Comparative Analysis of System Performance and Thermal Comfort for an Integrated System with PVT and GSHP Considering Two Load Systems: Convective Heating and Radiant Floor Heating

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Abstract: The zero-energy building (ZEB) concept has a high potential for securing energy savings in the building sector. To achieve ZEB, various active systems, including renewable systems such as photovoltaic, solar heating, and geothermal systems, have been developed. However, the existing systems are costly or not optimized. To overcome these issues, the authors previously developed an integrated tri-generation system. In this research, the previously developed system was comprehensively analyzed considering the indoor thermal comfort and energy efficiency to develop a design and operation method for the integrated system. Two different heating systems (convective heating and radiant floor heating) were employed in the tri-generation system, and their system performance, predicted mean vote (PMV), and predicted percentage of dissatisfied (PPD) were compared using simulations. The results showed that the heating coefficient of power of the radiant floor heating system was 18.8% higher than that of the convective heating system. Moreover, the radiant floor heating system (Case 4) met the PMV and PPD standards during all the heating periods. Overall, radiant floor heating was found to be more efficient than convective heating. The results confirm that radiant floor heating is more suitable than convective heating considering the indoor thermal comfort of occupants.

Keywords: tri-generation system; photovoltaic-thermal module; ground-source heat pump; convective heating; radiant floor heating; energy efficiency; thermal comfort

1. Introduction

In recent years, issues in energy saving in the building sector have become significant from the aspects of not only global warming and energy security but also economic stability. Energy cost accounts for 50% of the total heating, ventilation, and air conditioning (HVAC) system cost of the building life cycle [1]. Therefore, energy costs must be considered during the building design stage to ensure energy saving in the building sector. The zero-energy building (ZEB) concept is one of the most suitable approaches aimed at achieving this purpose. Many leading countries have actively participated in the dissemination of ZEB through policy support or financial incentives. In the U.S., Executive Order (EO) 13,834 requires all new federal buildings entering the planning process in 2020 to be designed to achieve annual reductions in building energy use and implement energy efficiency measures that reduce costs [2]. In South Korea, ZEB became mandatory for new public buildings from 2020, and this will be mandated for new private buildings by 2025 [3]. The expected market size of ZEB in Korea will also reach up to 17 billion dollars by 2024 [4]. Under this background, various active
systems, including renewable systems such as photovoltaic (PV), solar heating, and geothermal systems, have been developed and applied to achieve ZEB.

In the pursuit of ZEB, PV systems are generally applied for energy production as an active system. However, the area available for PV installation in urban buildings is often insufficient to meet the total energy requirement of the buildings. Furthermore, solar heating systems are fundamentally limited by the time gap between heat collection and heat use. A potential solution is to use a ground-source heat pump (GSHP) systems for space heating, cooling, and hot water. However, the initial cost for GSHP installation is quite high, and it requires electricity for independent operation. In an attempt to overcome the weaknesses of individual renewable systems, the authors previously developed an integrated system using multiple renewable energies and applied it in buildings [5–7]. The feasibility of the system was analyzed, and a tri-generation system integrating PV thermal collector (PVT) and GSHP was found to be acceptable with a payback period of 8 years in the market of HVAC systems if a subsidy for installation is provided [7]. Nevertheless, it is necessary to find a solution for achieving affordability and feasibility without an additional subsidy, because government policy support, such as incentives and subsidies, cannot be depended upon forever.

System feasibility depends on the building and climate conditions, because the system capacity and energy consumption of HVAC systems could also change according to the weather or building load conditions [8,9]. Papadopoulos et al. [10] suggested the optimum heating and cooling setpoint temperatures of HVAC systems for large office buildings, considering energy savings and occupant thermal comfort. For this purpose, they analyzed the potential of energy savings and thermal comfort in seven climate zones across the U.S. Kim et al. [11] investigated the indoor thermal environment induced by four HVAC systems—namely, constant air volume (CAV), variable air volume (VAV), underfloor air distribution (UFAD), and dedicated outdoor air system (DOAS). In addition, they analyzed the cooling energy consumptions and CO₂ emissions using a dynamic analysis program. As a result, the cooling energy was increased by about 23.3% when they applied CAV compared to VAV. Moreover, the CO₂ emissions were 34,679 kg of CO₂ in the VAV system and 42,400 kg of CO₂ in the CAV system. Robledo-Fava et al. [12] analyzed the influence of clothing factor and human metabolic index for calculating the energy demand of buildings based on ISO 7730. Kim et al. [13] analyzed the influence of the control methods of a variable refrigerant flow (VRF) system on energy consumption and indoor thermal comfort using a simulation tool. The energy consumption of VRF using the PMV control showed a reduction of about 24% compared with the 26 °C indoor set point control. In addition, the results of thermal comfort indicated that approximately 86.8% of occupants were satisfied with acceptable thermal comfort levels. Mohelniková et al. [14] evaluated the energy consumption, indoor thermal comfort, and daylight in school buildings and found that windows are key elements influencing indoor climate and visual comfort. However, it was not of significant importance for heating energy reduction. Elhadad et al. [15] assessed the building energy and PMV for common residential houses in Egypt using simulation simplification methods considering four scenarios. All the simplification scenarios exhibited a marginal average deviation in the total energy demand and indoor thermal comfort by 20%. Park et al. [16] investigated the thermal comfort of a combined radiation-convection floor heating system through an experimental test and found that the convection heating type is inappropriate for occupants who sleep on the floor because the floor temperature is too low. In contrast, the combined radiation-convection heating type provided a thermally comfortable environment for the occupants. Fan et al. [17] evaluated the indoor thermal comfort and performance characteristics of an under-floor-air-distribution (UFAD) system under heating mode. They recommended an optimal return vent height of 1.05–1.25 m to maximize the level of indoor thermal comfort and indoor air quality with minimum energy input over a whole year.

Many studies have reported that the use of a tri-generation system for achieving occupant thermal comfort according to setpoint temperature and indoor air temperature control can be a reasonable solution to the realization of ZEB [18–21]. In this research, a tri-generation system was comprehensively analyzed to develop a method for its design and operation, considering human thermal comfort.
and energy efficiency. To this end, an integrated simulation was performed with building, individual equipment, and weather models. In the integrated renewable energy system, the PVT and GSHP systems were used as energy sources, and two different heating systems were compared in terms of the system performance and PMV based on annual dynamic energy simulation.

2. Integrated Simulation Model

2.1. Overview of the Integrated Simulation Model

Figure 1 shows a summary of the integrated simulation model used in this research. The system has PVT and GSHP as heat source systems, a fan coil unit (FCU) and floor heating as space heating systems, and an FCU as a cooling system. The integrated model has a thermal storage tank for storing solar and ground heat. The building is based on ISO 13,790 and the calculation condition presented in [18]. The energy efficiency was calculated based on the total energy production and consumption by all equipment in the system. Furthermore, the PMV was calculated by the indoor air temperature, air velocity, humidity, mean radiation temperature (MRT), metabolic rate (MET), and clothing factor (Clo.), which are based on ISO 7730 [22].

![Summary of the integrated simulation model.](image)

The PVT model, thermal storage model, heat pump model, and ground heat exchanger (GHX) model used in the integrated simulation are based on the tri-generation model built by the author in a previous study [7]. In this study, an convective heating system model using an FCU and a radiant floor heating system was constructed to analyze the energy efficiency and thermal comfort according to the heating type and operating conditions. Figure 2 shows a schematic of the integrated model for the convective heating system.

Figure 3 shows the control strategy of the integrated system. The integrated system is operated based on the load of the building and the indoor setpoint temperature. The operation of each unit (PVT module and heat pump) is determined by the setpoint temperature of the heat storage tank (HST) and outlet temperature of the PVT module. Figure 3 shows the control strategy of the integrated system. The control strategy includes the heating operation and heat storage operation. A heating operation is performed when the average temperature of the heat storage tank is 45 °C or higher.
Figure 2. Schematic diagram of the integrated model for convective heating and radiant floor heating.

The heat storage operation using the PVT module is conducted when the outlet temperature of the PVT module is 2 °C higher than the average temperature of the HST. If the heat storage operation cannot be conducted using the PVT module, the heat storage operation is performed with the heat pump. At this time, the heat pump is used the solar-thermal storage tank (SST) and GHX; the heat pump is utilized with a higher outlet temperature between SST and GHX. The PVT module stores heat to the SST when the HST average temperature is above 45 °C.

2.2. Analysis Method (Equation) of Integrated Simulation Model

Figure 4 shows the structure of the system of thermo-active construction elements. Thermo-active elements, such as the radiant floor model, are used to condition buildings by integrating a fluid system into the building.
A two-dimensional temperature field occurs in the plane of the cross-section of the thermally active component due to the finite distance between the pipes. Thermal input or output through the pipe loop changes the water temperature in the pipe [23]. A stationary solution for the temperature distribution in the x-y plane is represented by Equation (1) for heat flow from the surface towards the indoor space:

$$q_a = \Phi \cdot U_1 \cdot (\delta_3 - \delta_1) + (1 - \Phi) \cdot \frac{U_1 \cdot U_2}{U_1 + U_2} \cdot (\delta_2 - \delta_1)$$  \hspace{1cm} (1)

$$\Phi = \frac{2 \cdot \pi \cdot \lambda_b \cdot \Gamma}{d_x (U_1 + U_2)}$$  \hspace{1cm} (2)

$$\Gamma = \left[ \ln \left( \frac{d_x}{\pi \cdot \delta} \right) + \frac{2 \cdot \pi \cdot \lambda_b}{d_x (U_1 + U_2)} + \sum_{s=1}^{\infty} \frac{\delta_1(s) + \delta_2(s)}{s} \right]^{-1}$$  \hspace{1cm} (3)

The calculation of MRT was defined by Fanger [24]. MRT is the uniform temperature of an imaginary enclosure in which the radiant heat transfer in the actual nonuniform enclosure. MRT varies according to the person’s posture and the color of clothing for short-wave radiation, and calculates based on a person in relation to the surrounding wall and the surface temperature of the enclosure [25].

$$T_{\text{MRT}} = T_1 F_{p-1} + T_2 F_{p-2} + \cdots + T_n F_{p-n},$$  \hspace{1cm} (4)

where $T_n$ is the surface temperature of the area and $F_p$ is the angular factor between a person and the area.

The calculation model of PMV is based on ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) Standard 55-2013 [26]. This PMV model can be calculated for a wide range of six parameters (air temperature, MRT, relative humidity, air velocity, Clo., and MET) based on the thermal load of a body in the environment:

$$\text{PMV} = L \left( 0.303 e^{-0.036 M} + 0.028 \right),$$  \hspace{1cm} (5)

$$L = (M - W) - E_s - E_{re} - C_{re} - R - C,$$  \hspace{1cm} (6)

$$M = \frac{21(0.23 Q_R + 0.77) Q_o}{A_D},$$  \hspace{1cm} (7)

$$A_D = 0.202 m^{0.425} \rho^{0.725},$$  \hspace{1cm} (8)

where PMV is the predicted mean vote, $L$ is the thermal load, $M$ is the metabolic rate, $W$ is the rate of mechanical work accomplished, $E_s$ is the total rate of evaporative heat loss from the skin, $E_{re}$ is the rate of evaporative heat loss from respiration, $C_{re}$ is the rate of convective heat loss from respiration, $R$ and $C$...
are the heat losses, $RQ$ is the respiratory quotient, $Q_{O_2}$ is the volumetric rate of oxygen consumption at $0 \, ^{\circ}C$, $AD$ is the DuBois surface area, $m$ is the person’s weight, and $l$ is the person’s height.

After calculating PMV, the predicted percentage dissatisfied (PPD) can also be estimated as:

$$PPD = 100 - 95e^{-0.03353PMV^4 + 0.2179PMV^2}. \quad (9)$$

In ASHRAE Standard 55-2013, the satisfaction range of PMV is $\pm 0.5 \ ( -0.5 < \text{PMV} < +0.5 )$, and the range of PPD is below 10% (PPD < 10%) [26].

The heat pump model is based on a user-supplied file [27]. This file contains catalog data for the source-side and load-side flow rate, entering load temperature, entering source temperature, capacity, and power. The heating and cooling coefficient of performance (COP) of the heat pump can be calculated by Equations (11) and (12) [27].

$$\text{COP}_{\text{HP,heating}} = \frac{Q_{\text{absorbed}} + P_{\text{heating}}}{P_{\text{heating}}}, \quad (10)$$

$$\text{COP}_{\text{HP,cooling}} = \frac{Q_{\text{rejected}} - P_{\text{cooling}}}{P_{\text{cooling}}}, \quad (11)$$

where $\text{COP}_{\text{HP,heating}}$ is the heat pump COP in heating mode, $Q_{\text{absorbed}}$ is the energy absorbed by the heat pump in heating mode, $P_{\text{heating}}$ is the power drawn by the heat pump in heating mode, $\text{COP}_{\text{HP,cooling}}$ is the heat pump COP in cooling mode, $Q_{\text{rejected}}$ is the energy rejected by the heat pump in cooling mode, and $P_{\text{cooling}}$ is the power drawn by the heat pump in cooling mode. The heat pump is controlled by fluid flow through absorbing energy in the heating mode or rejecting energy in the cooling mode.

COP was calculated considering the total heat production and total energy consumption. The total heat production is the sum of the heat production of the photovoltaic-thermal module and the heat pump. The total energy consumption is the sum of the power consumption of the FCU, the circulating pumps, and the heat pump.

$$\text{COP}_{\text{SYS}} = \frac{Q_{\text{PVT}} + Q_{\text{HP}}}{P_{\text{FCU}} + \sum P_{\text{pump}} + P_{\text{HP}}}. \quad (12)$$

where $\text{COP}_{\text{SYS}}$ is the coefficient of the performance of the system, $Q_{\text{PVT}}$ is the heat production by the photovoltaic-thermal module, $Q_{\text{HP}}$ is the heat production by the heat pump, $P_{\text{FCU}}$ is the power consumption of the fan coil unit, $\sum P_{\text{pump}}$ is the sum of the power consumption of all circulating pumps, and $P_{\text{HP}}$ is the power consumption of the heat pump.

The thermal and electrical performance of the PVT module model was calculated based on the solar radiation, module area, electrical efficiency, and slope. The calculation method of the PVT module model is based on the book “Solar Engineering of Thermal Process,” described by Duffie and Beckman [28]. The GHX model is based on the duct ground heat storage (DST) developed by Hellstrom [29]. In addition, the heat storage tank can be modeled assuming that the tanks are made up of equal volume segments [30]. The degree of stratification and tank volume are determined by the user. More details of the system unit (PVT module, heat pump, GHX, and HST) can be found in the author’s previous study [7].

3. Simulation Condition and Cases

3.1. Building Model Conditions

Figure 5 shows the floor plan of the building model. The building model was implemented utilizing the TRNSYS simulation program. The building model was based on the standard design of the low-energy residential houses proposed by the Korean Ministry of Agriculture, Food and Rural Affairs [31]. The floor area of the building model was 139.75 m$^2$ and the height from the floor to
the ceiling was 2.4 m. The window to wall ratio (WWR) according to the cardinal direction was 1.2% in the east, 9.9% in the north, and 14.1% in the south.

![Floor plan of the building model.](image)

**Figure 5.** Floor plan of the building model.

Table 1 shows the U-values of the building model, which are based on the “Green Buildings Construction Support Act” proposed by the Korean Ministry of Land, Infrastructure, and Transport [3]. The U-values were input assuming the building model to be located in Seoul.

**Table 1. U-value of the building model.**

<table>
<thead>
<tr>
<th>Construction Type</th>
<th>U-Value (W/m²·K)</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>External wall</td>
<td>0.242</td>
<td>Green Buildings Construction Support Act–2018</td>
</tr>
<tr>
<td>Ground Floor</td>
<td>0.240</td>
<td>(Korean Ministry of Land, Infrastructure, and Transport)</td>
</tr>
<tr>
<td>Roof</td>
<td>0.151</td>
<td></td>
</tr>
<tr>
<td>Exterior window</td>
<td>1.49</td>
<td></td>
</tr>
</tbody>
</table>

Figure 6 shows the schematic diagram of the floor structure. The floor structure is composed of a finishing material, mortar, a pipe, insulation, and a concrete slab. Table 2 shows the thermal properties of the construction material.

![Schematic diagram of the floor structure.](image)

**Figure 6.** Schematic diagram of the floor structure.
Table 2. Thermal properties of the construction material.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Finishing</th>
<th>Mortar</th>
<th>Insulation</th>
<th>Concrete</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity (W/m·K)</td>
<td>0.166</td>
<td>1.51</td>
<td>0.032</td>
<td>1.52</td>
</tr>
<tr>
<td>Thermal capacity (kJ/kg·K)</td>
<td>1</td>
<td>0.8</td>
<td>1.47</td>
<td>0.9</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>800</td>
<td>2000</td>
<td>30</td>
<td>2300</td>
</tr>
</tbody>
</table>

Figure 7 shows the schedules of occupants, lights, equipment, and system for a day. The schedule values of internal heat gain were based on ASHRAE 90.1-2004 [32]. The system operated for 24 h.

Table 3 shows the indoor conditions of the building model. The occupants were assumed to be seated and relaxed. The number of occupants was set to four, considering the floor area. In addition, the lighting power density (LPD), equipment power density (EPD), and infiltration were set based on ASHRAE 90.1-2004 and ASHRAE 90.2-2004 [33].

Table 3. Indoor conditions of the building model.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>People</td>
<td>Sensible heat 75 W</td>
<td>ASHRAE Standard 55-2013</td>
</tr>
<tr>
<td></td>
<td>Latent heat 45 W</td>
<td></td>
</tr>
<tr>
<td>Light</td>
<td>10.76 W/m²</td>
<td>ASHRAE Standard 90.1-2004</td>
</tr>
<tr>
<td>Equipment</td>
<td>9.68 W/m²</td>
<td>ASHRAE Standard 90.1-2004</td>
</tr>
<tr>
<td>Infiltration</td>
<td>0.5 ACH</td>
<td>ASHRAE Standard 90.2-2004</td>
</tr>
</tbody>
</table>

Table 4 shows the thermal comfort conditions. The occupants, who were wearing simple clothes, were assumed to be seated and relaxed.
Table 4. Thermal comfort conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clothing factor</td>
<td>Winter 1.0 Clo., Summer 0.5 Clo.</td>
<td>ASHRAE Standard 55-2013</td>
</tr>
<tr>
<td>Metabolic rate</td>
<td>1 MET</td>
<td></td>
</tr>
<tr>
<td>Air velocity</td>
<td>0.1 m/s</td>
<td></td>
</tr>
</tbody>
</table>

3.2. Simulation Cases

In this study, a case study was conducted on the heating type and operating conditions. Table 5 shows the case study conditions. The heating type was classified as (1) the convective heating system using FCU and (2) the radiant floor heating system. Thermal comfort, according to the heating type, was analyzed through Case 1 and Case 2. The indoor setpoint temperatures for Case 1 and Case 2 were based on the ASHRAE 90.2-2004 single zone [33]. Case 3 and Case 4 focused on the thermal comfort of the occupant rather than the energy efficiency of the system. For the indoor setpoint temperatures for Case 3 and Case 4, the range of comfortable heating and cooling temperatures suggested by ASHRAE standard 55-2013 was considered [26].

Table 5. Case study conditions.

<table>
<thead>
<tr>
<th>Heating Type</th>
<th>Indoor Setpoint Temperature</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>Heating 20 °C</td>
<td>ASHRAE 90.2-2004</td>
</tr>
<tr>
<td>Case 2</td>
<td>Heating 22 °C</td>
<td>ASHRAE standard 55-2013</td>
</tr>
<tr>
<td>Case 3</td>
<td>Heating 20 °C</td>
<td>ASHRAE 90.2-2004</td>
</tr>
<tr>
<td>Case 4</td>
<td>Heating 22 °C</td>
<td>ASHRAE standard 55-2013</td>
</tr>
</tbody>
</table>

4. Simulation Result

4.1. Building Load

Figure 8 shows the ambient temperature of Seoul and the monthly energy demand of the building model. The annual average temperature in Seoul is 11.8 °C; the minimum is −14.5 °C and the maximum is 33.8 °C. The annual heating and cooling loads were calculated to be 11,039 kWh and 1335 kWh, respectively. Furthermore, the heating and cooling peaks were 9.35 and 2.72 kW, respectively. Therefore, the building model of the case study was assumed to be a heating-dominant building. The system capacity of each component was set according to the results of load calculation, and Table 6 presents the specific information of the system.

Figure 8. Ambient temperature and energy demand of the building model.
### Table 6. Component conditions of the integrated simulation model.

<table>
<thead>
<tr>
<th>Component</th>
<th>Name</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>PVT collector</td>
<td>Type</td>
<td>Unglazed water</td>
</tr>
<tr>
<td></td>
<td>Length and width</td>
<td>1.01 m × 19.72 m (19.95 m²)</td>
</tr>
<tr>
<td></td>
<td>Number of tubes</td>
<td>210</td>
</tr>
<tr>
<td></td>
<td>PV efficiency</td>
<td>20%</td>
</tr>
<tr>
<td></td>
<td>Slope</td>
<td>35°</td>
</tr>
<tr>
<td>Heat pump</td>
<td>Type</td>
<td>Water-to-water</td>
</tr>
<tr>
<td></td>
<td>Heating capacity</td>
<td>10.5 kW</td>
</tr>
<tr>
<td></td>
<td>Cooling capacity</td>
<td>11.3 kW</td>
</tr>
<tr>
<td>Ground heat exchanger</td>
<td>Number of boreholes</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>Borehole radius</td>
<td>0.15 m</td>
</tr>
<tr>
<td></td>
<td>Borehole depth</td>
<td>200 m</td>
</tr>
<tr>
<td>Circulating pump</td>
<td>Number of pumps</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Power consumption</td>
<td>2 kW</td>
</tr>
<tr>
<td>Fan coil unit</td>
<td>Heating capacity</td>
<td>15 kW</td>
</tr>
<tr>
<td>(Convective heating)</td>
<td>Cooling capacity</td>
<td>12 kW</td>
</tr>
<tr>
<td></td>
<td>Fan power</td>
<td>0.09 kW</td>
</tr>
<tr>
<td>Floor heating pipes</td>
<td>Pipe spacing</td>
<td>0.2 m</td>
</tr>
<tr>
<td>(Radiant floor heating)</td>
<td>Pipe outside diameter</td>
<td>0.02 m</td>
</tr>
<tr>
<td></td>
<td>Pipe wall thickness</td>
<td>0.001 m</td>
</tr>
<tr>
<td></td>
<td>Pipe wall conductivity</td>
<td>1.36 kW/ h·m·K</td>
</tr>
</tbody>
</table>

### 4.2. Energy Efficiency

#### 4.2.1. Thermal and Electric Performance on a Representative Day

Figure 9 shows the thermal and electrical performance of the system on a representative day (January 28) in winter. The heat exchange rate (HER) of the heat pump was calculated as 105 and 110 kW for convective heating (Case 1) and radiant floor heating (Case 2), respectively. There was no significant difference in the amount of heat and electricity produced by the PVT collector according to the heating type.

![Figure 9. Thermal and electric performance on a representative day in winter (Case 1 and Case 2).](image-url)
Under the same heating load condition, the convective heating (Case 1) system operated continuously, and the radiant floor heating (Case 2) system operated intermittently. The electricity consumption of the heat pump was calculated as 24.3 kW for convective heating (Case 1) and 20.9 kW for radiant floor heating (Case 2). The operation time of the heat pump was 50% longer for convective heating than for radiant floor heating. More heat was produced via radiant floor heating than via convective heating, although the operating time of the heat pump was shorter.

Figure 10 shows the COP of the heat pump on a representative day for heating. The average COP of the heat pump on the representative day of heating was calculated as 4.3 and 5.2 for convective heating (Case 1) and radiant floor heating (Case 2), respectively. The heating COP of radiant floor heating was 18.8% higher than that of convective heating, indicating that the heating performance of the heat pump of the radiant floor heating was more efficient. For the cooling COP, there was no difference between the two systems.

4.2.2. Weekly Performance of Heat Pump and System

Figure 11 shows the HER, electric consumption, heat pump COP, and system COP from January 12 to January 18. The average COP of the heat pumps for convective heating (Case 1 and Case 3) and radiant floor heating (Case 2 and Case 4) was calculated as 4.3, 5.3, 4.3, and 5.3, respectively. The average COP of the system for convective heating (Case 1 and Case 3) and radiant floor heating (Case 2 and Case 4) was calculated as 3.5, 4.5, 3.5, and 4.5, respectively. The heating type had a more substantial influence on the COP of the heat pump and the system than the indoor set temperature. Table 7 shows a summary of the average COP in Figure 11c,d.

4.2.3. Monthly Performance of Heat Pump and System

Figure 12 shows the monthly performance of the heat pump and the system. The average COP of the heat pump in the winter season was calculated as 4.6, 5.4, 4.6, and 5.4 from Case 1 to Case 4, respectively. In addition, the average system COP was calculated as 4.5, 5.3, 4.3, and 5.2. The heating performance varied with the heating type. The heat pump COP and system COP for radiant floor heating were 18.7% and 21.8% higher than those for convective heating. These results clearly show the advantage of radiant floor heating over convective heating in terms of the HER and electricity consumption (operation time), as shown in Figure 9.
4.2.3. Monthly Performance of Heat Pump and System

Figure 12 shows the monthly performance of the heat pump and the system. The average COP of the heat pump in the winter season was calculated as 4.6, 5.4, 4.6, and 5.4 from Case 1 to Case 4, respectively. In addition, the average system COP was calculated as 4.5, 5.3, 4.3, and 5.2. The heating performance varied with the heating type. The heat pump COP and system COP for radiant floor heating were 18.7% and 21.8% higher than those for convective heating. These results clearly show the advantage of radiant floor heating over convective heating in terms of the HER and electricity consumption (operation time), as shown in Figure 9.

4.3. Indoor Thermal Comfort

4.3.1. Thermal Environmental Parameters

Figure 13a shows the indoor air temperatures ($T_a$) of convective heating and radiant floor heating on a representative day in winter (28 January). The indoor air temperature of convective heating was calculated as 21.9 °C and 22.2 °C for Case 1 and Case 3, respectively. In addition, the indoor air temperature of radiant floor heating was calculated as 19.9 °C and 21.9 °C for Case 2 and Case 4, respectively. For radiant floor heating, the indoor air temperature was confirmed to be similar to...
Figure 13. Indoor air temperature on a representative day and week.

Figure 13b shows the indoor air temperatures ($T_a$) of the convective heating and radiant floor heating for a week period (January 25–31). Convective heating exhibited a larger variation in indoor air temperature than radiant floor heating. On the other hand, the indoor air temperature of the radiant floor heating is almost constant. This is because the heat output from the heated floor is controlled according to the indoor air temperature due to the self-control effect of radiant floor heating [34].

Figure 14a shows the MRT on a representative day. Changes in the MRT of convective heating and radiant floor heating were similar to those of indoor air temperature (Figure 12). For radiant floor heating, MRT was higher than the indoor air temperature by approximately 1.3 °C. In contrast, for convective heating MRT was lower than the indoor air temperature by approximately 1.4 °C.

Figure 14b shows the MRT for a week. For convective heating, the average MRT during the week was calculated as 20.6 and 21.0 °C in Case 1 and Case 3, respectively. For radiant floor heating, the corresponding values were 21.2 and 23.4 °C. The MRT exhibited a variation according to the heating type. The MRT of Case 4 increased by 2.3 °C compared to Case 3. The radiant floor heating system is a higher MRT than the convective heating system because of the increased radiant heat exchange.

Figure 15a shows the monthly average indoor air temperature ($T_a$) for one year. For convective heating, the indoor air temperature was higher than the indoor set-point temperature by approximately 1 °C. This is because the convective heating system can supply high-temperature heat to the zone in a short time compared to the radiant floor heating system. In contrast, for radiant floor heating, the indoor air temperatures were 20.6 and 22.3 °C in Case 2 and Case 4, respectively.
In contrast, radiant floor heating (Case 4) met the PMV standard during all the heating seasons. The average PMVs for radiant floor heating and convective heating at low heating loads (April, May, and September) were 4.3 and 5.2 for convective heating (Case 1) and radiant floor heating (Case 2), respectively. The heating efficiency and indoor thermal comfort of two different heating systems were compared based on renewable energy system considering the indoor thermal comfort and energy efficiency. The energy efficiency of convective heating show that the PMV was below \(0.5\) throughout the day and did not reach the thermal comfort zone. PMV was higher with radiant floor heating than with convective heating. In particular, the average PMV of Case 4 for radiant floor heating met the thermal comfort zone.

4.3.2. Indoor Thermal Comfort

Figure 16a shows the predicted PMV on a representative day. The red line represents the one of the ranges of the PMV standard \((-0.5 < \text{PMV} < 0.5)\) proposed by ASHRAE Standard 55-2013 [22]. The cases of convective heating show that the PMV was below \(-0.5\) throughout the day and did not reach the thermal comfort zone. PMV was higher with radiant floor heating than with convective heating. In particular, the average PMV of Case 4 for radiant floor heating met the thermal comfort zone.

Figure 16b shows PMV for a week. The average PMVs in Case 1 to Case 4 were estimated at \(-0.8\), \(-0.9\), \(-0.7\), and \(-0.4\), respectively. The MRT levels for radiant floor heating (Case 2) and convective heating (Case 3) were similar; however, the PMV was higher for convective heating (Case 3) than for radiant floor heating (Case 2). Despite the higher MRT, the PMV decreased with a decreasing indoor air temperature. On the other hand, convective heating (Case 3) reached a period of satisfying PMV on January 27. However, most other periods did not meet the PMV standard. Radiant floor heating (Case 4) satisfied the PMV for a week. These results confirm that the radiant floor heating system is superior to the convective heating system in terms of indoor thermal comfort.

Figure 17a shows the predicted percentage of dissatisfied (PPD) on a representative day. The red line represents the PPD standard (PPD < 10%) suggested by ASHRAE Standard 55-2013 [22]. From Case 1 to Case 4, the PPDs were 18.6%, 24.6%, 16.5%, and 9.5%, respectively. With radiant floor heating, the PPDs were 18.6%, 24.6%, 16.5%, and 9.5%, respectively. With radiant floor heating, the PPDs were 18.6%, 24.6%, 16.5%, and 9.5%, respectively.
(Case 4), though the PPD standard (PPD < 10%) was exceeded once, the standard was sustained throughout the most of the time.

Figure 17 shows the PPD for a week. For convective heating (Case 3), the standard was met once but was not satisfied during the remaining period. The radiant floor heating system satisfied the PPD standard only when the indoor set-point temperature was 22 °C. Even when the indoor air temperature reached a generally acceptable level, it hardly met the PPD standard at a low MRT.

Figure 18a illustrates the monthly average PMV. Neither convective heating (Case 1) nor radiant floor heating (Case 2) met the PMV standard during the heating periods, except in May and October, when the heating load was low. Meanwhile, convective heating (Case 3) did not meet the PMV standard, even though the average indoor air temperature exceeded 22 °C in January and February. In contrast, radiant floor heating (Case 4) met the PMV standard during all the heating periods. The convective heating system did not meet the PMV standard due to the low MRT.

Figure 18b summarizes the monthly average PPD. Both convective heating (Case 1) and radiant floor heating (Case 2) failed to meet the required PPD standard during all the heating periods. Convective heating (Case 3) did not meet the required PPD either in January, February, or December, whereas radiant floor heating (Case 4) met the PPD criteria in all the heating periods. This result confirms that radiant floor heating is more suitable for the indoor thermal comfort of occupants than convective heating.

5. Conclusions

An integrated simulation was performed to develop the design and operation method for a renewable energy system considering the indoor thermal comfort and energy efficiency. The energy efficiency and indoor thermal comfort of two different heating systems were compared based on dynamic energy simulations. The conclusions of this study can be summarized as follows.
(1) The average COPs of the heat pump on a representative day in the winter season were 4.3 and 5.2 for convective heating (Case 1) and radiant floor heating (Case 2), respectively. The heating COP for radiant floor heating was 18.8% higher than that for convective heating. Overall, the radiant floor heating system was more efficient than the convective heating system.

(2) The radiant floor heating system was more effective for MRT than the convective heating system. The MRT showed a difference of 2.3 °C according to the heating type. The MRT difference was larger in January, February, and December, when the heating load was high. This is because the available time for radiant floor heating to heat the floor structure increased with the increasing heating load. However, the MRT was similar for both radiant floor heating and convective heating at low heating loads.

(3) Although the MRT was higher for radiant floor heating (Case 1) than for convective heating (Case 3), the PMV standard (−0.5 < PMV < 0.5) was not met when the indoor air temperature was low. This result confirms that MRT has a strong influence on PMV only if an appropriate level of indoor air temperature (22 °C) is secured. For convective heating (Case 3), the standards of PMV and PPD were not met in January and February when the heating load was high. For radiant floor heating (Case 4), the PMV and PPD standards were met in all heating periods. This result confirms that the radiant floor heating system is more suitable than the convective heating system for the indoor thermal comfort of occupants.

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Nomenclature

Symbols

- \( q_a \) Heat flow [W]
- \( U \) Coefficient of thermal transmittance [-]
- \( d \) distance, thickness [m]
- \( s \) Time constant [-]
- \( T_a \) The air temperature [K]
- \( T_{MRT} \) The mean radiant temperature [K]
- \( T_n \) The surface temperature of the area [K]
- \( F_p \) The angular factor between a person and the area [-]
- \( \text{PMV} \) Predicted mean vote [−3 to 3]
- \( L \) Thermal load [W/m²]
- \( M \) Metabolic rate [W/m²]
- \( W \) Rate of mechanical work accomplished [W/m²]
- \( E_s \) Total rate of evaporative heat loss from skin [W/m²]
- \( E_{re} \) Rate of evaporative heat loss from respiration [W/m²]
- \( C_{re} \) Rate of convective heat loss from respiration [W/m²]
- \( R \) Sensible heat loss from skin [W/m²]
- \( C \) Sensible heat loss from skin [W/m²]
- \( RQ \) Respiratory quotient [W/m²]
- \( Q_{O2} \) Volumetric rate of oxygen consumption at conditions of 0 °C [W/m²]
- \( A_D \) DuBois surface area [m²]
- \( m \) Person’s weight [kg]
- \( l \) Person’s height [m]
- \( \text{PPD} \) Predicted percentage of dissatisfied [%]
COP\textsubscript{HP,heating} The heat pump coefficient of performance in heating mode [-]
COP\textsubscript{HP,cooling} The heat pump coefficient of performance in cooling mode [-]
\(\text{Q}_{\text{absorbed}}\) The energy absorbed by the heat pump in heating mode [W]
\(\text{Q}_{\text{rejected}}\) The energy rejected by the heat pump in cooling mode [W]
\(\text{P}_{\text{heating}}\) The power drawn by the heat pump in heating mode [W]
\(\text{P}_{\text{cooling}}\) The power drawn by the heat pump in cooling mode [W]
COP\textsubscript{SYS} The system coefficient of performance [-]
\(\text{Q}_{\text{PVT}}\) The heat production by photovoltaic-thermal module [W]
\(\text{Q}_{\text{HP}}\) The heat production by heat pump [W]
\(\text{P}_{\text{FCU}}\) The power consumption of fan coil unit [W]
\(\text{P}_{\text{pump}}\) The power consumption of circulating pump [W]
\(\text{P}_{\text{HP}}\) The power consumption of heat pump [W]

Greek Symbols
\(\Phi\) Correction factor [-]
\(\vartheta_1\) Room 1 temperature [K]
\(\vartheta_2\) Room 2 temperature [K]
\(\vartheta_3\) Outside surface temperature of the pipe [K]
\(\delta\) Outside diameter of the pipe [m]
\(\lambda_b\) Thermal conductivity of material [W/m·K]

Acronyms and abbreviations
ZEB Zero-energy building
MRT Mean radiant temperature
PMV Predicted mean vote
PPD Predicted percentage of dissatisfied
COP Coefficient of performance
PVT Photovoltaic-thermal
GHX Ground heat exchanger
SST Solar-thermal storage tank
HST Heat storage tank
FCU Fan coil unit

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