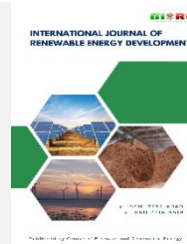




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Research Article

Techno-economic and environmental assessment of source-side and load-side PV/T-assisted ground source heat pump systems

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Abstract. Building heating consumes substantial energy and contributes significantly to global carbon emissions. Integrating photovoltaic/thermal (PV/T) modules with ground source heat pumps (GSHP) offers a promising sustainable solution, effectively addressing the soil thermal imbalance and low energy efficiency inherent in conventional GSHP systems. This study proposes and mathematically models two hybrid configurations: PV/T modules integrated on the source side and the load side, respectively. Component models were validated against experimental data (maximum error < 6%) to conduct a comprehensive 20-year comparative assessment across energy, economic, environmental, and soil thermal balance dimensions. Simulation results indicate that both configurations successfully mitigate soil temperature decline, achieving long-term soil thermal balance and effective seasonal solar thermal storage. However, the source-side PV/T-GSHP system exhibits superior overall performance. It delivers a higher Primary Energy Ratio (PER) and Energy Efficiency Ratio (EER), a lower Levelized Cost of Energy (LCOE) of 0.36 RMB/kWh, an 11.14% higher annual total cost-saving rate, and an 8.47% improvement in CO₂ emission reductions compared to the load-side system. These findings provide optimized configuration strategies and reliable performance data for PV/T-GSHP systems, significantly promoting the efficient utilization of solar-geothermal hybrid energy in low-carbon building applications.

Keywords: PV/T-GSHP system; Nonlinear modeling; Sustainable building heating; Performance comparison.



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1. Introduction

Energy consumption for space heating and domestic hot water (DHW) accounts for nearly 40% of total societal carbon emissions (Wan *et al.*, 2020; W. W. Wang & Zhang, 2015). To mitigate this environmental impact, the integration of solar and shallow geothermal energy has emerged as a promising sustainable solution. Specifically, photovoltaic/thermal (PV/T) technology overcomes the efficiency degradation of conventional photovoltaic panels by simultaneously generating electricity and capturing waste heat (Healy & Ugursal, 1997; Oguma, Matsumoto, & Kakizaki, 2015; Wu, Dai, Lie, Oppong, & Xu, 2018). Concurrently, ground source heat pump (GSHP) systems (Kwon, Bae, Chae, & Nam, 2024; Qian, Yan, An, & Iop, 2018) provide highly efficient space heating but often suffer from long-term soil thermal imbalance (Huang, Sun, Liu, Deng, & Wang, 2026; Sultan & Efzan, 2018; Veeramanikandan, Sathish, Jeryraj Kumar, & Boovendrarman, 2020). Coupling these two technologies into a hybrid PV/T-GSHP system offers a synergistic approach: the PV/T unit supplies power for the system and supplements thermal energy to the ground, thereby enhancing the GSHP's seasonal performance and ensuring long-term sustainable operation (H. Q. Li *et al.*, 2026; T. Li *et al.*, 2024; X. Wang *et al.*, 2024).

Deploying GSHP systems significantly mitigates building thermal demands while reducing greenhouse gas emissions (P.

D. Li, Gao, Li, & Zhou, 2022; S. D. Wang, Bo, & Xiaoyu, 2016; Y. F. Wang, Sun, Xue, & Ieee, 2016). However, the inherent seasonal mismatch between continuous ground heat extraction and limited thermal injection inevitably disrupts the long-term soil thermal balance, severely degrading systemic performance and local ecological conditions. Furthermore, the inherent intermittency of solar irradiance challenges the reliability of standalone solar supply. To overcome these dual constraints, this study proposes a solar-geothermal synergistic approach: the PV/T-assisted GSHP system. In this hybrid architecture, the PV/T unit simultaneously generates the electricity required to drive the system and supplements thermal energy directly to the ground heat exchanger. This process not only accelerates soil thermal recovery—boosting the heat pump's heating COP—but also actively cools the photovoltaic cells, thereby optimizing their electrical yield.

Despite the theoretical advantages of PV/T-GSHP systems, previous practical applications and studies have revealed several specific technical and economic limitations (Q. B. Li, Huang, Liang, Fu, & Peng, 2016; Peng, Peng, & Li, 2015). First, poorly optimized PV/T-GSHP systems often suffer from continuous long-term soil temperature drops in cold climates, which can severely degrade the heat pump's coefficient of performance (COP) over a 10- to 20-year operational lifespan. Second, many existing designs exhibit suboptimal economic feasibility, often resulting in a high Levelized Cost of Energy

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(LCOE) due to mismatched thermal grading between the solar thermal output and the building's dynamic heating load. Most critically, while individual components have been optimized, previous research has largely overlooked the impact of the physical coupling topology. It remains unclear whether integrating the PV/T array on the source side versus the load side of the heat pump better mitigates these specific long-term performance and economic degradation issues. While highly complex or mixed-coupling architectures exist, the source-side and load-side integrations represent the two fundamental, mutually exclusive boundary conditions for hybrid PV/T-GSHP systems. Isolating and comparing these two foundational topologies is a prerequisite step. It unequivocally reveals how the physical coupling location inherently dictates the thermodynamic efficiency limits, long term soil thermal response, and economic feasibility, thereby establishing a critical baseline for any future design of more complex multi-source systems. To address these specific limitations, this study proposes two PV/T-GSHP systems, with PV/T arranged on the source side and load side respectively, namely the PV/T (source side)-GSHP system and the PV/T (load side)-GSHP system. A comparative study on the performance of the two systems is conducted from the perspectives of energy, economy, environment, and soil thermal balance.

Recent literature has increasingly emphasized the critical role of dynamic operation and control strategies in optimizing PV/T-GSHP systems. For instance, Li *et al.* (H. Q. Li *et al.*, 2026) compared various intelligent control strategies, highlighting that dynamic control based on real-time solar availability significantly improves seasonal performance. Similarly, Li *et al.* (T. Li *et al.*, 2024) proposed an advanced control approach for solar-assisted ground source heat pumps, demonstrating enhanced energy efficiency during dynamic load fluctuations. Xia *et al.* (Xia, Ma, Kokogiannakis, Wang, & Gong, 2018) further investigated dynamic operational switching strategies to minimize heat pump power consumption. However, these advanced optimization studies primarily focus on complex algorithmic controls for a single, fixed system architecture. There remains a notable lack of critical comparison regarding how different physical topologies (i.e., source-side vs. load-side coupling) inherently respond to these dynamic control logics under long-term operation.

While previous studies have extensively investigated the feasibility of hybrid PV/T-GSHP systems, most research has been confined to the optimization of a single system architecture. There remains a critical research gap regarding the direct, long-term comparative analysis of different integration topologies. To address this, this study distinctly compares the source-side and load-side integration of PV/T modules. The core scientific contributions of this work include the topological configuration modeling, in which source-side and load-side PV/T-GSHP integrations are proposed and mathematically modeled to comprehensively evaluate their respective operational dynamics and control strategies. This work also performs a long-term soil thermal balance assessment by conducting a robust 20-year dynamic simulation to quantitatively compare how different PV/T coupling locations mitigate continuous soil temperature decline and achieve seasonal thermal storage. Furthermore, a comprehensive techno-economic analysis is provided with an in-depth comparative assessment of thermodynamic parameters (PER and EER) alongside economic indicators (LCOE and ATCSR) to establish definitive guidelines for the optimal design of low-carbon building heating systems.

2. Modeling of the PV/T-GSHP system

2.1 Composition of the PV/T-GSHP system

The PV/T-GSHP system (Abu-Rumman, Hamdan, & Ayadi, 2020; Zhao, Chen, & Liu, 2023) is mainly composed of PV/T modules, water-source heat pumps, buried ground pipes, heat exchangers, and water tanks. They are primarily used to provide space heating and domestic hot water (DHW) for buildings. During system operation, the PV/T modules absorb solar radiation heat and convert solar energy into electricity simultaneously. In the heating season, the GSHP extracts heat from the ground through vertical buried pipes, exchanges heat with the hot water at the PV/T outlet and uses the combined heat for heat pump heating. The surplus heat is stored in the water tank to supply hot water and enhance the heating performance of the heat pump system. The PV/T modules capture solar energy to generate thermal energy, which compensates for the soil heat consumption during the heating

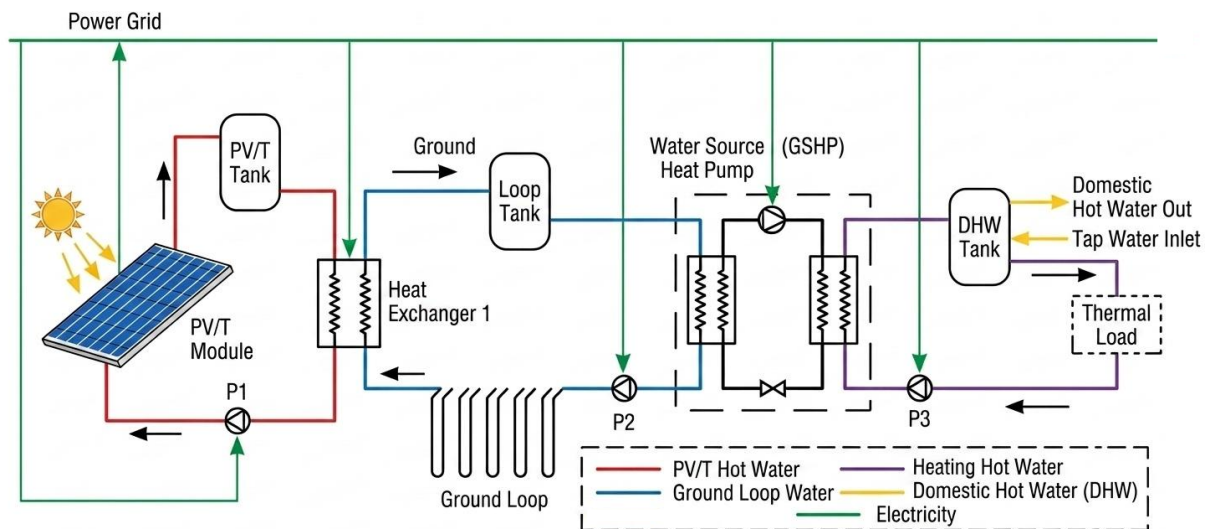


Fig. 1 Schematic diagram of the PV/T (source side)-GSHP system.

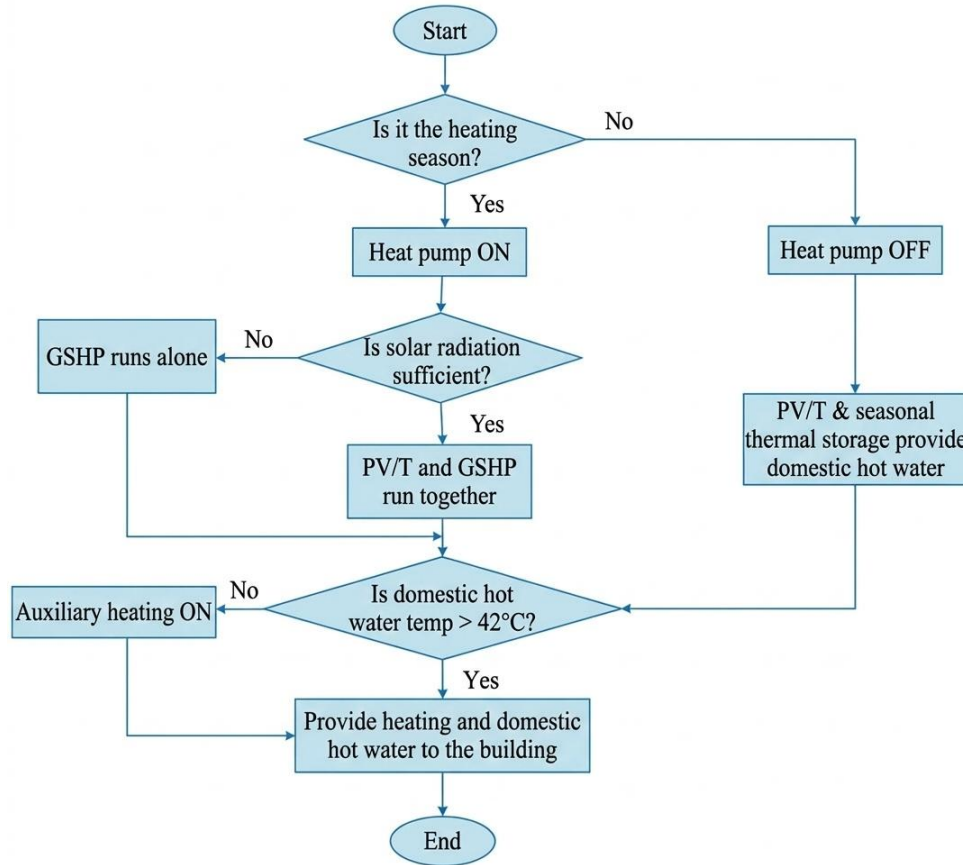


Fig. 2 Control strategy of the PV/T (Source Side)-GSHP system.

season. In the non-heating season, the thermal energy generated by the PV/T modules is transferred to the soil via circulation pumps to address the issue of soil thermal imbalance. By storing solar heat and injecting it into the soil to replenish heat, the system also provides hot water, achieving long-term thermal balance of the soil and seasonal storage of solar energy.

To eliminate descriptive redundancy, it is important to establish the operational baseline shared by both configurations during the non-heating season. From spring through autumn, space heating demands cease. Consequently, the heat pump compressor remains completely deactivated to minimize systemic energy consumption. During this extensive period, the system relies exclusively on the PV/T arrays. The solar heat generated is prioritized to fulfill the daily DHW load, with all surplus thermal energy continuously injected into the buried ground pipes. This unified non-heating strategy effectively executes seasonal solar thermal storage and rebalances the soil temperature deficit. The differentiating operational logic between the source-side and load-side topologies is therefore exclusively activated during the winter heating season, as detailed below.

Figure 1 shows the PV/T (source side)-GSHP system, in which water in the buried ground pipes enters the heat pump after exchanging heat with the hot water produced by the PV/T modules via a heat exchanger. Consequently, the inlet temperature at the heat source side of the heat pump is increased, which further improves the operating efficiency of the heat pump and significantly enhances the heating performance of the entire heat pump system. The control strategy of the PV/T (source side)-GSHP system is illustrated in Figure 2. The heat pump unit adjusts its operation according to

the building's thermal load demand. During the heating season, the heat pump is activated, and the system operates in two modes: (1) GSHP standalone mode: When solar radiation is insufficient, the GSHP alone is responsible for space heating. (2) PV/T-GSHP combined heating mode: When solar radiation is sufficient, PV/T and GSHP operate collaboratively to meet heating demand. If the temperature of the supplied DHW fails to reach 42 °C, auxiliary heating is activated to compensate for the remaining DHW requirement. Once the DHW temperature meets the 42 °C threshold, the system simultaneously satisfies the building's thermal load and hot water demand.

The PV/T (load side)-GSHP system is shown in Figure 3. When the heat generated by PV/T is insufficient to reach the set heating temperature, this portion of heat is stored underground via a heat exchanger to reduce energy consumption, thereby improving the system's energy efficiency. In this scenario, the return water from the thermal load is directly heated by the PV/T system, which reduces the electricity consumption of the heat pump. Furthermore, during the non-heating season, the heat produced by the PV/T heat pump is injected into the ground through buried pipes, which helps mitigate the issue of soil temperature imbalance and ensures the stability and efficiency of the system throughout its annual operation.

The operation strategy of the PV/T (load side)-GSHP system is illustrated in Figure 4. In the heating season, to ensure optimal dynamic performance, the system's operational modes are governed by specific algorithmic control logic. The switching threshold relies heavily on the real-time PV/T outlet temperature ($T_{pv,out}$) and the solar irradiance (G):

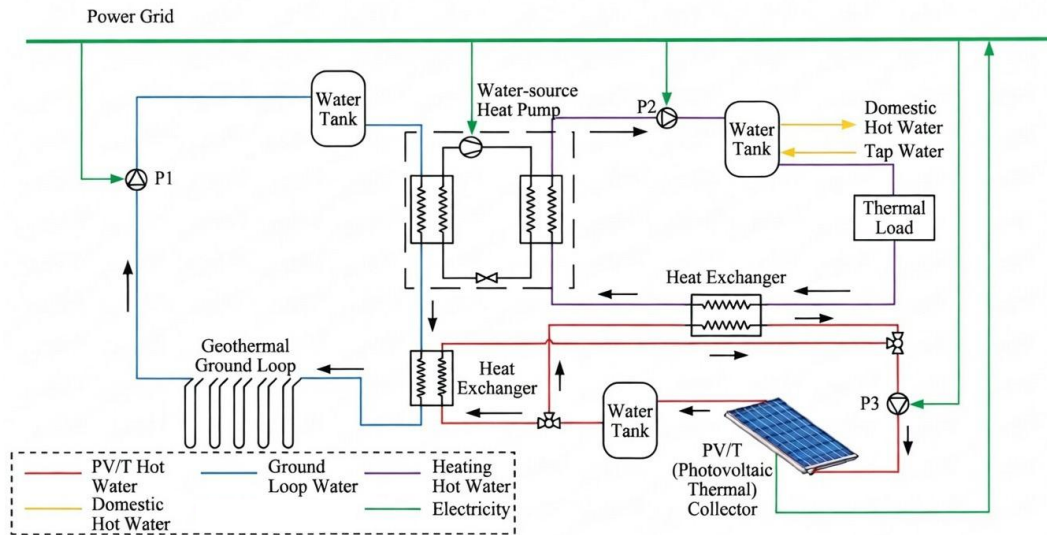


Fig. 3 Schematic diagram of the PV/T (load side)-GSHP system.

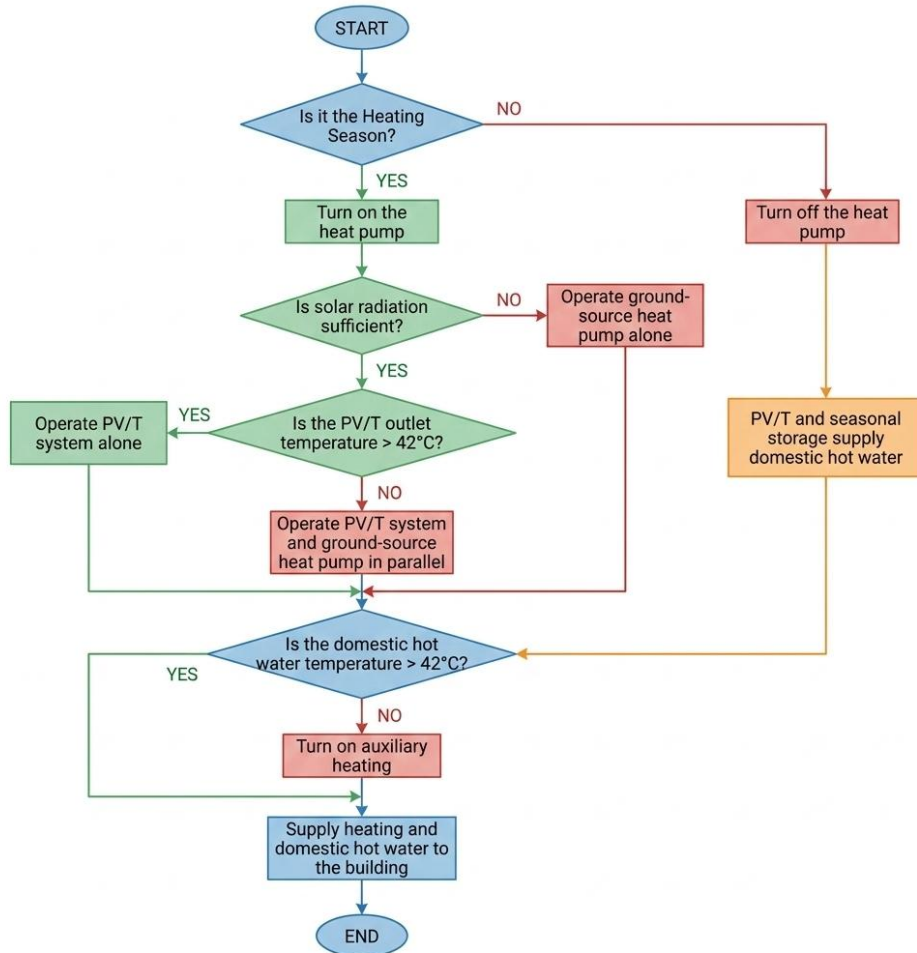


Fig. 4 Operation strategy of the PV/T (Load Side)-GSHP system.

- 1) PV/T standalone heating mode: Activated strictly when $G > 300 \text{ W/m}^2$ and $T_{pv,out} \geq 42^\circ\text{C}$. In this state, the thermal quality of the solar array is sufficient to directly meet the load, allowing the heat pump compressor to completely shut down to save electricity.
- 2) PV/T-GSHP combined heating mode: Triggered when solar radiation is present but $T_{pv,out} < 42^\circ\text{C}$. The

PV/T array preheats the return water, and the GSHP is activated to elevate the fluid to the required supply temperature, maintaining a high system EER.

- 3) GSHP standalone heating mode: Initiated when $G < 100 \text{ W/m}^2$ (e.g., at night or heavily overcast days). The heat pump operates solely by extracting heat from the ground heat exchanger, which has been thermally recharged during the non-heating season.

Auxiliary heating logic: An electric auxiliary heater acts as the final safety redundancy. It is logically triggered only if the combined output of the active modes fails to maintain the domestic hot water tank temperature above the 42°C threshold for more than 10 consecutive minutes.

Compared to complex predictive algorithms heavily reliant on extensive forecasting data, the rule-based dynamic control strategies proposed in this study (as illustrated in Figures 2 and 4) offer high operational reliability and engineering feasibility. By dynamically switching operation modes based on real-time solar radiation and specific domestic hot water temperature thresholds, the system effectively balances seasonal thermal storage in the non-heating season with optimal solar-geothermal synergistic heating in winter. This strategy ensures that the unique advantages of both source-side and load-side configurations are fully realized under dynamic conditions.

2.2 Mathematical model of equipment for the PV/T-GSHP system

To ensure the rigorous reproducibility of the dynamic simulation (conducted with a 1-hour time step using Meteonorm weather data), the operating conditions, boundary parameters, and fundamental assumptions were explicitly defined based on the target building model:

- a) Building Characteristics and Heating Load: The simulated target is a 3-story staff dormitory with a living area of 1458 m^2 and 51 rooms. The building envelope parameters strictly adhere to energy efficiency standards, with average heat transfer coefficients set to $\leq 0.55 \text{ W/(m}^2\cdot^\circ\text{C)}$ for the roof, $\leq 0.6 \text{ W/(m}^2\cdot^\circ\text{C)}$ for outer walls, and $\leq 2.2 \text{ W/(m}^2\cdot^\circ\text{C)}$ for windows. Under these conditions, the annual peak space heating load occurs in January, reaching 33.84 kW .
- b) Dynamic Load and Occupancy Fluctuations: To accurately capture the transient performance of the system, the space heating load is continuously computed based on real-time hourly indoor-outdoor temperature differentials. The domestic hot water (DHW) demand utilizes a dynamic fractional profile rather than a static baseline. This profile strictly mirrors realistic building occupancy schedules, integrating stochastic water draw events that peak during high-occupancy residential hours (08:00–09:00 and 21:00–22:00) and drop significantly during daytime working hours. This transient, 1-hour step simulation ensures that the heat pump and PV/T arrays dynamically respond to genuine operational load variations.
- c) Key Component Parameters: To establish a reliable comparative baseline, the specific equipment parameters were standardized. For the source-side and load-side PV/T-GSHP systems respectively, the PV/T array areas are set to 460 m^2 and 490 m^2 (with an absorptance of 0.98 and a heat exchange factor of 0.96). The rated heating capacities of the heat pumps

are 50 kW (source-side) and 63 kW (load-side). The ground heat exchanger features boreholes with a depth of 100 m , utilizing U-tubes with an inner/outer diameter of $0.016 \text{ m} / 0.032 \text{ m}$ and a grout material thermal conductivity of $3.5 \text{ W/(m}\cdot^\circ\text{C)}$. It must be emphasized that while parameters such as borehole spacing, depth, and thermal storage tank volume are critical to absolute system performance, they were not treated as optimization variables in this study. Instead, they were strictly held constant as control variables across both configurations. This methodological choice was deliberately made to eliminate confounding factors, ensuring that the observed comparative differences in thermodynamic and economic performance are exclusively attributable to the distinct physical integration topologies.

- d) Soil and Fluid Assumptions: The soil surrounding the ground heat exchanger is assumed to be a homogeneous and isotropic medium. The thermophysical properties of the soil are considered constant throughout the 20-year simulation period, and groundwater advection is neglected. The fluid flow within the heat exchangers and pipes is assumed to be one-dimensional and incompressible, with heat losses from the insulated connecting pipes to the ambient environment neglected.

(1) PV/T model

While the PV/T module (Ventura, Tina, Gagliano, & Aneli, 2021) generates electricity from solar radiation using the photovoltaic component, it simultaneously recovers waste heat from the photovoltaic module via a heat exchanger and converts it into thermal energy, thereby achieving the dual utilization of energy. The PV/T module captures solar irradiation, and the buried ground pipes store heat in the soil during the non-heating season for space heating in winter. As a high-efficiency solar energy utilization technology, the PV/T module not only improves energy efficiency but also contributes to the green and low-carbon transition of the energy structure. The electrical efficiency of the PV/T module is given as follows.

$$\eta_{el} = \frac{P}{\tau_g G A_{PV}} \quad (1)$$

Where, G : Solar irradiance (kW/m^2), P : Electrical power (kW), τ_g : Transmittance of the glass cover on the cell surface, A_{PV} : Area of the PV cell (m^2).

The heat production rate and thermal efficiency of the PV/T module are calculated as follows:

$$\dot{Q} = c_p \dot{m} (T_{out} - T_{in}) \quad (2)$$

$$\eta_{th} = \frac{\dot{Q}}{\dot{E}x_{solar,in}} = \frac{\dot{Q}}{\tau_g G A_{PV}} \quad (3)$$

Where, c_p : Specific heat capacity of the cooling fluid ($\text{kJ/(kg}\cdot^\circ\text{K)}$), $\dot{m}$: Mass flow rate of the cooling water (kg/s), T_{out} : Outlet temperature of the cooling fluid (K), T_{in} : Inlet temperature of the cooling fluid (K).

Therefore, the total efficiency of the PV/T module is calculated as follows:

$$\eta_{PV/T} = \eta_{el} + \eta_{th} \quad (4)$$

This total efficiency parameter not only reflects the instantaneous energy conversion capability of the PV/T module but also serves as a critical variable in determining the sizing of the solar array required to meet the dynamic thermal load of the building.

(2) Water Source Heat Pump Model

The water source heat pump system uses groundwater, surface water, or reclaimed water as the heat source or sink for the heat pump. By extracting low-grade thermal energy from the groundwater source and delivering it to the high-temperature heat source system driven by electric power, the water source heat pump realizes the transfer and conversion of energy, thereby achieving the heating function.

The heat exchange capacity of the evaporator in the water source heat pump (kW) is calculated as follows:

$$\dot{Q}_E = \dot{m}_E c_p (T_{E,in} - T_{E,out}) \quad (5)$$

Where, $\dot{m}_E$: Mass flow rate of the fluid in the evaporator (kg/s), $T_{E,in}$: Inlet temperature of the fluid in the evaporator (K), $T_{E,out}$: Outlet temperature of the fluid in the evaporator (K).

The heat exchange capacity of the condenser (kW) is calculated as follows:

$$\dot{Q}_C = \dot{m}_C c_p (T_{C,out} - T_{C,in}) \quad (6)$$

Where, $\dot{m}_C$: Mass flow rate of the fluid in the condenser (kg/s), $T_{C,out}$: Inlet temperature of the fluid in the condenser (K), $T_{C,in}$: Outlet temperature of the fluid in the condenser (K).

The coefficient of performance (COP) is commonly used to evaluate the operating performance of the water source heat pump, and its calculation formula is as follows:

$$COP_{HP} = \frac{\dot{Q}_C}{\dot{W}_{HP}} = \frac{\dot{Q}_C}{\dot{Q}_C - \dot{Q}_E} \quad (7)$$

Where, $\dot{W}_{HP}$: Heat pump energy consumption (kW).

(3) Ground Heat Exchanger Model

The ground heat exchanger (GHE) is a key component of the ground-source heat pump (GSHP) system. In this study, a double-U type GHE is adopted, and the underground pipe loop is arranged in series, which yields high heat transfer performance. The pipe diameter is determined to maintain the minimum pumping power and ensure the heat transfer efficiency between the fluid and the inner pipe wall. Therefore, the outer diameter of the selected GHE is 0.032 m, and the inner diameter is 0.016 m. To prevent shutdown of the heat pump unit and avoid accelerated aging of the GHE due to excessive temperatures (which would reduce its heat exchange efficiency), the design minimum and maximum inlet temperatures of the GHE during the heating season are set to 0°C and 33°C, respectively. The borehole is assumed to be embedded in a cylindrical heat storage body, and its volume is calculated as follows:

$$V = \pi N_b H (0.525B)^2 \quad (8)$$

The borehole thermal resistance within the heat storage body is expressed as follows:

$$R_b = \frac{\bar{T}_f - \bar{T}_b}{q} \quad (9)$$

Where, N_b : Number of boreholes, H : Borehole depth (m), B : Borehole spacing (m), $\bar{T}_f$: Average temperature inside the ground heat exchanger (°C), $\bar{T}_b$: Average temperature of the borehole wall (°C), q : Heat exchange rate per unit length of the ground heat exchanger (kW).

The accurate calculation of this borehole thermal resistance is crucial, as it directly dictates the required length of the ground heat exchanger and practically limits the maximum rate at which solar thermal energy can be injected into the soil during the non-heating season.

The initial parameters of the model include the length of the ground heat exchanger and the number of boreholes. The maximum heat rejection and maximum heat absorption of the heat pump system are calculated by the following equations.

$$Q' = Q \times \left(1 + \frac{1}{COP}\right) \quad (10)$$

Where, Q' : Maximum heat absorption rate of the heat pump (kW), Q : Maximum design heating load of the building (kW).

Based on the heat exchange rate per unit length of the ground heat exchanger in winter, the required length of the ground heat exchanger is estimated using the following formula:

$$L' = \frac{Q' \times 1000}{q} \quad (11)$$

$$N = \frac{L'}{nl} \quad (12)$$

Where, q : Heat absorption rate per unit length of the ground heat exchanger (W/m), N : Number of boreholes, n : Number of U-tubes per borehole ($n=2$ for single U-tube, $n=4$ for double U-tube) l : Borehole depth (m).

(4) Water Tank Model

The thermal storage water tank is modeled using a standard uniform mixing energy balance approach. The dynamic heat exchange is determined by the temperature difference between the mixed fluid inside the tank and the inlet water, ensuring the conservation of energy during both the thermal storage and release processes.

(5) Water Pump Model

The water pumps ensure fluid circulation within the PV/T and GSHP loops. Their operating power and shaft power are calculated based on the fluid mass flow rate, required system pressure drop, and standard motor efficiencies, with the heat dissipation from the motor appropriately accounted for in the fluid's outlet temperature.

(6) Energy and heat balance

For the PV/T-GSHP system, when the electricity generated by PV/T meets the system's electrical load, all the electrical load of the system is supplied by PV/T. When the electricity generated is insufficient to meet the system's electrical demand, the deficit is supplemented by the power grid.

$$E_{sys} = \begin{cases} E_{grid} - E_{PV/T}, & E_{sys} > E_{PV/T} \\ E_{PV/T}, & E_{sys} < E_{PV/T} \end{cases} \quad (13)$$

During the heating season, the user's thermal load is satisfied by the combined heat supply from PV/T and GSHP. In the non-heating season, since the heat pump is not in operation, the heat generated by PV/T is used to supply the user's DHW (Domestic

Hot Water) load. At night, when PV/T cannot generate heat due to the absence of solar radiation, the DHW load is supplied by the heat stored in the ground heat exchanger during the heating season.

$$\begin{cases} Q_{PV/T} + Q_{GSHP} = Q_{Heating} + Q_{DHW}, & \text{In heating season} \\ Q_{PV/T} = Q_{DHW}, & \text{In nonheating season} \end{cases} \quad (14)$$

3. Performance evaluation indicators for PV/T-GSHP system

3.1. System Energy Evaluation Indicators

In the system energy analysis, the indicator η_{sys} is introduced, which covers the efficiencies of energy in all links such as development, conversion, and utilization. The energy efficiency is expressed as follows.

$$\eta_{sys} = \frac{P + Q_h + Q_{DHW}}{W_{con} + En_{solar,in}} \quad (15)$$

The calculation formulas for the heating load and DHW load of the system (kW) are as follows:

$$Q_h = C_p m (T_{out} - T_{in}) \quad (16)$$

$$Q_{DHW} = K_h \frac{n q_r C (T_r - T_1) \rho_r}{t_1} C_r \quad (17)$$

Incoming solar radiation intensity (kW) is as follows:

$$En_{solar,in} = \tau_g G A_{pv} \quad (18)$$

Where, W_{con} : System power consumption (kW), P : Power generation of PV/T (kW), C_p : Specific heat capacity of refrigerant (kJ/(kg·K)), m : Refrigerant mass flow rate (kg/s), T_{out} : Outlet temperature of refrigerant (K), T_{in} : Inlet temperature of refrigerant (K), K_h : Hourly variation coefficient, n : Number of people, q_r : Domestic hot water quota per capita per day (m³/h), C : Specific heat capacity of water (kJ/(kg·K)), T_r : Hot water temperature (K), T_1 : Cold water temperature (K), ρ_r : Density of hot water (kg/m³), t_1 : Daily operating hours (h), C_r : Heat loss coefficient of hot water system, $C_r = 1.1 - 1.15$, τ_g : Transmittance of the glass cover plate on the photovoltaic cell surface, G : Solar irradiance intensity received by the photovoltaic cell (kW/m²), A_{pv} : Area of the PV/T collector (m²).

The total energy consumption of the system includes the energy consumption of the heat pump during the heating season and the energy consumption of the three water pumps operating throughout the year, as follows:

$$W_{con} = W_{HP} + W_{pump1} + W_{pump2} + W_{pump3} \quad (19)$$

Where, W_{HP} : Power consumption of the heat pump (kW), W_{pump1} , W_{pump2} , W_{pump3} : Power consumption of heat pump 1, 2, and 3 respectively (kW).

Selecting EER (Energy Efficiency Ratio) as the thermodynamic performance indicator for the two comparison systems is an important parameter. EER is defined as the ratio of available heat output to the required operating energy. A higher EER value indicates that the equipment or system is more efficient in energy utilization.

$$EER = \frac{Q_h + Q_{DHW}}{W_{con}} \quad (20)$$

PER (Primary Energy Ratio), also referred to as the system's total energy utilization efficiency, is the ratio of the system's output energy to input energy, i.e., the ratio between the actual energy obtained and the primary energy consumed to acquire this energy. As a key indicator for measuring energy utilization efficiency, it reflects the conversion efficiency and energy losses associated with energy from its primary form to its final utilization. The calculation formula is as follows

$$PER = \frac{P + Q_h + Q_{DHW}}{W_{con}/\eta_e} \quad (21)$$

Where, η_e : Power generation efficiency of thermal power plants, taken as 30%.

3.2. System Economic Evaluation Indicators

LCC refers to all costs incurred by a system throughout its entire life cycle, from design, manufacturing, and operation to decommissioning. For the PV/T-GSHP heating system, it includes investment cost (IC), operating cost (OC), and maintenance cost (MC). Within the full life cycle, they can be expressed as:

$$LCC = IC + OC + MC \quad (22)$$

$$IC = CC \cdot r(1+r)^t / [(1+r)^t - 1] \quad (23)$$

$$OC = Y_e W_{con} \quad (24)$$

Where, CC : Capital cost of equipment and installation, t : Operating years, r : Real discount rate, taken as 4.9%, Y_e : Unit energy consumption cost (yuan/kWh).

Assuming that all components of the system have a useful life of 20 years without the need for partial replacement, using the LCOE (Levelized Cost of Energy) to measure the 20-year LCC of the system per unit of power generation is a useful evaluation criterion.

$$LCOE = \left(IC + \sum_{t=1}^y \frac{OC}{(1+r)^t} \right) / \sum_{t=1}^n \frac{P}{(1+r)^t} \quad (25)$$

ATCSR is an indicator that measures the cost-saving benefits achieved by a system through various cost-reduction measures over its life cycle. A higher ATCSR value indicates that the system performs better in cost control and achieves more efficient resource utilization.

$$ATCSR = 1 - \frac{LCC_{PV/T(source\ side)-GSHP}}{LCC_{PV/T(load\ side)-GSHP}} \quad (26)$$

To ensure complete transparency in the economic evaluation, the precise initial capital cost assumptions for all major system components are detailed in Table 1.

Furthermore, to isolate the economic differences between the two configurations without introducing speculative financial variables, the life-cycle analysis utilizes a constant real discount rate of 4.90%. The unit energy consumption cost (C_e) and annual maintenance costs are treated as constant baseline values throughout the 20-year operational lifespan, intentionally excluding dynamic inflation rates or fluctuating electricity tariffs to maintain a straightforward comparative baseline.

Table 1
Initial capital cost assumptions for system components.

Component	Unit Price (RMB)	Unit
PV/T modules	680.34	/m ²
Water source heat pump	45000.00	/unit
Ground heat exchanger	75816.06	/set
Heat exchanger	1000.00	/unit
Water pump	150.00	/unit
Water tank	325.00	/m ³

3.3. System Environmental Evaluation Indicators

CO₂ is considered the primary pollutant from heating systems. To assess the environmental impact of the system, an analysis was conducted on its CO₂ emission reduction (kg). The calculation formula for the system's CDER (Carbon Dioxide Emission Reduction) is as follows:

$$CDER_{PV/T-GSHP} = f_{CO_2} \left(\frac{Q_h}{\eta_{CFB}} - \frac{W_{con,PV/T-GSHP} - P_{PV/T-GSHP}}{\eta_e \eta_{net}} \right) \quad (27)$$

When $P_{PV/T-GSHP} > W_{con,PV/T-GSHP}$, CDER of PV/T-GSHP is 0.

Where, η_{CFB} : Thermal efficiency of the coal-fired boiler heating system, taken as 0.8, η_e : Power generation efficiency of the coal-fired power plant, taken as 0.3, η_{net} : Power transmission efficiency of the power grid, taken as 0.92, f_{CO_2} : CO₂ emission factor of standard coal, taken as 2.439 kg/kgce.

3.4 Soil Thermal Balance Evaluation Indicators

Soil thermal balance is a critical factor affecting the feasibility of GSHP systems. TIR is an indicator used to describe the degree of imbalance between heat absorption and heat release during the soil heat exchange process. It reflects the heat accumulation or depletion of the soil within the operation

period. The calculation formula for this ratio is generally expressed as:

$$TIR = \frac{Q_{ext} - Q_{inj}}{\max(Q_{ext}, Q_{inj})} \cdot 100\% \quad (28)$$

Where, Q_{ext} : Total heat extracted from the borehole (kW), Q_{inj} : Total heat injected into the borehole (kW).

A positive TIR indicates that the heat injected into the ground is greater than the heat extracted from the ground. Conversely, a negative TIR means that the heat extracted from the ground exceeds the heat rejected into the ground.

4. Operating characteristics of the PV/T-GSHP system

4.1 Model validation

In a PV/T-GSHP system (Bhadra & Mwesigye, 2025; Choi et al., 2025; Tung, Kumar, Fung, & Leong, 2025; Zafaranchi, Riddell, Chan, Saliba, & Leung, 2025), the main components include PV/T modules, water-source heat pumps (Fu et al., 2025; Hu & Shen, 2025a, 2025b; Ju et al., 2025; Tan et al., 2024; C. L. Wang et al., 2025), ground heat exchangers, and other heat exchangers. Among them, PV/T and GSHP have a significant impact on system performance. By validating the models of

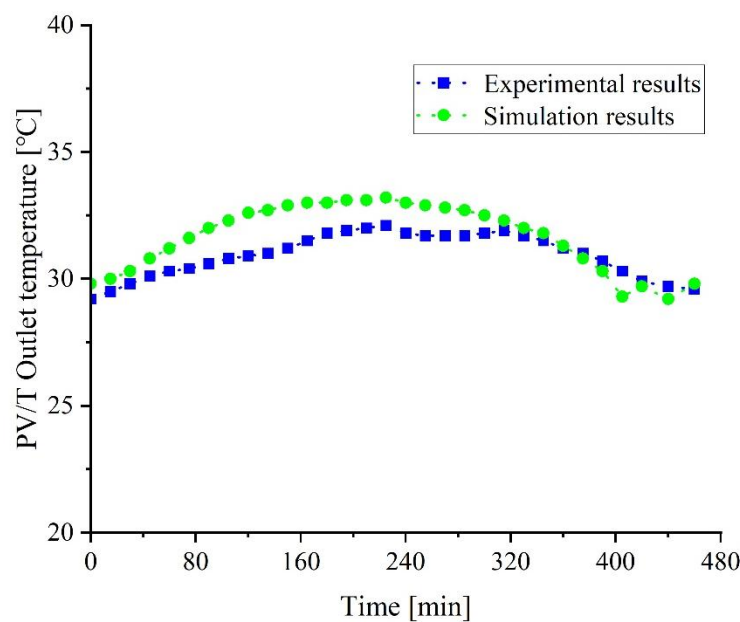


Fig. 5 Comparison of simulation and experimental data for PV/T model outlet temperature.

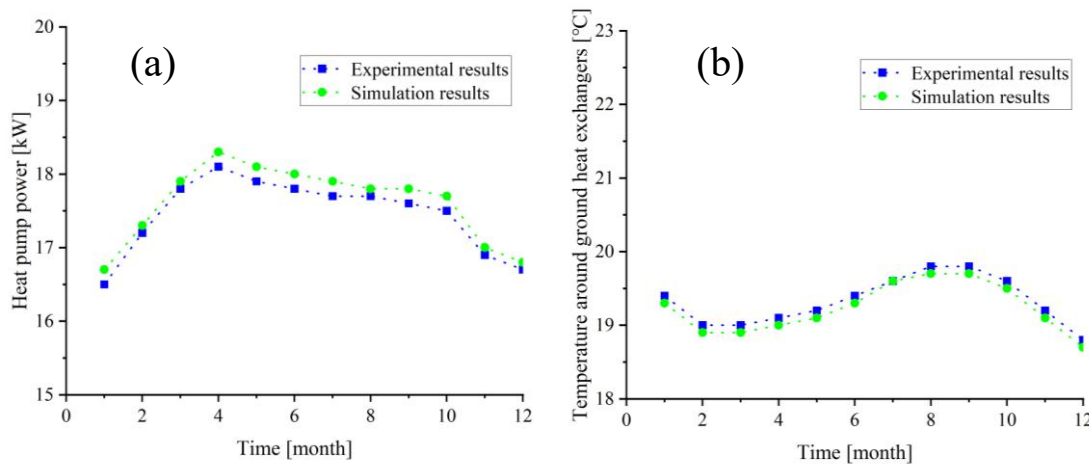


Fig. 6 Comparison of simulation and experimental data for heat pump power (a) and ground heat exchanger surrounding temperature (b).

these two key components, the validation of the entire system model is effectively achieved. The simulation results of the PV/T model and GSHP model are compared with the corresponding experimental data.

To ensure the rigorousness and reproducibility of the experimental validation, an uncertainty analysis was conducted on the measurement instruments. Temperatures were measured using Pt100 RTD sensors with an accuracy of ± 0.15 °C. The fluid mass flow rates were recorded by electromagnetic flowmeters with an uncertainty of ± 0.5 %. Solar irradiance was measured using a first-class pyranometer with an accuracy of ± 2.0 %. Based on the standard Kline and McClintock error propagation method, the maximum overall relative uncertainty for the calculated experimental heat capacity and efficiency was determined to be less than ± 3.5 %, confirming the high reliability of the foundational experimental dataset.

A comparison between the simulation results and experimental results of the PV/T model is shown in Figure 5. The PV/T outlet temperature predicted by the simulation model is in excellent agreement with the experimental results. Statistical analysis reveals a Root Mean Square Error (RMSE) of 1.01 °C and a Mean Absolute Percentage Error (MAPE) of 2.83%, demonstrating a high degree of predictive reliability. Therefore, the PV/T model adopted in the system can be considered reasonable.

It should be noted that while acquiring a continuous 20-year experimental dataset for long-term validation is practically unfeasible, the validation methodology employed here strictly calibrates the fundamental heat transfer coefficients and thermodynamic governing equations using transient experimental data. Because the core physical principles of mass and energy conservation remain invariant over time, these experimentally anchored component models provide a highly reliable theoretical basis for extrapolating and forecasting the long-term (20-year) dynamic operational characteristics and soil thermal responses under varying climatic boundary conditions.

As shown in Figure 6, a comparison is made between the simulation and experimental results of the GSHP model. For the GSHP model, the predicted power consumption and surrounding soil temperature tightly track the experimental measurements. The power consumption prediction yields an RMSE of 0.16 kW and a MAPE of 0.91 %, while the soil temperature prediction shows an RMSE of 0.10 °C and a MAPE of 0.48 %, fully validating the high transient accuracy of the dynamic simulation models. The errors between the simulation

results and the experimental data are small, which verifies the accuracy and reliability of the model proposed in this study. It is acknowledged that the TRNSYS ground heat exchanger model calculates an average bulk soil temperature rather than providing a high-resolution 3D spatial thermal distribution. While spatial mapping offers detailed localized thermal interference patterns, the bulk average temperature approach is widely validated and adopted for long-term (20-year) dynamic macroscopic energy and economic evaluations, offering an optimal balance between computational feasibility and systemic accuracy without compromising the comparative validity of the topologies.

4.2 Comparative analysis of operating characteristics of the PV/T-GSHP system

Before delving into the specific energy indicators, it is crucial to address how these configurations manage seasonal weather variability and high-frequency solar intermittency. Solar energy is inherently intermittent, which can cause severe operational instability in standalone solar heating systems. However, in both the source-side and load-side PV/T-GSHP systems, the massive thermal inertia of the surrounding soil acts as a robust geological thermal battery. High-frequency solar fluctuations (such as intermittent cloud cover during the day) are buffered by the soil and the thermal storage water tank. Furthermore, the rule-based dynamic control strategies seamlessly transition the system between PV/T standalone, combined, and GSHP standalone modes in response to seasonal meteorological shifts. This synergistic operation ensures that the building's thermal demand is consistently met, maintaining long-term thermodynamic stability despite the inherent unpredictability of solar irradiance.

Figure 7(a) shows the annual variation of η_{sys} for the two systems, with similar overall trends: both reach their peaks in the heating season, decline in the transition seasons, and reach their lowest points in summer. Compared with the PV/T (load side)-GSHP system (Conte, Zanetti, Tancon, & Azzolin, 2025; El Kasti & Ozcan, 2025; Mehdipour, Garvey, Baniamerian, & Cardenas, 2025; Mehdipour, Garvey, Cardenas, Baniamerian, & Wood, 2025; Pallotta *et al.*, 2025), the PV/T (source side)-GSHP system generates higher electricity output. In the non-heating season, when the load is small, the system output power is correspondingly lower, resulting in a relatively low system output. Meanwhile, the electricity generation in the heating

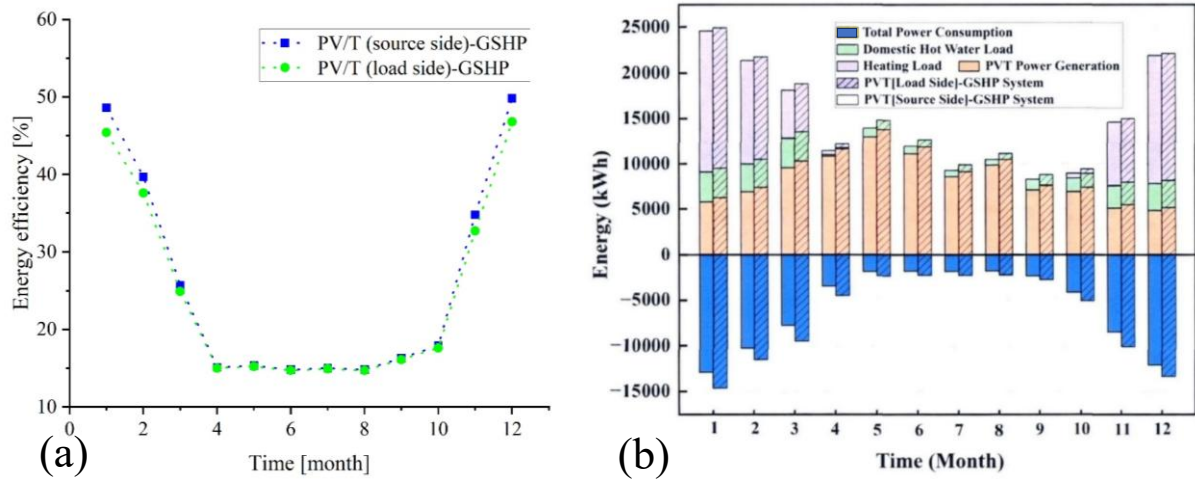


Fig. 7 Annual energy performance of the two configurations: (a) Total energy efficiency highlighting the peak utilization during the winter heating season, and (b) Annual energy variation detailing the electricity generation versus systemic consumption.

season is higher than that in the non-heating season. This is because the energy consumption of the system during the non-heating season is lower than that in the heating season, and there is no thermal load in the non-heating season. In the transition seasons and summer, the per capita hot water consumption decreases, and the solar radiation intensity is high, so the hot water produced by photovoltaic power generation can meet the demand. Therefore, the share of solar energy utilization is close to 15% in the non-heating season and reaches up to 50% in the heating season. Reducing the use of heat pumps not only effectively saves electric energy but also enables more sufficient utilization of solar energy.

In terms of energy utilization, although the energy utilization rate in the non-heating season is relatively low, it is worth noting that the heat pump remains in a non-operating state during this period, which reduces the power consumption of the system. As shown in Figure 7(b), this figure presents the annual variations in electricity generation and consumption of the PV/T (load side)-GSHP system and the PV/T (source side)-GSHP system. In the heating season, due to the high demand

for space heating and domestic hot water (DHW), the heat pump is activated and operates in conjunction with PV/T modules to meet the heating and DHW supply. The system's power consumption includes that of the heat pump and water pumps, leading to the highest total power consumption during this period. In the transition seasons, no space heating is required, so the heat pump remains inactive. The heat generated by PV/T is sufficient to satisfy the DHW demand, thereby reducing the system's total power consumption. In summer, with stronger solar irradiation, PV/T produces more heat and electricity. Meanwhile, the per capita demand for DHW decreases during this period.

Overall, both the output power and power consumption of the PV/T (load side)-GSHP system are higher than those of the PV/T (source side)-GSHP system (Alrasheedi, Chinnaamy, & Prakash, 2025; Di Matteo, Fiorini, Vollaro, Oclon, & Vallati, 2025; Flores *et al.*, 2025; Graf & Cadenbach, 2025; Illan-Gómez, Garcia-Cascales, Velasco, & Otón-Martínez, 2025; Kaufmann *et al.*, 2025). Specifically, during the transition seasons, the PV/T electricity generation reaches its peak in May, with values of

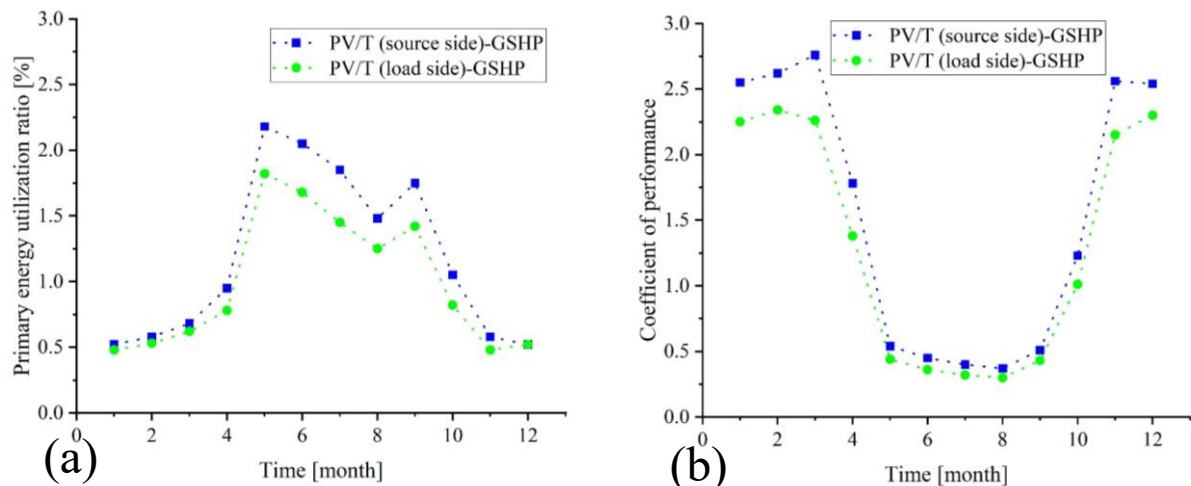


Fig. 8 Thermodynamic performance indicators: (a) Primary Energy Ratio (PER) demonstrating peak values during transition seasons, and (b) Energy Efficiency Ratio (EER) illustrating the performance penalty caused by parasitic pump power during non-heating months.

13791 kWh and 12942 kWh, accounting for 93.2% and 92.8% of the total annual electricity generation of the PV/T (load side)-GSHP system and PV/T (source side)-GSHP system, respectively. In the heating season, the total power consumption of the systems reaches its maximum in January, with values of 14608 kWh and 12887 kWh, respectively.

Figure 8 illustrates the annual variations in EER and PER for the two systems. It can be seen that both systems exhibit the same trend: the PER and EER of the PV/T (source side)-GSHP system are consistently higher than those of the PV/T (load side)-GSHP system. Owing to the increased electricity generation and reduced power consumption of the PV/T modules, the PER remains at a relatively high level from May to August. However, the higher frequency of rainy days in July leads to a slight decrease in PER. Since the heat pump is not activated during non-transition seasons and summer, there is no space heating output. Consequently, the EER of both systems drops significantly to around 0.5 during the transition and summer seasons. The physical mechanism behind this sharp decline is rooted in the ratio of parasitic power to useful thermal output. During non-heating months, space heating is disabled, and the system's useful thermal output is limited strictly to the domestic hot water (DHW) load. Although the absolute thermal demand is quite low, the circulation water pumps must remain operational to capture solar heat and recharge the soil. The parasitic electrical consumption of these pumps becomes disproportionately high relative to the small DHW thermal output. This severe mismatch between the constant auxiliary pumping power and the low useful thermal delivery mathematically drives down the EER, despite the PV/T modules experiencing peak solar irradiance.

The fundamental thermodynamic mechanism driving the superior efficiency of the source-side configuration lies in its direct modulation of the heat pump's evaporation temperature. In thermodynamics, the efficiency of a vapor-compression cycle is highly sensitive to the temperature lift between the evaporator and the condenser. By injecting solar thermal energy directly into the source side, the fluid temperature entering the evaporator is elevated. This higher evaporation temperature significantly reduces the required pressure ratio across the

compressor, thereby drastically lowering the compressor's mechanical work and intrinsically boosting the Coefficient of Performance (COP). Conversely, while the load-side PV/T integration effectively preheats the return water (reducing the overall thermal load the heat pump must satisfy), the heat pump itself is still forced to extract heat from an unheated, colder ground source. Consequently, the compressor operates across a larger temperature lift, resulting in a lower intrinsic thermodynamic efficiency compared to the source-side topology.

The 20-year life cycle cost (LCC) of the PV/T (source side)-GSHP system is generally similar to that of the PV/T (load side)-GSHP system, with higher operating cost (OC) and maintenance cost (MC) but relatively lower initial cost (IC). The LCC of the PV/T (source side)-GSHP system is lower. As shown in Figure 9, the IC and OC of the PV/T (source side)-GSHP system account for 6.28% and 65.05% of its LCC, respectively, while those of the PV/T (load side)-GSHP system account for 5.82% and 67.58% of its LCC, respectively. The PV/T (source side)-GSHP system exhibits better economic performance: it has a higher initial investment but lower OC. The levelized cost of energy (LCOE) of the two systems is 0.36 RMB/kWh and 0.39 RMB/kWh, respectively. To provide practical context, these LCOE values were benchmarked against existing heating solutions. The LCOE of 0.36 RMB/kWh for the source-side configuration is highly competitive, falling significantly below the standard local grid electricity tariff of approximately 0.55 RMB/kWh. Furthermore, it outperforms conventional standalone GSHP systems without PV/T integration, which typically exhibit LCOE values ranging from 0.42 to 0.48 RMB/kWh due to higher continuous power consumption and long-term COP degradation. The LCOE of the PV/T (load side)-GSHP system is relatively higher, which is attributed to its larger power consumption and consequently higher operating cost. The results indicate that, compared with the PV/T (load side)-GSHP system, the annual total cost saving rate (ATCSR) of the PV/T (source side)-GSHP system reaches 11.14%.

Given the inherent volatility of energy markets, a sensitivity analysis was conducted to assess the robustness of the economic feasibility against fluctuating grid electricity

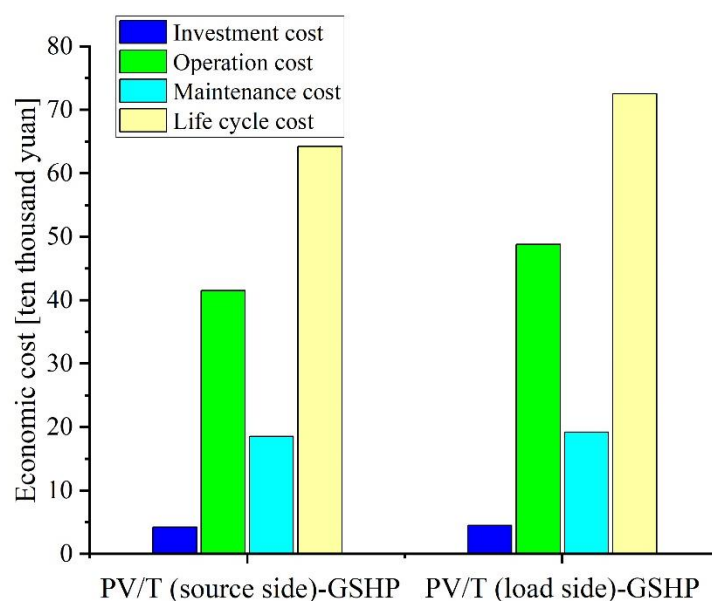


Fig. 9 20-year economic assessment comparing Life Cycle Cost (LCC) and Levelized Cost of Energy (LCOE), demonstrating the long-term financial superiority of the source-side integration despite higher initial capital costs.

tariffs. Assuming the operation cost (OC) is directly proportional to the electricity tariff, when the baseline electricity price increases by 20% (simulating future energy shortages), the Life Cycle Cost (LCC) of the PV/T (source side)-GSHP system rises to 72.50 ten thousand yuan, while the PV/T (load side)-GSHP system rises to 82.26 ten thousand yuan. Under this scenario, the economic advantage of the source-side system becomes even more pronounced, with its Annual Total Cost Saving Rate (ATCSR) increasing to 11.86%. Conversely, even under a scenario where electricity prices drop by 20%, the source-side system maintains a robust ATCSR of 10.90%, proving its strong financial viability and resilience across diverse energy pricing environments.

To provide a realistic annualized economic perspective for the 1458 m² targeted building scale, the 20-year operation costs were annualized using a standard annuity factor based on the 4.9% real discount rate. The corrected equivalent annual energy consumption cost is rigorously calculated as 33.0 thousand yuan for the PV/T (source side)-GSHP system, and 38.8 thousand yuan for the PV/T (load side)-GSHP system. Consequently, the source-side configuration yields a net equivalent annual operational saving of 5.8 thousand yuan over the load-side counterpart. These annualized figures accurately align with the operational scale of a 33.84 kW peak heating load, definitively confirming the superior economic feasibility of the source-side system on a realistic engineering scale. From an economic perspective, it is proven that the PV/T (source side)-

GSHP system is more economically feasible, with a lower LCC and a higher net profit.

The CDER values of the PV/T (source side)-GSHP system and the PV/T (load side)-GSHP system are 55852 kg and 51122 kg, respectively. The CDER of the PV/T (source side)-GSHP system is 8.47% higher than that of the PV/T (load side)-GSHP system, which is attributed to the larger power consumption of the heat pump in the PV/T (source side)-GSHP system. While this study focuses primarily on operational Carbon Dioxide Emission Reduction (CDER), a comprehensive evaluation of environmental sustainability necessitates acknowledging Life Cycle Assessment (LCA). The embodied energy associated with PV/T module manufacturing, deep borehole drilling, and the Global Warming Potential (GWP) of heat pump refrigerants represent initial carbon debts. However, by significantly enhancing the operational COP and maximizing solar utilization, the proposed hybrid topologies are expected to achieve a rapid carbon payback period. This effectively offsets the initial embodied emissions, validating their long-term environmental sustainability despite the lack of a full cradle-to-grave LCA in the current scope. It is important to note that the CDER values calculated above solely represent operational emission reductions. A comprehensive environmental evaluation must also acknowledge broader lifecycle impacts. The manufacturing of PV/T modules and the extensive drilling required for ground heat exchangers incur significant embodied energy and initial carbon footprints. Furthermore, the potential

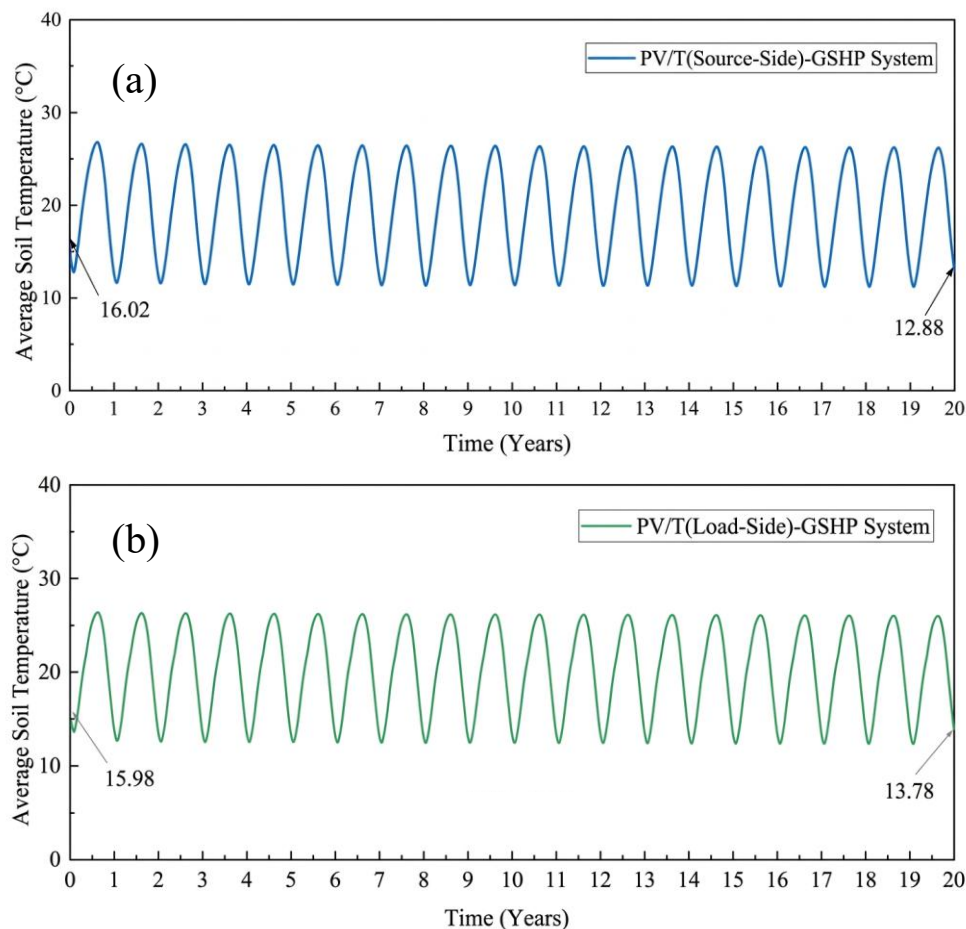


Fig. 10 Simulated 20-year long-term soil thermal response: (a) PV/T (source side)-GSHP system and (b) PV/T (load side)-GSHP system, both exhibiting successful mitigation of severe temperature depletion via seasonal solar thermal injection.

leakage of synthetic refrigerants (e.g., R410A or R134a) utilized in the heat pump carries a high Global Warming Potential (GWP). However, existing lifecycle assessment literature indicates that the substantial operational CO₂ savings of hybrid PV/T-GSHP systems typically achieve a carbon payback period of 3 to 5 years, which effectively offsets these initial embodied emissions and lifecycle risks over the 20-year operational span.

The comparison of the average soil temperatures between the PV/T (source side)-GSHP system and the PV/T (load side)-GSHP system is shown in Figure 10. It can be seen that with the operation of the systems, the average soil temperatures of both systems exhibit the same fluctuating downward trend. By the 20th year, the average soil temperature of the PV/T (source side)-GSHP system decreases from the initial 16.02°C to 12.88°C, with a total drop of 3.14°C and a temperature imbalance ratio (TIR) of 0.20. For the PV/T (load side)-GSHP system, the average soil temperature decreases from 15.98°C to 13.78°C, with a total drop of 2.20°C and a TIR of 0.14. The TIR serves as a critical quantitative metric for operational sustainability. Unlike conventional standalone GSHP systems in cold climates, where the TIR often exhibits severe imbalance leading to system failure or ground freezing, the calculated TIR values of 0.20 and 0.14 indicate a highly manageable thermal deficit. This low imbalance ensures that the fluid temperature entering the heat pump's evaporator remains permanently within the operational safety margins (> 0°C). Consequently, it guarantees that the compressor operates under stable suction pressures over the entire 20-year lifecycle, preventing the long-term COP degradation typical of unbalanced systems. Beyond maintaining the thermodynamic efficiency of the heat pump, mitigating long-term soil thermal degradation is vital for underground ecological preservation. Continuous temperature drops in standalone GSHP systems can lead to localized permafrost, altering groundwater physical chemistry and disrupting subterranean microbial ecosystems. By effectively injecting solar heat and stabilizing the soil temperature drop to a moderate and safe margin over 20 years, the PV/T-GSHP systems prevent extreme thermal depletion, thereby actively safeguarding the surrounding subterranean ecological environment.

Research indicates that by designing distinct PV/T positional configurations and control strategies for these two systems, the respective advantages of each system can be effectively exploited. The PV/T module can absorb solar radiation and accomplish solar heat recharge into the soil. Consequently, the soil temperature is significantly elevated, mitigating the soil temperature decline caused by the long-term continuous operation of the GSHP system. Both systems are capable of achieving soil thermal balance and seasonal solar energy storage.

The results distinctly demonstrate an intrinsic cascade effect from thermodynamic optimization to economic and environmental viability. The source-side configuration's ability to directly elevate the heat pump's evaporation temperature fundamentally minimizes compressor work, physically yielding the higher PER and EER observed. This enhanced thermodynamic efficiency translates directly into a reduction in cumulative operational electricity consumption. Consequently, this lower parasitic energy demand drives the robust economic superiority of the source-side system—manifesting as a highly competitive LCOE of 0.36 RMB/kWh and a significant 11.14% ATCSR—despite its slightly higher initial integration complexity. Ultimately, this minimized reliance on grid electricity naturally culminates in the highest observed CDER (55,852 kg). Therefore, the source-side topology represents a physically

sound, financially viable, and ecologically optimal alignment for low-carbon building applications.

It is essential to contextualize the practical applicability of these proposed topologies across different climatic regions. The superior performance of the PV/T-GSHP configurations demonstrated in this study is inherently tied to cold-climate or heating-dominated environments, where the primary geological challenge is long-term soil thermal depletion. In such regions, the PV/T array's thermal injection effectively rebalances the ground temperature. However, applying these identical configurations in cooling-dominated or tropical climates would be counterproductive. In warm climates, the continuous rejection of building heat already causes soil thermal accumulation; introducing additional solar thermal energy into the ground would severely exacerbate soil overheating and degrade the heat pump's cooling COP. Therefore, the proposed source-side solar-geothermal synergy is strictly recommended for moderate-to-cold latitudes with distinct heating demands.

While the source-side topology demonstrates thermodynamic superiority, its practical engineering implementation entails greater complexity compared to conventional systems. Integrating the PV/T array directly into the source-side hydraulic loop requires additional heat exchangers, anti-corrosion pipelines, and precise variable-frequency circulation pumps. Furthermore, operational reliability is heavily contingent on the automated control valves, which must sustain high-frequency switching between modes. Maintenance challenges also emerge; for instance, the PV/T cooling fluid requires specific antifreeze solutions during severe winters to prevent pipe bursting, and the photovoltaic glass covers demand regular cleaning to prevent efficiency degradation from dust accumulation. These practical factors must be carefully weighed against the long-term energy savings during the initial engineering design phase.

5. Conclusions

This study proposed two PV/T-GSHP systems with PV/T configured on the source side and load side respectively, and conducted a comprehensive comparative analysis of their performance from energy, economic, environmental and soil thermal balance perspectives. The results demonstrated that both systems could achieve soil thermal balance and seasonal solar energy storage, with PV/T modules effectively supplementing soil heat via solar radiation absorption and mitigating soil temperature decline caused by long-term GSHP operation. The PV/T (source side)-GSHP system outperformed the load side counterpart in overall performance: it exhibited higher PER and EER throughout the year, lower 20-year LCC and LCOE (0.36 RMB/kWh vs 0.39 RMB/kWh), a notable annual total cost saving rate of 11.14%, higher annual net profit (a 69 thousand yuan advantage), and an 8.47% higher CDER value. However, it is crucial to objectively acknowledge the limitations of these findings. The demonstrated superiority of the source-side configuration is strictly bound by the specific simulation assumptions utilized in this study, including the idealized constant isotropic soil properties, the fixed 1458 m² building dynamic load profile, and the specific cold-climate meteorological boundaries.

Furthermore, while the individual components were rigorously validated against transient experimental data, the absence of a long-term, full system-scale experimental prototype remains a limitation of this study. The simulated 20-year performance trajectories represent theoretical optimal conditions that do not fully account for potential real-world equipment degradation, unexpected control valve failures, or

extreme inter-annual climatic anomalies. Therefore, while this study establishes a solid thermodynamic baseline proving the strategic advantage of source-side PV/T integration, future research must prioritize the construction of a physical pilot-scale system. Long-term empirical monitoring under real-world occupancy and weather conditions is imperative to ultimately validate the systemic reliability and practical significance of these proposed solar-geothermal hybrid configurations.

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