivity errors, and the direct thermal measurement nature of thermopile detectors, we considered the MRT measured by high.RES sensor ($\overline{t_{r,high.RES}}$) as ground truth and compared corrected black globe MRT measurements. We evaluated the performance of these sensors with and without interior blind shading. These experiments were performed for a single thermal load of 2640 W or 15 W/m^2 consisting of 15 light bulbs (40 W each), a convective radiator (1050 W), and three radiant panels (130 W each, 390 W total). The layout of these internal loads is illustrated in Figure 3. This set of experiments enables us to select the measurement method in consecutive experiments.

Beyond evaluating sensor precision, we aimed to understand the potential benefits of controlling longwave infrared radiation through windows and its subsequent influence on MRT. A large contributor to the thermal environment in the space is a floor-to-ceiling glass wall on the west side that had external shading and blinds installed on the interior. We conducted measurements under two conditions: with and without deploying interior blinds. Besides comparing sensor accuracy, we analyze the effect of interior blinds on the thermal load distribution. Consequently, we undertook a comprehensive examination of (i) the thermal implications of both shaded and unshaded windows and (ii) the influence of the chilled panel's surface temperature on the room's MRT distribution. We experimented with 12 and 18 °C chilled water supply temperatures for the radiant panels. The measurement and thermal load layout are outlined in Figure 3b, which includes one measurement point for the high.RES sensor. One of the ThermCondSys 5500 was stationed at the room's center, next to the high.RES sensor.

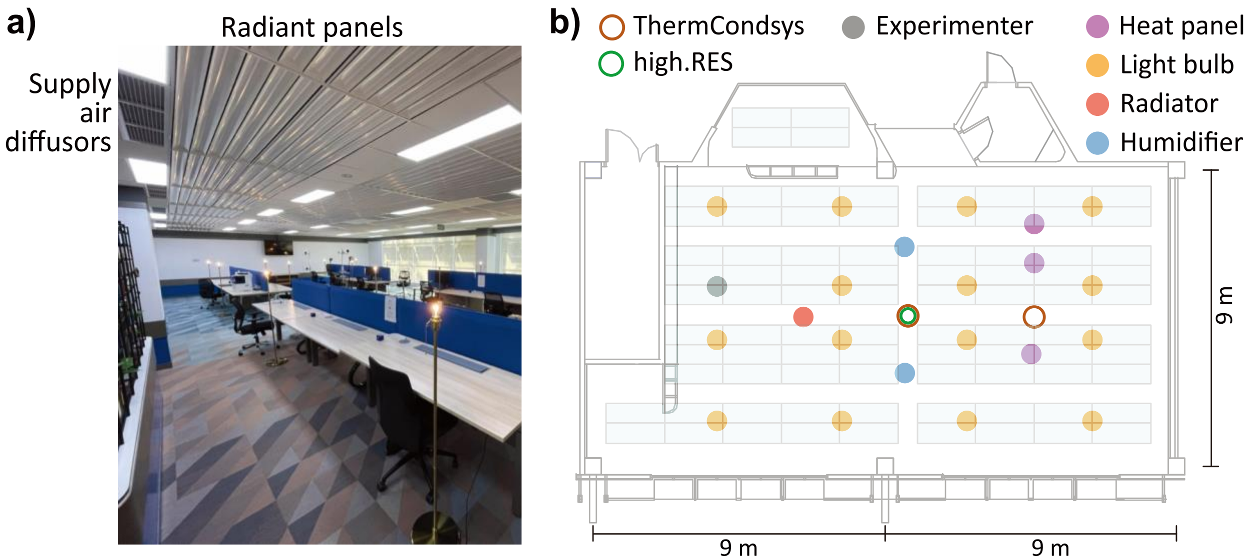

Figure 3. a) the photo of the space and b) the distribution of sensors and loads in the spaces with a thermal load of 2640 W. The top of the figure is east, and the windows are oriented west (bottom of the figure). The 86 radiant metal panels are shown in light blue.

2.3. Environmental Thermal Performance, Heat Flux, Electrical Power Consumption

DOAS operated at the full capacity of 236 L/s (850 m^3/h, 1.8 air changes per hour). We undertook a comprehensive analysis of the energy consumption across the air and radiant system, providing sensible cooling and ventilation. Internal load variations—comprising both sensible and latent load changes emulated conditions from full to partial loads, mirroring the flux in occupancy. Given that an individual typically releases ~100 W (ISO 8996 [41]) of sensible heat, our simulations considered a full occupancy of 24 individuals, complemented by the sensible load of other electronic devices within the space. For our *full load* scenario, the parameters set included a sensible load of 2700 W (by 2 radiators and 15 light bulbs) and a 1.6 kg/h (2 humidifiers) release of water vapor. To represent partial occupancy, we simulated sensible heat loads of 2100 W (18 occupants or 75% occupancy, by 2 radiators), 1450 W (12 occupants or 50% occupancy, by 1 radiator and 10 light bulbs), and 990 W (8 occupants or 33% occupancy, by 15 light bulbs and 3 radiant heating mats). We used 23 °C, a typical room temperature setpoint in Singapore [42,43], for the air system as the benchmark representing the most prevalent local operating condition. Additionally, we compared the air and radiant systems at the same setpoint of 26 °C, which is believed to offer better thermal comfort. As the outdoor climate influenced the external load—an aspect beyond our control—we replicated each experimental scenario 3 times to account for varied external load conditions. A summary of experimental conditions can be found in Table 1.

Table 1. Experimental conditions

System Type	Sensible Load [W]	Latent Load [kg/h]	Setpoint conditions
Air	990, 1450, 2100, 2700	0.8, 1.6	26 °C, 23 °C
Radiant	990, 1450, 2100, 2700	0.8, 1.6	26 °C

Based on the results from the first set of experiments in which we characterized the thermal environment, we strategically placed ThermCondSys 5500 measurement trees at two locations (Figure 3 and Figure A1). Alongside these metrics, we also gathered external environmental data from the nearest weather station and recorded readings from surface sensors implemented for control (Figure 3).

2.4. Heat Flux Analysis

We evaluated and compared the sensible heat removed from the space by the radiant and air systems to maintain the desired setpoint conditions. We compared the heat flux of the air system operating at 23 and 26 °C and the radiant system at 26 °C operative temperature. The removal of heat by FCUs and radiant panels is measured by a combination of water temperature measurements and water flow rate measurements. For FCUs, measurement takes place at each FCU, and for radiant zones, multiple panels connected in parallel consist of a zone where measurement takes place.

2.5. Electrical Power Consumption Analysis

This analysis focuses on comparing the total electrical power consumption between the air system at two different operating temperatures (23 and 26 °C) and the radiant system set at 26 °C operative temperature. As previously detailed in the HVAC section, the air system's HVAC configuration includes a DOAS and four FCUs. The DOAS features a pre-cooling stage and a heat and mass exchanger stage. We calculated the pre-cooling heat exchange using actual chilled water flow rate measurements and the temperature differential across the pre-cooling coil (Q_{DOAS} in *W*). This chilled water is sourced from a high-efficiency chiller with an average coefficient of performance (COP) of 6. The package system conditioned one air stream in the crossflow heat and mass exchanger that conditioned outdoor air supplied as ventilation. Power consumption, which includes the packaged unit ($P_{DOAS,package}$ in *W*) and the power drawn by the fan ($P_{DOAS,fan}$ in *W*) to condition the outdoor air supplied to the space, is monitored using electrical meters on each device's circuits. Thus, the DOAS's total electrical power consumption ($P_{DOAS,total}$ in *W*) can be expressed by Eq. (1).

$$P_{DOAS,total} = P_{DOAS,fan} + P_{DOAS,package\ unit} + \frac{Q_{DOAS}}{6} \tag{1}$$

The total electrical power consumed by the four FCUs comprises the power to transport and air conditioning. The power to transport includes the power for the pump delivering chilled water to the coils ($P_{FCU,pump}$ in *W*) and the fan circulating room air through the FCU ($P_{FCU,fan}$ in *W*). Power meters on their respective electrical circuits measure both components' power consumption. The total electrical power consumed for air conditioning is the power needed to cool the water provided by the chiller (Q_{FCU}^{i} in *W*, *i* represents the zone number of 1, 2, ,3 or 4). The COP of the chiller is given as an average of 6. Hence, the electrical power consumption for the FCUs ($P_{FCU,total}$ in *W*) is summarized as:

$$P_{FCU,total} = P_{FCU,fan} + P_{FCU,pump} + \frac{\sum_{i=1}^{4} Q_{FCU}^{i}}{6} \tag{2}$$

Lastly, the power consumption of the radiant panels includes the power needed for chilled water circulation ($P_{Radiant,pump}$ in *W*) and the power to deliver the cooling energy ($Q_{Radiant}^{i}$ in *W*, *i* represents the zone number of 1, 2, ,3 or 4). Seshadri et al. [44] measured COP of 8 in a high-efficiency low-lift vapor-compression chiller for high-temperature cooling applications. Application of this chiller for sensible cooling can further enhance energy savings, considering that latent load and outdoor air processing are decoupled. We calculated the total electrical power for the radiant panels for the standard high-efficiency chiller using COP of 6 and the low-lift chiller using COP of 8 as follows:

$$P_{Radiant,total}^{standard} = P_{Radiant,pump} + \frac{\sum_{i=1}^{4} Q_{Radiant}^{i}}{6} \quad (3)$$

$$P_{Radiant,total}^{low-lift} = P_{Radiant,pump} + \frac{\sum_{i=1}^{4} Q_{Radiant}^{i}}{8} \quad (4)$$

2.6. Thermal Comfort Evaluation

The third set of experiments was designed to assess occupants' thermal environment perception, satisfaction, and preference when exposed to two different HVAC systems: the radiant and the air systems. Both systems operated at 26 °C, for the radiant system based on operative temperature and for the air system based on air temperature. A total of 78 participants took part in this experiment. We employed a crossover study design, where participants were exposed to both HVAC systems in different sequences. Participants were divided into 5 groups of ~16 members each (Figure A2). Each group experienced conditions set by the two HVAC systems. Every participant was exposed to one system for 90 minutes, followed by a 60-minute break, and then another 90 minutes of exposure to the other system (Figure 4). Depending on their assigned group and the sequence of exposures, a participant's first exposure might be to the air system and the second to the radiant system, or vice versa. Gender and age demographics distribution is depicted in Figure A4 in the Appendix. Slightly more females participated in the experiment, and most of the participants were between 18 and 29 years old (Figure A4 in the Appendix).

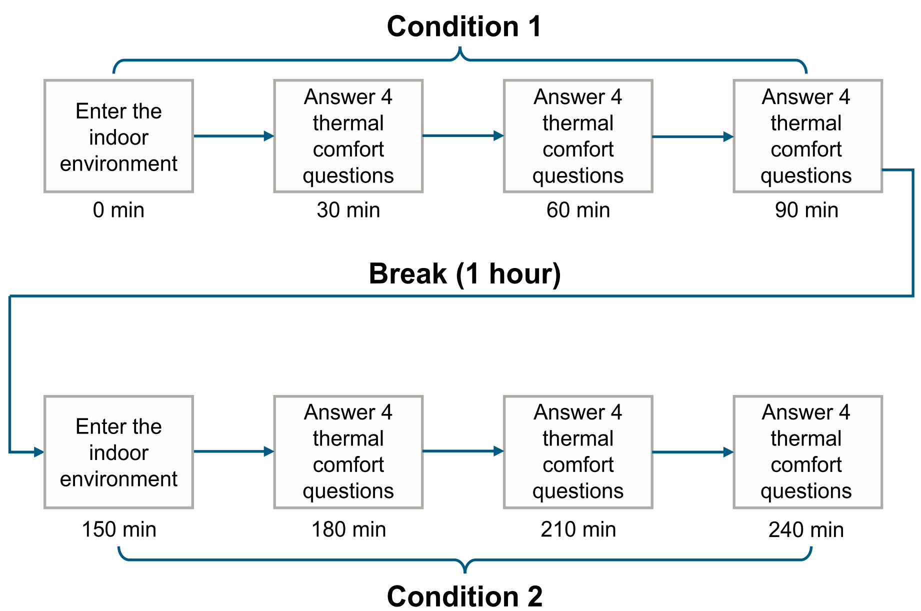

Figure 4. Schematics of the survey sequence during one group of participants.

Upon entering the space, participants initially provided basic demographic information. Subsequently, they responded to four thermal comfort questions asked at 30, 60, and 90 min after exposure started. Participants were surveyed six times in total (three times per HVAC system condition). Schematics of the survey sequence during one group of participants is depicted in Figure 4. Questions and scales are depicted in Figure A3 and follow ASHRAE 55 2023 recommendations. All survey responses were gathered using the Qualtrics platform.

In the ASHRAE 55-2023 thermal comfort satisfaction scale [45], the categories “slightly satisfied”, “satisfied”, and “very satisfied” are considered positive, reflecting an acceptable thermal state. These points indicate that the occupants feel thermal comfort to varying positive degrees. The answer “neutral” was considered a separate category. For the analysis of thermal perception, responses indicating cooler sensations (“cold”, “cool,” and "slightly cool") were grouped under the "cooler side" category. "slightly warm," "warm," and "hot" responses were categorized as the "warmer side". We kept "neutral" as a separate category.

2.7. Thermal comfort statistical analysis

Upon data collection, the raw survey responses were processed using *Python 3.9* programming language. *Pandas 2.1.2* library was employed for data manipulation. The Wilcoxon signed-rank test was employed to analyze the data. This non-parametric statistical test is suitable for assessing differences between two related samples, matched pairs, or repeated measurements on a single sample and does not require the assumption of normally distributed differences between pairs. Our sample was ordinal and paired because the same group of participants was exposed to both tested conditions. Our study used this test to compare thermal satisfaction, preference, sensation, and preference for air movement between the air and radiant cooling systems. Each participant's scores/data points under the two conditions were paired, and the differences were calculated. The Wilcoxon test then ranked these differences, considering their absolute values. The test statistic, denoted as W, is the sum of the ranks of the differences with positive signs. This statistic was used to determine whether the differences between the paired observations were statistically significant. A significance level of $\alpha = 0.03$ was set for the analysis. For example, in comparing radiant and air conditions for the question "How do you feel right now?", we referred to the number of answers provided in Table A1 in the Appendix. The hypothesis was Null hypothesis (H0): the effect of condition A is the same as that of condition B.

Besides the Wilcoxon signed-rank test, we employed Cliff's δ as a non-parametric effect size measure to assess the magnitude and direction of the differences between the two independent groups [46]. This measure is particularly suited for ordinal data or non-normally distributed interval data, which aligns with the nature of our data set. Cliff's δ calculates the probability that a randomly selected score from one group will be greater than a randomly selected score from another group, and vice versa. The value of Cliff's δ ranges from -1 to +1. A value of zero indicates no effect or similarity between the groups. Positive values suggest a greater likelihood of higher scores in the first group, while negative values indicate higher scores in the second group. The calculation of Cliff's δ involves comparing each pair of observations across the groups and counting the number of times scores in one group are greater than, equal to, or less than scores in the other group. This count is then normalized to fall within the -1 to +1 range. There is no strong consensus about the effect size values, but the interpretation of Cliff's δ could be as follows: (i) $\delta = 0$: No difference between the two groups, (ii) $0.147 < |\delta| \leq 0.33$: small effect, (iii) $0.33 < |\delta| \leq 0.474$: medium effect, (iv) $|\delta| > 0.474$: large effect. In our study, Cliff's δ was utilized to evaluate the difference in thermal satisfaction, preference, sensation and air movement preference between air and radiant system cooling. This provided a robust measure of the effect size that is less sensitive to outliers and does not require the assumption of normality. The interpretation of Cliff's δ helped us to understand not only if there were differences between the groups but also the extent and direction of these differences.

3. Results and Discussion

3.1. MRT measurements and impact of shading

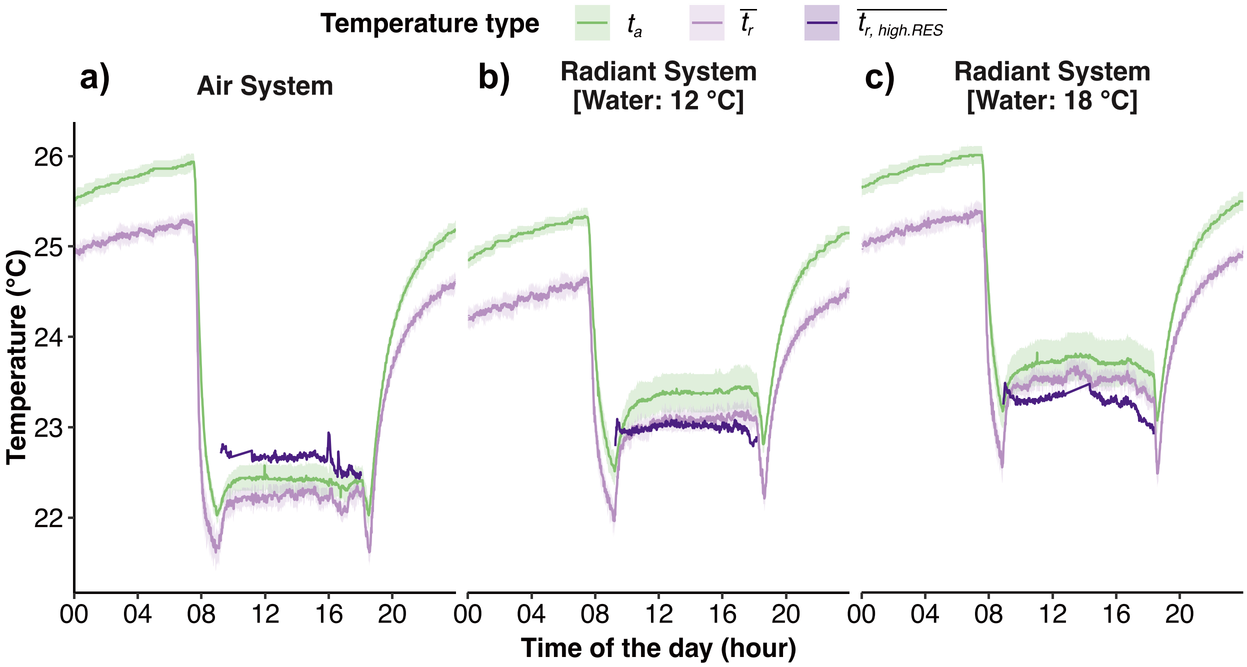

Figure 5. Measurements with air temperature and MRT using high.RES sensor and ThermCondSys black globes with the mixed convection correction under a) the air system, b) the radiant system supplied with 12 °C chilled water, and c) the radiant system supplied with 18 °C chilled water.

Figure 5 shows that MRT measured with different techniques follows a similar trend whenever the space was cooled by the air or the radiant system with different supplied chilled water temperatures. MRT measured with ThermCondSys black globes consistently registered below the air temperature under three different HVAC conditions. By contrast, MRT measured by high.RES was slightly higher than air temperature when the space was conditioned by the air system.

Results in Figure 6 show air to MRT difference measured with two sensors. Measurements with high.RES sensor showed air to MRT difference of generally <0.5 °C. While the high.RES sensor is able to pick up a statistically significant difference between test conditions, the effect is small. The 150 mm black globe sensors do not exhibit any statistically significant differences between test conditions, and a comparison between measurements depicted in Figure 6 also indicates that the potential error of the black globe sensors was within the observed air and mean radiant temperature difference of 0.5 °C. Hence, we can conclude that since the difference between the air and mean radiant temperatures is small in these test conditions, it may be acceptable to use either the 150 mm black globes or the high.RES sensors to measure MRT during the experiments as the error in the black globes will be roughly bounded by the air temperature to MRT separation.

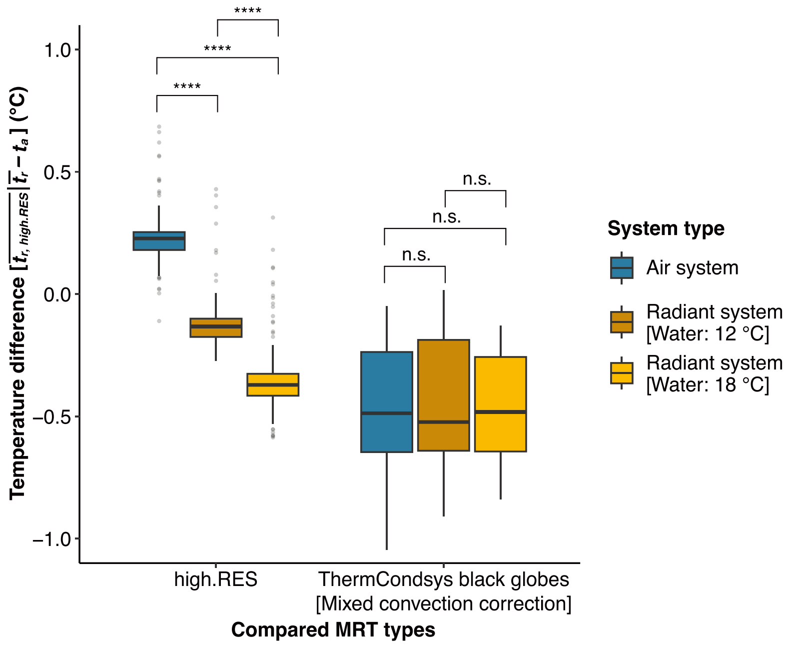

Figure 6. Air temperature and MRT difference for high.RES and black globe measurements.

In Singapore, cloud coverage is ~60% of the time [47]. This indicates that ~60% of the sky is typically covered by clouds, which can significantly affect the amount of solar radiation reaching the ground, influencing both direct and diffuse solar radiation components. When the sky is clear and the sun is high, about 85% of solar radiation is direct, and 15% is diffuse; however, during cloudy days, the amount of direct solar radiation decreases significantly while the diffuse component increases. Because of the significant impact of diffuse solar radiation, it is important to understand its impact on the indoor environment. Results in Figure 7 show a comparison between measurements with blinds open and closed. They indicate no statistically significant difference in air temperature to MRT difference when blinds were open and closed. We decided to keep blinds closed in all subsequent measurements.

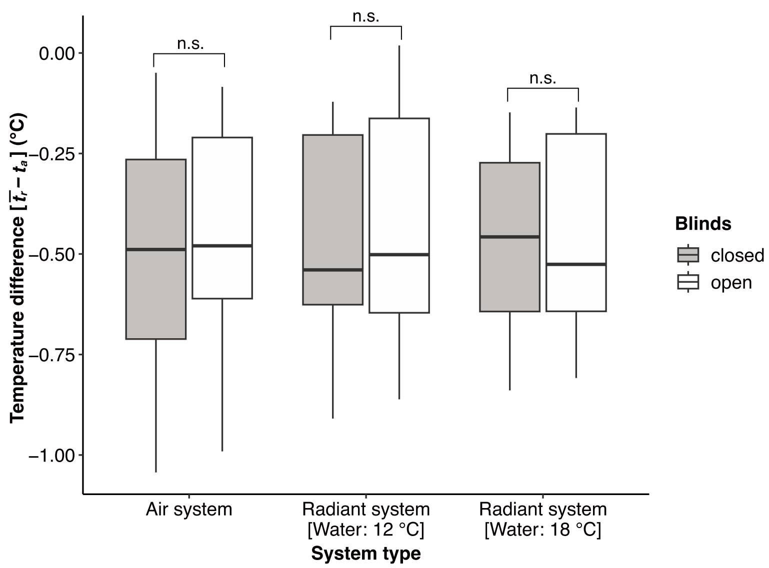

Figure 7. Air temperature and MRT difference when blinds are open (white) and closed (grey) for all experimental conditions.

Results depicted in Figure 5, Figure 6, and Figure 7 show the difference between air and mean radiant temperatures is below 0.5 °C. This confirms conclusions from previous studies that in high-performing buildings measuring air temperature can be sufficient to practically estimate thermal comfort [34]. We may conclude that if the building's location, climate, and radiant panel design yield a bounded system where MRT cannot deviate from the air temperature by significantly more than 0.5 °C, measuring the MRT is unnecessary.

We can also extend this conclusion to controls and suggest that measuring air temperature to drive control by thermal comfort is sufficient to reasonably encapsulate the impact of all other thermal indicators. We limit our conclusion to our case, which was a high-performing building (BCA ZEBPlus) in Singapore where the difference between indoor and outdoor air temperature was ~6 °C with limited direct solar radiation. We also need to emphasize that we used proper ground truth methods to understand air and mean radiant temperature differences. Control suggestions made in our study differ from Oxizidis and Papadopoulos [48] that suggested that radiant cooling systems should be controlled with operative temperature and not the indoor air temperature. More research is necessary to create a full set of criteria that can be practically implemented to control buildings with radiant and air systems considering ambient climate conditions (solar radiation and air temperature), indoor conditions, and facade properties.

3.2. Thermal performance, heat flux, electrical power consumption

Environmental thermal performance

The results indicate that for both the air and the radiant system, the air temperature and MRT were a little bit above the setpoint temperature of 23 and 26 °C (Figure 8). We can also observe that the radiant system was closer to the setpoint than the air system. Data on air velocity revealed that the radiant system maintained a median airspeed of approximately 0.06 m/s, which falls below the minimum velocity of 0.1 m/s as stipulated by the Singaporean Standard 554 [49] and can be considered still air.

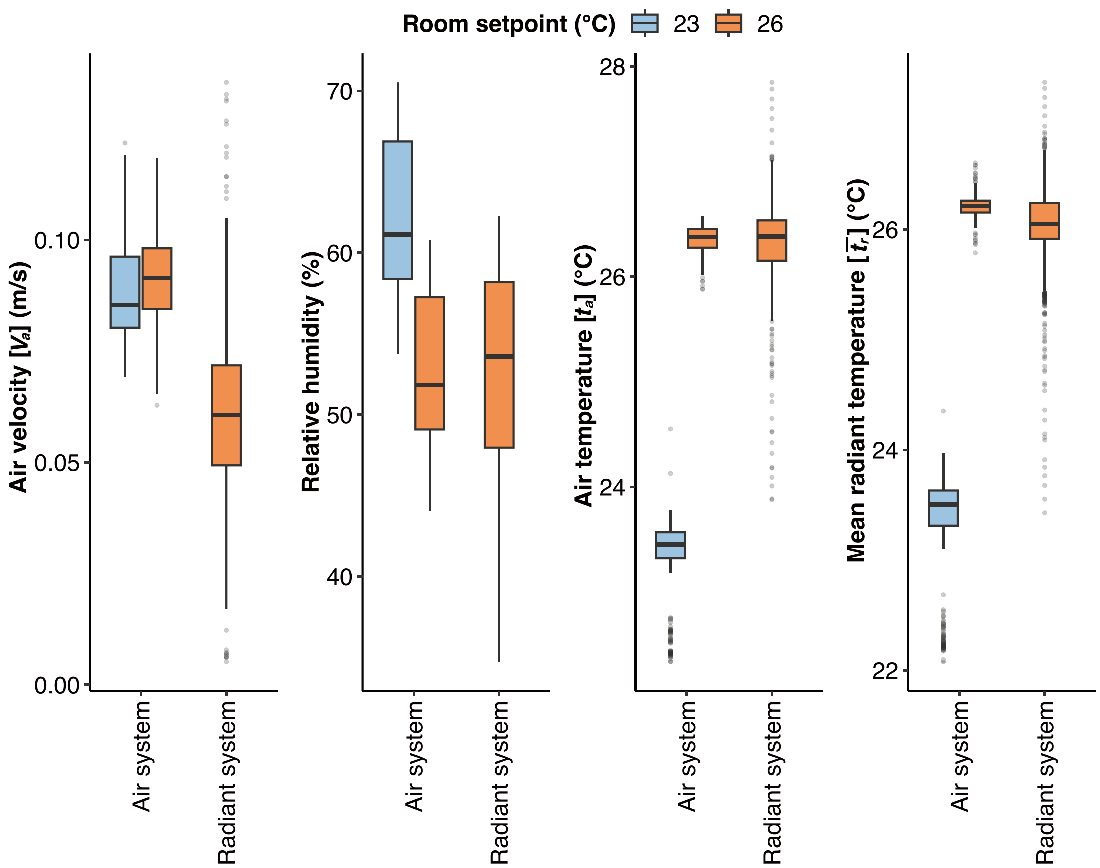

Figure 8. Measured air velocity, RH, air temperature, and MRT during all experimental runs.

The average temporal performance depicted in Figure 9 reveals that during the system's off phase, the room cooled by the air system starts from a 0.5 °C higher air temperature and MRT than the radiant system. This is an indication that the radiant system, while operating, removes more energy (higher heat extraction rate) than the air system, resulting in lower starting temperatures in the morning. This was theoretically and experimentally demonstrated in previous research [16,17,50]. In both cases, when the system is off, the air temperature is, on average, 1 °C higher than the MRT; this difference practically disappears when the system is in operation. Both systems were able to maintain a semi-steady state around the setpoint temperature during the day.

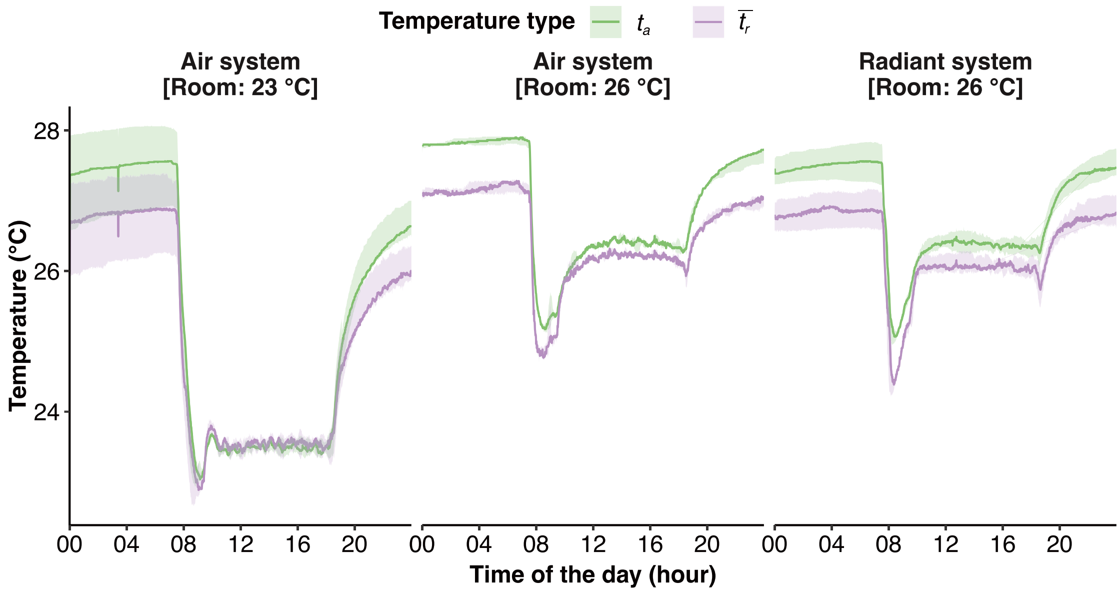

Figure 9. Average temporal performance of air and radiant systems.

<u>Heat flux and electrical power consumption</u>

Figure 10a presents data indicating that when the air system operates at 23 °C, it removes 8.3 kW of heat from the space by the median. However, at an operating temperature of 26 °C, the system extracts around 3.9 kW, marking a substantial decrease. This significant difference in heat extraction capacity underscores the system's sensitivity to temperature settings. When comparing the performance of the air system and the radiant system, both set at 26 °C, we find that the radiant system removes about 5.2 kW of heat from the space, which is 33% more than the air system at the same air temperature setting. These findings corroborate the experimental results previously reported by Feng et al. [16] and Woolley et al. [17]. Notably, the results depicted in Figure 10a are among the first in-situ measurements from an actual building that validate trends previously observed in laboratory settings [16,17]. Figure 10a also demonstrates that the DOAS is required to remove close to 5 kW of heat, suggesting that during operation at 26 °C, the heat flux necessary for thermal conditioning and dehumidification of outdoor air is proportional to the sensible heat removed by the cooling system. Additional data in Figure A5, located in the Appendix, reveal that both the air and radiant systems exhibit variability in the heat fluxes extracted across four zones within the building, with differences of up to 27% compared with the mean value of the panels. This variability is an important consideration in the design and operation of HVAC systems for maintaining consistent thermal comfort throughout the space.

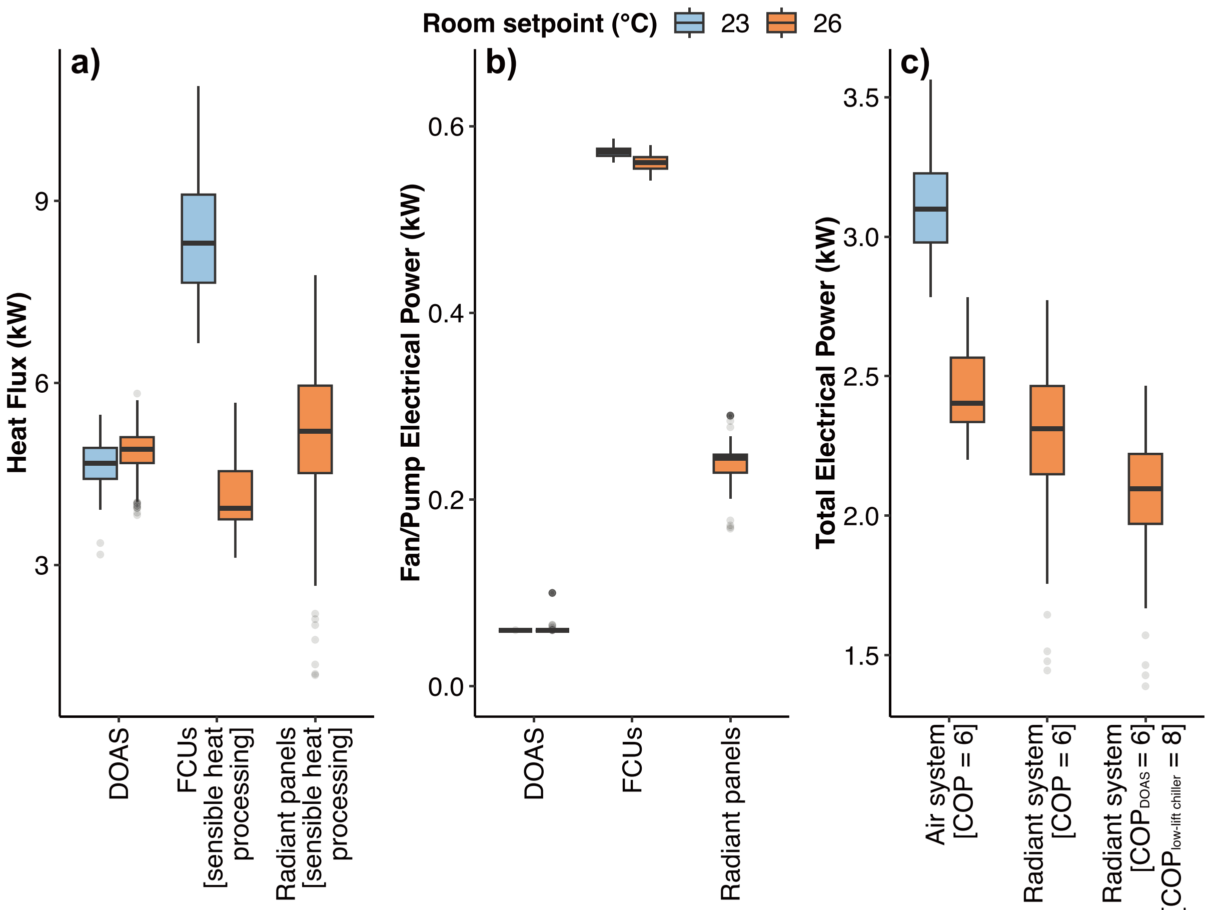

Figure 10. a) Heat extracted from the room by DOAS, sensible heat extracted by FCUs, and sensible heat extracted by radiant panels; b) Electrical power is necessary to operate fans and pumps; and c) total electrical power consumed by the two systems. The room was set to air temperatures of 23 and 26 °C for the air system and an operative temperature of 26 °C for the radiant system. When calculating the total electrical power, a COP of 6 was used for both air and radiant systems. In addition, a COP of 8 was used for the chiller water supplied to the radiant panels for the radiant system.

Results in Figure 10 (b) show that the air system requires 0.57 kW of electrical power to transport necessary chilled water and air, while the radiant system requires 0.24 kW of electrical power for pumps and fans. This shows, as suggested by Rhee et al., that a radiant system can reduce the energy required to transport the cooling medium [2]. In the current comparison, the air system requires two times the power for transporting the fluids around the system.

Results depicted in Figure 10 (c) show that the radiant system delivers similar physical thermal conditions as the air system but consumes 4% less electrical energy when both systems are operated at 26 °C operative and air temperature. The electrical energy savings of the radiant system are 34% compared to the air system operating at 23 °C. This electrical energy consumption is based on the chilled water supply from the same chiller system with a COP of 6. When a low-lift chiller with COP = 8 is used for conditioning chilled water used by radiant panels, we can see that energy savings compared to the air system operated at the same temperature are 15%. Decoupling of sensible and latent cooling also opens up a possibility for the deployment of efficient dehumidification methods like liquid desiccant [51].

3.3. Thermal Comfort Evaluation

Environmental Measurements

The data presented in Figure 11 illustrates the thermal conditions recorded during the operation of air and radiant heating systems during the subjective evaluation. The median air temperature was approximately 26.1 °C, with no notable difference observed between the two systems. RH was consistent at around 43%, again showing no statistically significant variation between the systems. The mean radiant temperature remained below 26 °C for experimental runs with both systems. However, air velocity did show a statistically significant difference; the median during air cooling system operation was 0.12 m/s, compared to less than 0.07 m/s when the radiant cooling system was active, but the differences of these air speeds may not be detected by a person, both speeds could be considered still air. Overall, the thermal environments provided by both systems were quite similar.

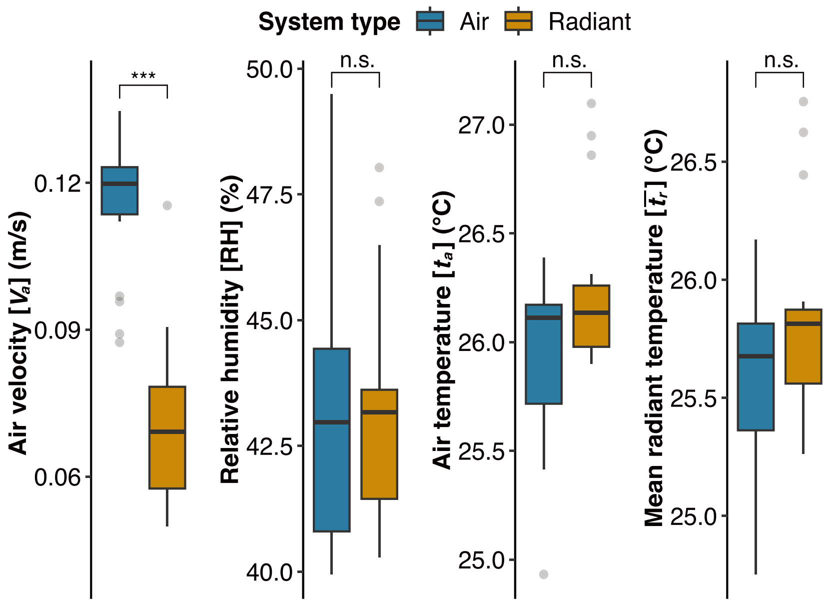

Figure 11. Environmental measurements during the thermal comfort experiments.

Thermal Satisfaction

Ninety minutes into the experiment, as shown in Figure 12 (a), participant feedback indicated that 76% were satisfied, 17% neutral, and 7% dissatisfied with the thermal environment under the air system operation. Similarly, during the radiant system operation, after the same duration, the satisfaction level stood at 62% (21% were neutral and 17% were dissatisfied). The Wilcoxon signed-rank test comparing the satisfaction distributions between the two systems indicated no statistically significant difference (α = 0.03, p = 0.04). We also applied Cliff's δ to assess the likelihood that randomly selected thermal satisfaction responses obtained during the operation of the radiant system would indicate higher thermal satisfaction compared to those collected during the operation of the air system. The results yielded a Cliff's δ value of -0.14. This value suggests a negligible effect size, implying that there is a very low probability that responses chosen randomly from the air system cooling phase would demonstrate higher thermal satisfaction compared to those from the radiant system cooling phase. This result implies that both air and radiant systems deliver comparable levels of thermal satisfaction among occupants. The mean thermal satisfaction vote for the air system was 1.45 with a standard deviation of 1.16, while that for the radiant system was 1.14 with a standard deviation of 1.36.

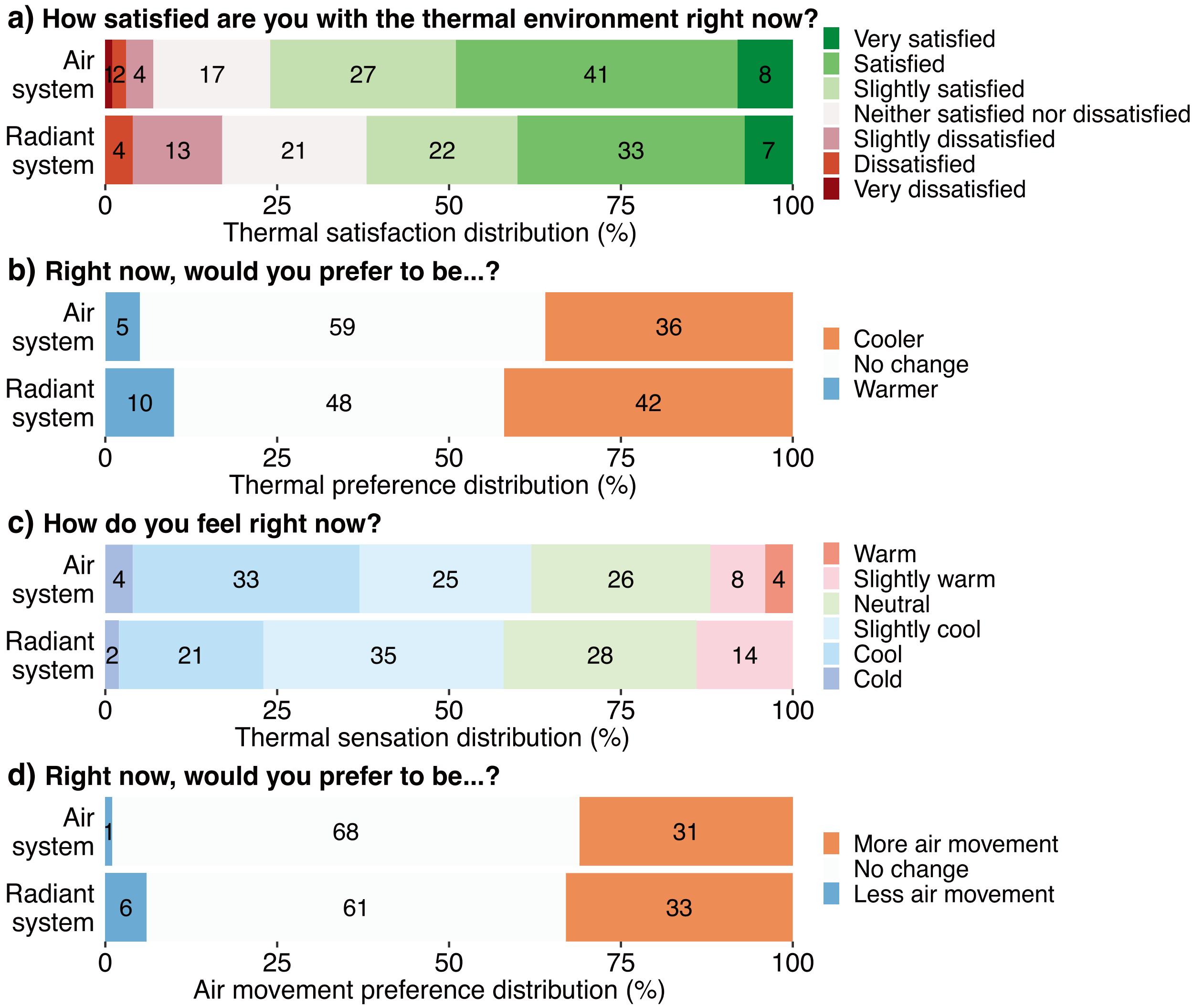

Figure 12. Subjective feedback of a) thermal satisfaction, b) thermal preference, c) thermal sensation, and d) air movement preference of participants captured 90 minutes after the experiment started for the air system and the radiant system. Numbers represent % of answers in each of the categories. 78 subjects for each system.

Thirty minutes after starting the experiment with the air system, 58% of participants expressed satisfaction with the thermal conditions, as shown in Figure A6. This satisfaction increased to 70% after sixty minutes (Figure A6) and reached 76% after ninety minutes (Figure 12a). Using the Wilcoxon signed-rank test, we found a statistically significant difference in satisfaction levels between thirty and ninety minutes ($\alpha = 0.03$, $p = 0.0006$), indicating that the satisfaction distributions varied over time at the beginning. Further, no significant difference was found between the sixty and ninety-minute marks ($\alpha = 0.03$, $p = 0.05$), implying that participants adapted and reached a state of thermal equilibrium within sixty minutes.

Thirty minutes into the experiment with the radiant system, 46% of participants reported satisfaction with the thermal environment, 17% were neutral, and 37% were dissatisfied (Figure A6). An hour into the experiment, the proportion of satisfied participants rose to 57% (Figure A6). By the ninety-minute mark, satisfaction levels had increased to 62% (Figure 12a). Statistical analysis revealed that there was no statistically significant difference between the satisfaction levels at sixty minutes and ninety minutes ($\alpha = 0.03$, $p = 0.89$). The thermal satisfaction trends for both the air and radiant systems indicate an improvement in comfort from the onset of the experiment. The data suggests that participants reached a state of thermal equilibrium approximately sixty minutes after the start of the experiment and maintained this comfort level thereafter.

Thermal Preference

Ninety minutes after the experiment started, 42% of participants preferred a cooler environment during radiant system cooling, and 36% preferred a cooler environment during air system cooling (Figure 12b). Applying the Wilcoxon signed-rank test, we can observe no statistically significant difference ($\alpha = 0.03$, $p = 0.24$) in thermal preference between air and radiant system cooling after 90 minutes into the experiment. To assess the effect size, we utilized Cliff's δ. The calculated value of δ was 0.46, indicating a medium-sized effect. This suggests a moderate probability that a randomly chosen participant who experienced the radiant cooling system would prefer a cooler environment compared to a randomly chosen participant who experienced air system cooling. When we consider in parallel thermal satisfaction results (Figure 12a) and thermal preference results (Figure 12b) this result suggests that we are approaching the upper limit of setpoint temperatures under which air and radiant cooling systems can deliver comfortable conditions to the occupants.

When an air system is operated, 35% (Figure 12b) of participants preferred a cooler environment 90 min after entering the room. Surveys conducted 30 min after the experiment started show that 57% (Figure A7) of participants preferred a cooler environment. The Wilcoxon signed-rank test comparison between 30 min and 90 min survey results suggests that distributions are not the same ($\alpha = 0.03$, $p = 0.02$), but the comparison between 60 and 90 min shows that distribution is the same ($\alpha = 0.03$, $p = 0.9$). These results suggest temporal progression of thermal preference. This shows that occupants were not in thermal equilibrium initially but have reached and maintained it.

When the radiant system was used for cooling, 90 min after the experiment started, 42% (Figure 12b) expressed a preference for the cooler environment. Similar to the trend displayed with the air system, the Wilcoxon signed-rank test shows that there is a statistically significant difference between results after 30 min, where 66% ($\alpha = 0.03$, $p = 0.008$) (Figure A7) of the participants preferred a cooler environment, but 60 min (Figure A7) distribution and 90 min distribution of thermal preferences are not statistically different ($\alpha = 0.03$, $p = 0.57$) suggesting thermal equilibrium.

Thermal Sensation

Under both radiant and air cooling conditions, 90 min after the experiment started, 28% (Figure 12c) and 26% (Figure 12c) of participants, respectively, reported feeling neutral. A majority indicated they felt cold to slightly cool, with 62% (Figure 12c) under radiant cooling and 58% (Figure 12c) under air cooling. The Wilcoxon signed-rank test revealed no significant difference in thermal perceptions between the radiant and air systems ($\alpha = 0.03$, $p = 0.18$), indicating that participants generally experienced similar thermal sensations with either cooling system.

When examining the progression of thermal sensation over time during air system operation, survey results at 30 minutes and 90 minutes from the experiment's start were statistically different ($\alpha = 0.03$, $p < 0.009$), suggesting a shift in perception from warmer to cooler conditions for the air system (Figure A8). A similar pattern was observed for the radiant system ($\alpha = 0.03$, $p < 0.01$). However, comparisons between surveys conducted at 60 minutes and 90 minutes did not reveal any statistically significant differences in thermal sensation for either the air ($\alpha = 0.03$, $p = 0.37$) or radiant systems ($\alpha = 0.03$, $p = 0.15$), suggesting that thermal sensations stabilized after the initial period. This has been observed across thermal satisfaction, preference, and sensation questions.

Upon reviewing the results for thermal sensation (Figure 10c) and thermal preference (Figure 10b), it is noted that 62% of participants found the environment slightly cool, cool, or cold during air system operation, and 58% reported the same during radiant system operation. However, only 5% preferred a warmer environment with the air system, and 10% with the radiant system. This is surprising because more participants would be expected to prefer a warmer environment based on their thermal perception. Yet, this finding aligns with previous studies in tropical climates, such as those by Buonocore et al. in Brazil [52], Guevara et al. in Ecuador [53], and Wong and Khoo in Singapore [54]. The consistency of these results suggests a phenomenon that warrants further investigation.

Air Movement Preference

In our study, 31% (Figure 12d) of participants preferred increased air movement while using the air system for cooling, and 33% (Figure 12d) expressed the same preference while using the radiant system. The Wilcoxon signed-rank test showed no statistical difference ($\alpha = 0.03$, $p = 0.53$) between the air and radiant systems. Cliff's δ value of 0.51 suggests a large size effect, indicating that there is a high probability that randomly chosen samples from each group are different. Research in Singapore has shown that people in hot and humid climates often want more air movement [55]. Providing fans to enable a higher room temperature setpoint can lead to major energy savings [39].

4. Conclusion

We experimentally assessed thermal environmental performance, energy consumption, and thermal comfort for an office environment conditioned with radiant or air systems in Singapore's tropical climate. Both systems were set at 26 °C, operative for the radiant system and air temperature for the air system. The energy performance was benchmarked against an air system set at the commonly applied 23 °C tropical setpoint. Utilizing a crossover study design for the subjective evaluation, we engaged 78 individuals distributed into five groups.

Environmental performance measurements revealed that both the air and radiant systems delivered very similar thermal conditions. The difference between MRT and air temperature was < 0.5 °C for both systems. This suggests that in real environments with radiant panels in direct contact with indoor air, significant differences between these two parameters are unlikely. Our findings also indicate that radiant systems can be effectively controlled using an air temperature sensor, especially in contexts like our study in Singapore, characterized by limited direct solar radiation, facade shading systems, and well-insulated buildings with low-E glass. However, the use of membrane-based radiant panels operating at low supply water temperatures may necessitate including MRT in control systems. The only environmental parameter with a significant numerical difference was the air velocity, but the practical implications of that difference are not impactful, and, in both cases, it is perceived as still air with more air movement desired.

Extracted heat flux was 33% higher for the radiant system than the air system. This is the first in-situ measurements from an actual building that validates trends previously observed in laboratory settings [16,17]. The practical implication of this real building confirmation of laboratory results is that thermal load calculations are dependent on the system implemented.

The electrical power necessary to transport the cooling medium and ventilation air to the indoor environment was smaller for the radiant system. The radiant system delivering very similar thermal conditions consumes 4% less electrical energy than the air system operated at 26 °C, or 34% less compared to the air system operated at 23 °C (the prevalent HVAC system and room temperature setpoint in Singapore).

Our findings indicate that there was no difference in thermal comfort satisfaction between the systems where over 60% of the participants expressed satisfaction with the thermal environment, ~20% voted neutral thermal satisfaction, and less than 17% were dissatisfied. ~40% of the participants expressed that they would prefer a cooler environment independent of the system when cooling systems were operated at 26 °C. ~30% of the participants preferred more air movement. Overall, our analysis suggests that both radiant and air systems can be effectively operated at 26 °C in Singapore's tropical climate, with tropically acclimatized people, providing similar levels of thermal comfort.

5. Acknowledgments

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Appendix

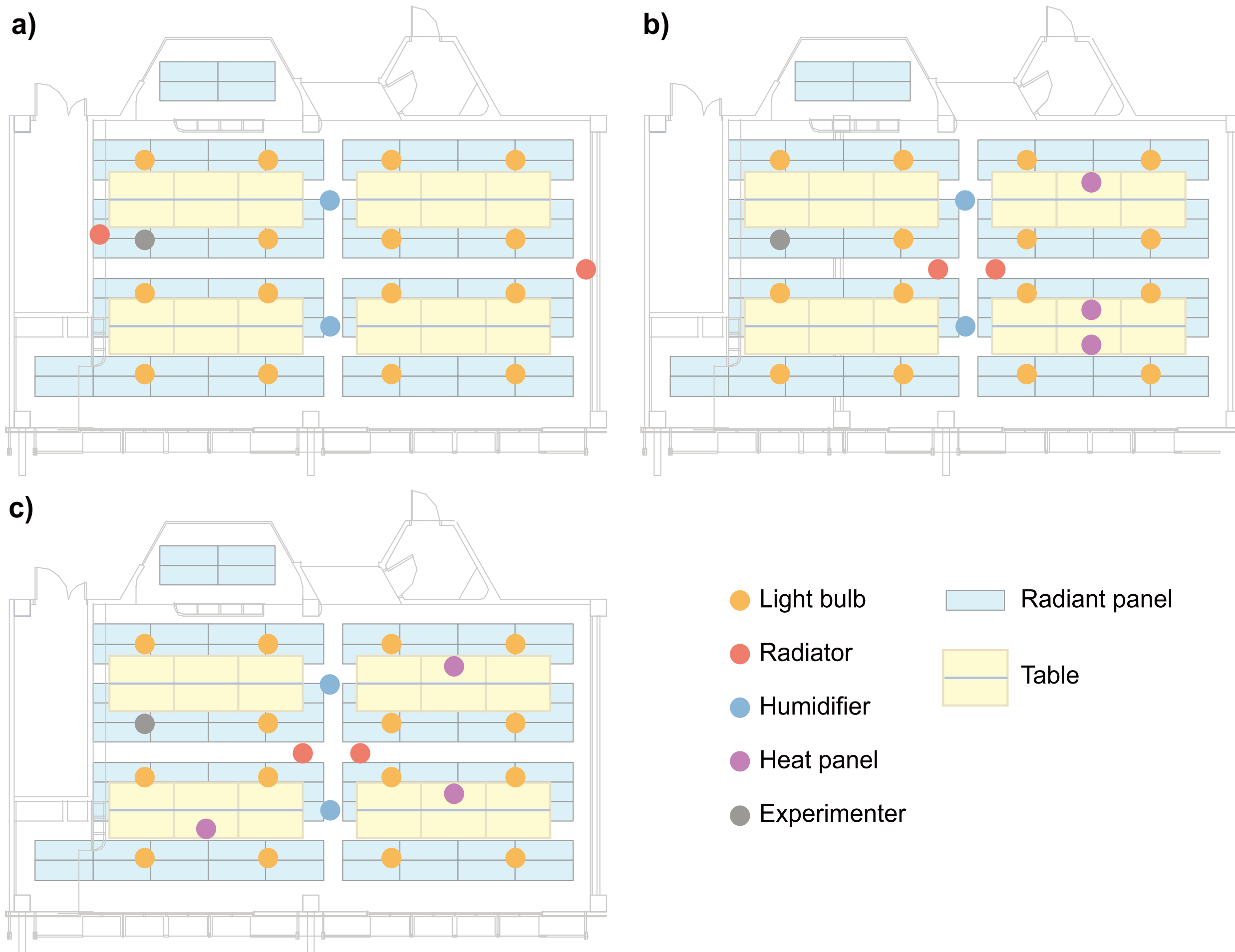

Figure A1. Load layouts: a) radiators near the wall; b) heat panels equally distributed; c) heat panels concentrated in a single zone.

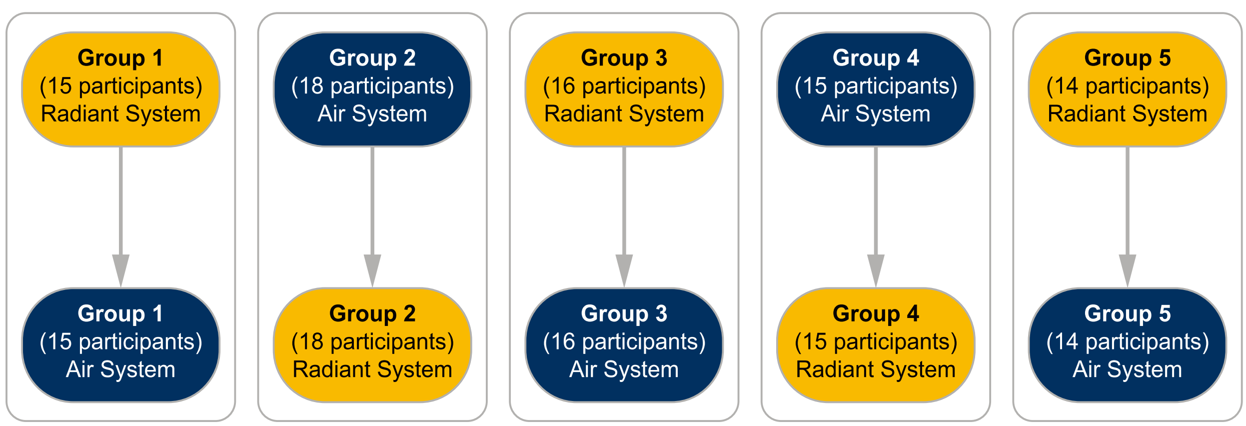

Figure A2. Schedule of crossover study design.

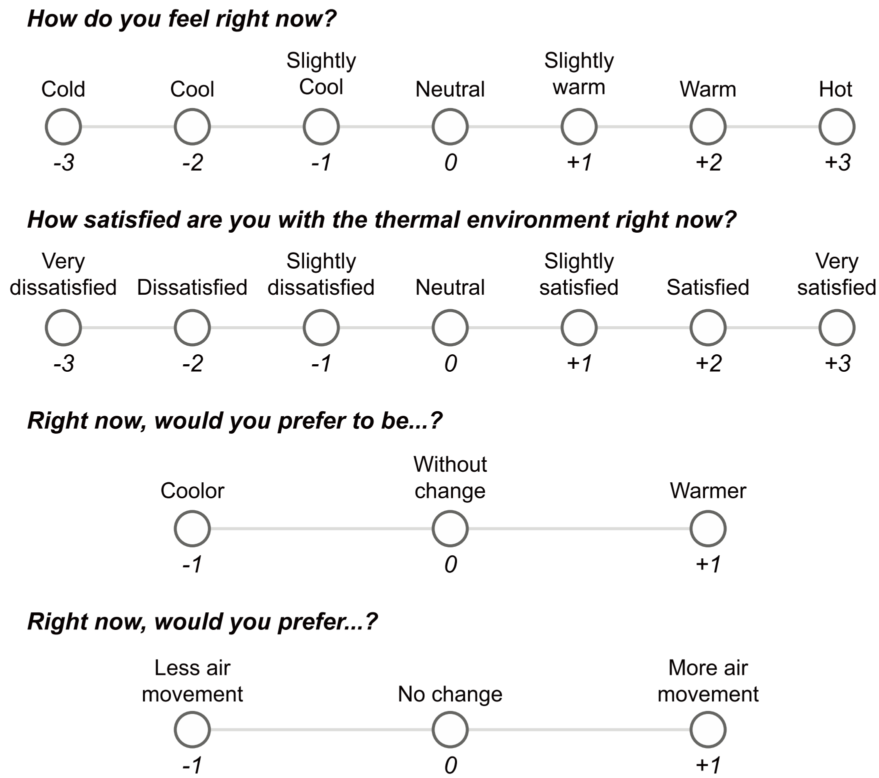

Figure A3. Thermal comfort questions.

Table A1. Contingency table for the question "How do you feel right now?" of groups exposed to air and radiant cooling conditions.

	Condition A: air system	**Condition B: radiant system**
Cold	7	3
Cool	62	33
Slightly cool	65	81
Neutral	74	68
Slightly warm	39	55
Warm	5	8

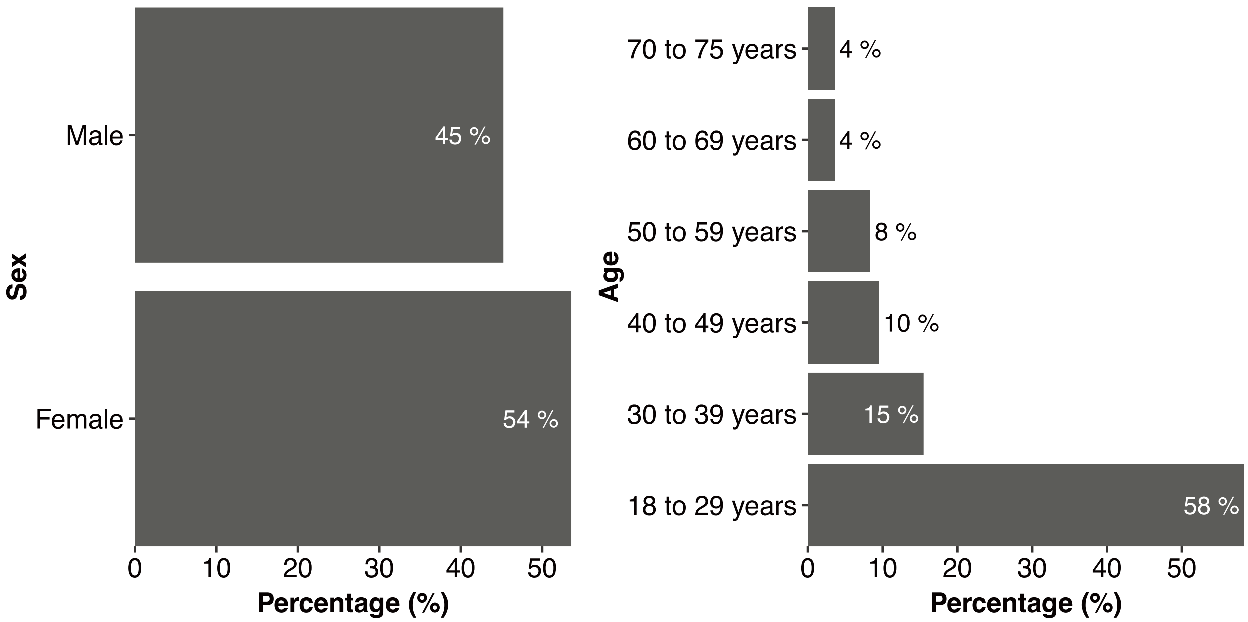

Figure A4. Sex and age distributions for all participants.

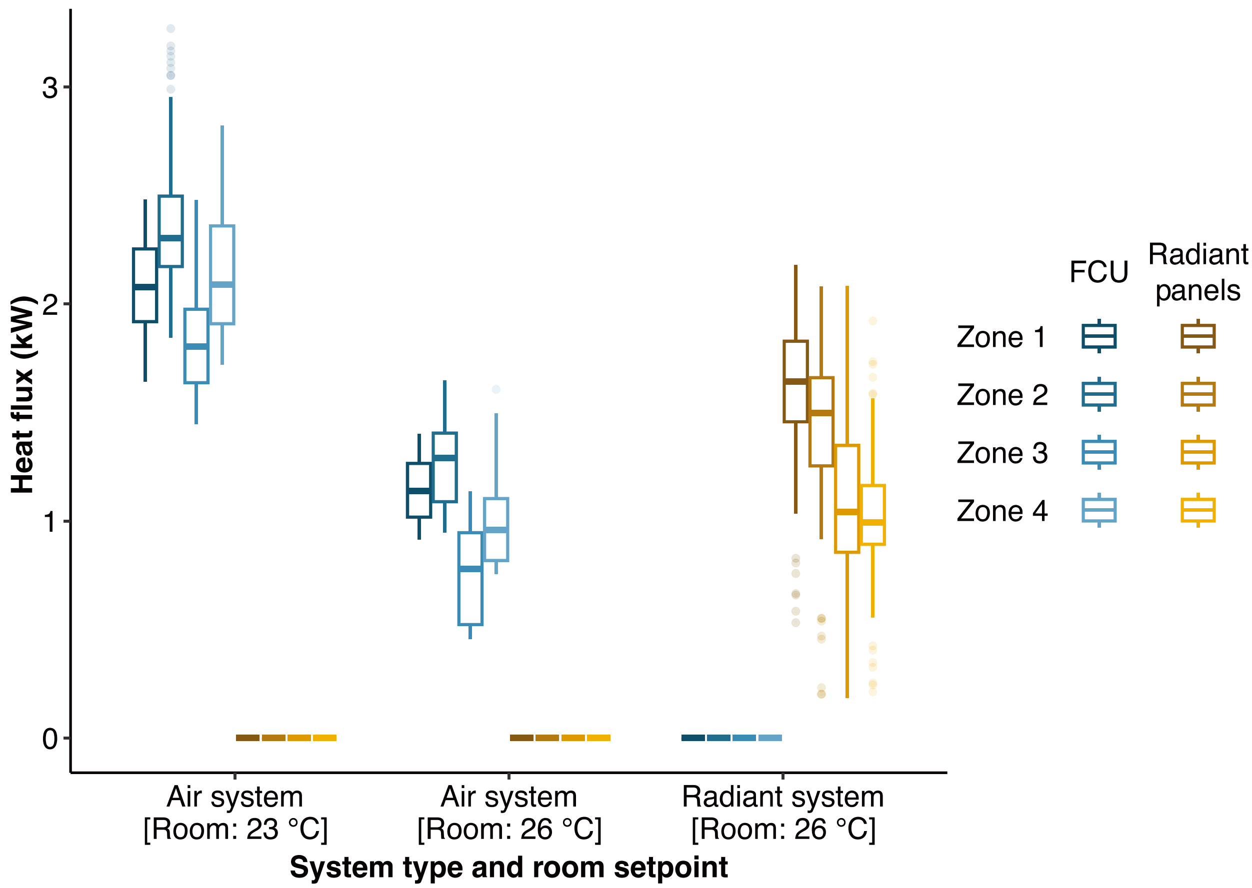

Figure A5. Heat flux extracted by air and radiant system per each zone.

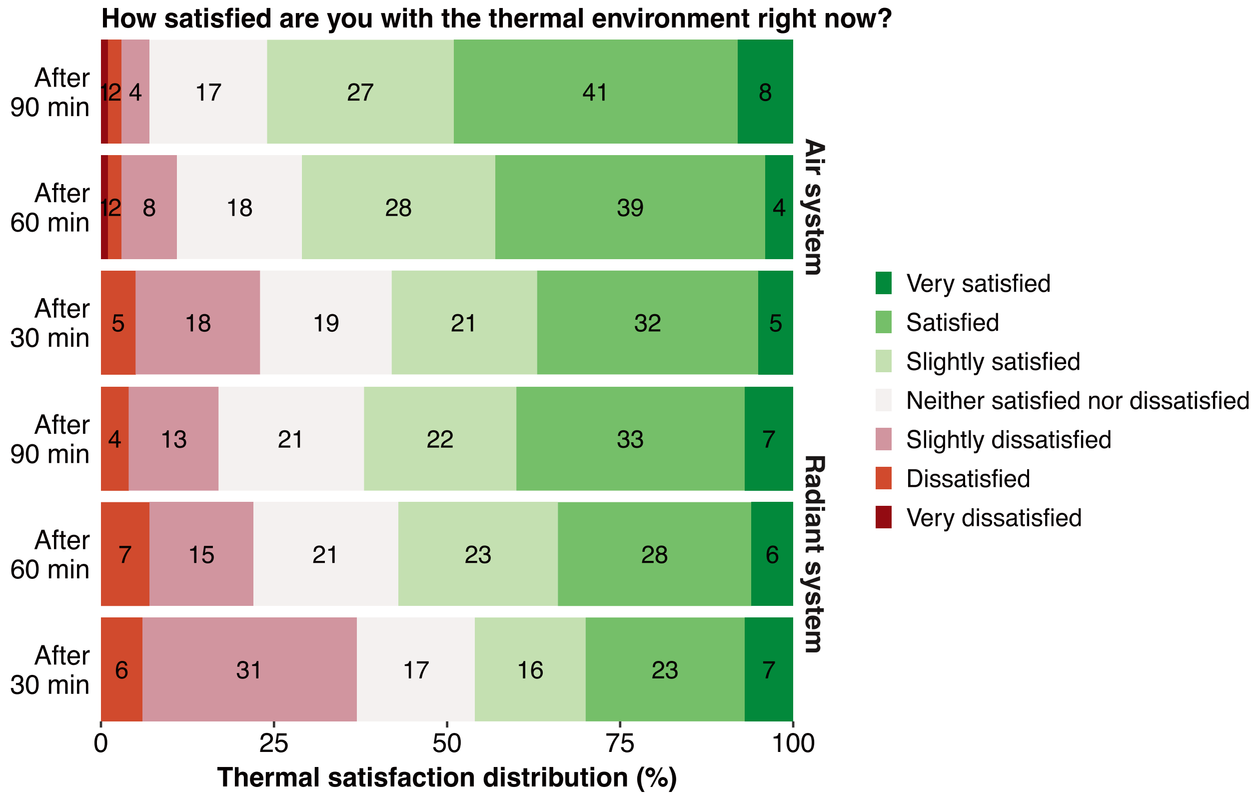

Figure A6. Thermal satisfaction under air and radiant systems cooling after 30, 60, and 90 min.

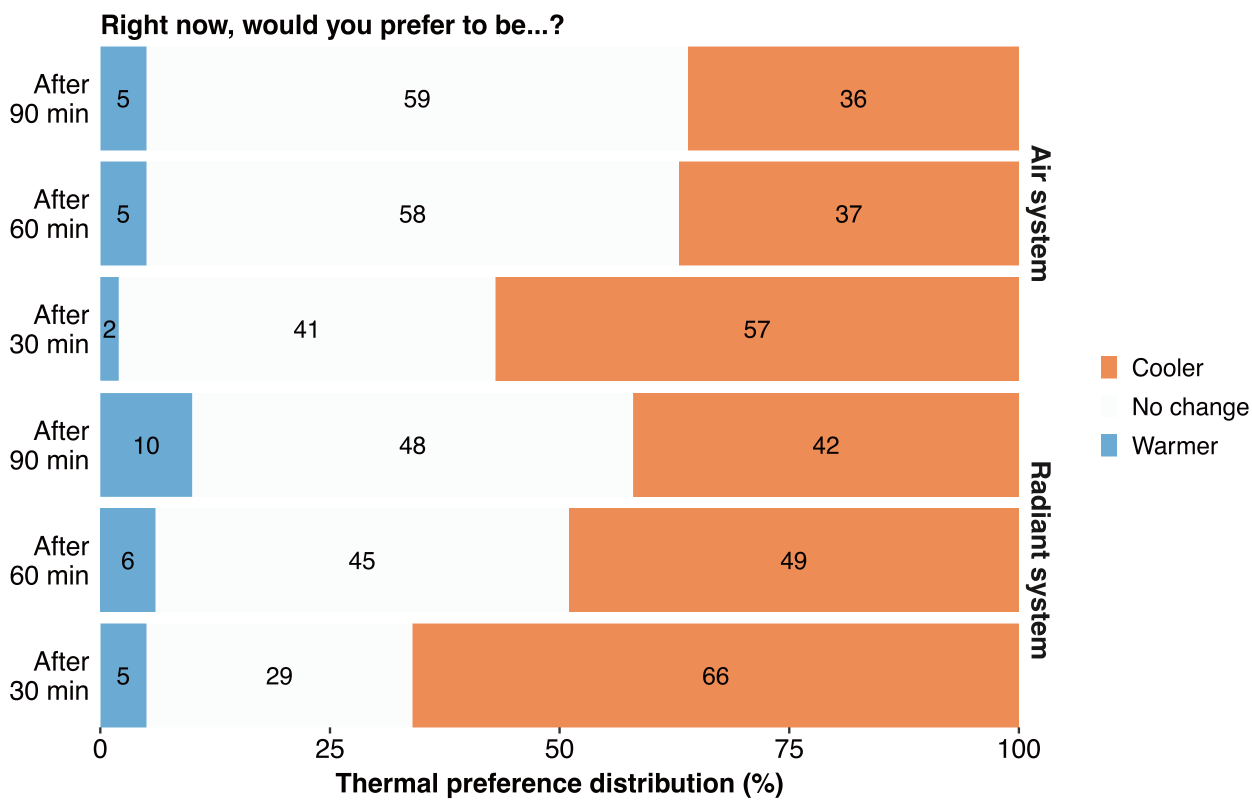

Figure A7. Thermal preference under air and radiant systems cooling after 30, 60, and 90 min.

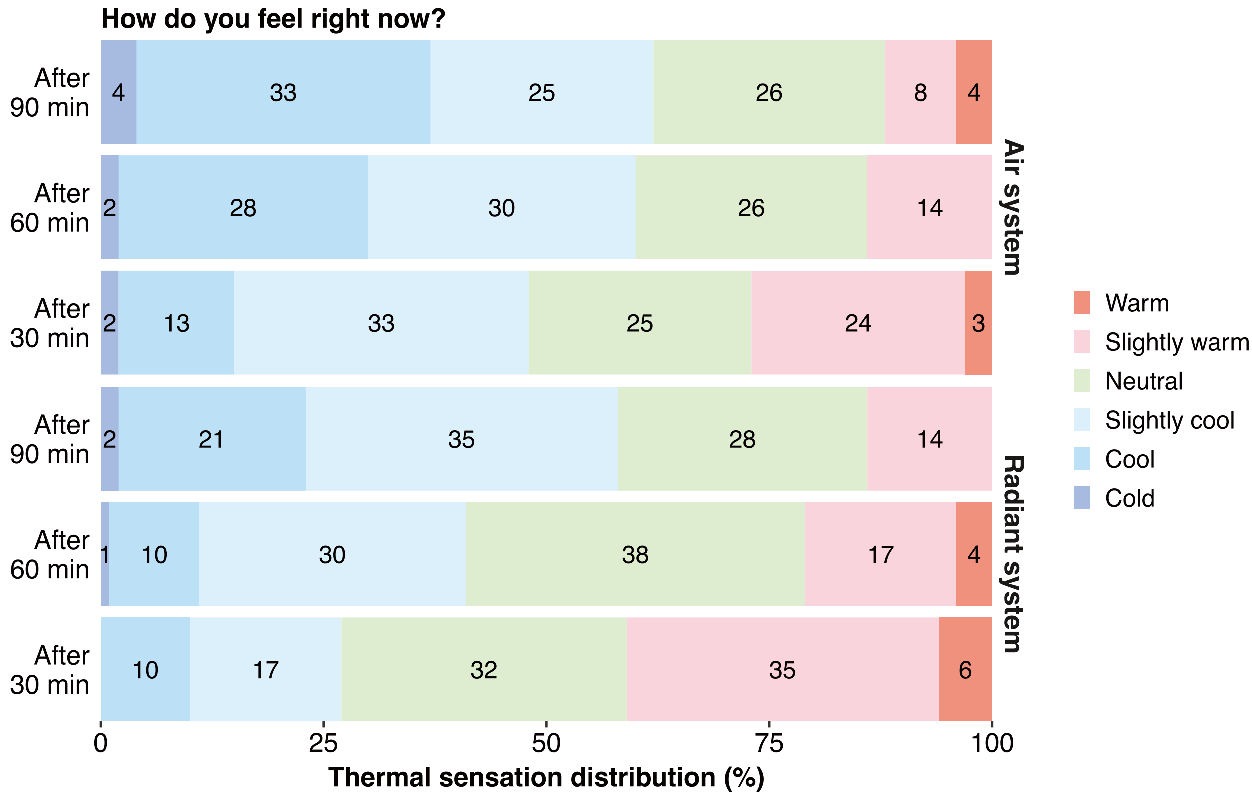

Figure A8. Thermal sensation under air and radiant systems cooling after 30, 60, and 90 min.

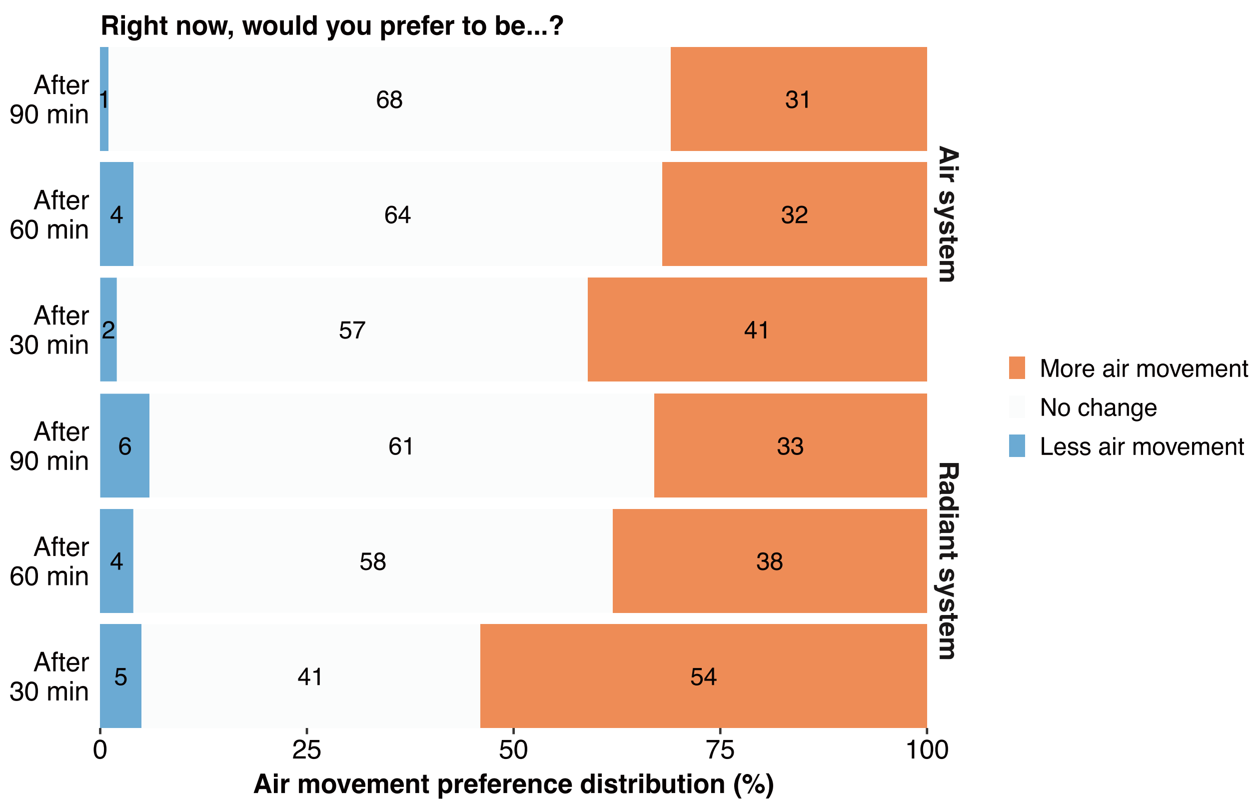

Figure A9. Air movement preference under air and radiant systems cooling after 30, 60, and 90 min.