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Design and Optimization of an Integrated Bathhouse Wastewater-Solar Heat Pump System

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How To Cite: Wang, Y.; Yuan, X.; Au-Yong, C.P.; et al. Design and Optimization of an Integrated Bathhouse Wastewater-Solar Heat Pump System. *Urban and Building Science* **2025**, *1*(1), 3. <https://doi.org/10.53941/ubs.2025.100003>

Received: 11 May 2025

Revised: 12 June 2025

Accepted: 8 July 2025

Published: 14 July 2025

Abstract: Public bathhouse wastewater exhibits significant waste heat recovery potential due to its high-temperature and high-flow characteristics. However, the temporal mismatch between wastewater availability cycles and bathing demand necessitates supplementary heat sources, with solar energy emerging as a critical renewable solution. Current research gaps persist in understanding multi-factor synergy mechanisms and parameter optimization in solar-wastewater source heat pump (WSHP) coupled systems. This study focuses on a university bathhouse in Changchun, China, designing a solar-WSHP hybrid heating system integrated with an air-source heat pump (ASHP) as auxiliary backup, tailored to academic calendar-driven operational cycles and bathing load characteristics. Utilizing the Transient System Simulation Tool (TRNSYS), a dynamic simulation model was developed to analyze annual performance, while a five-level orthogonal experimental design optimized four key parameters: solar collector area, storage tank volume, WSHP heating capacity, and ASHP heating capacity. Results demonstrate that the optimized system not only meets the hot water load demand, but also improves energy performance, with solar heat collection increased by 13.22% and the coefficient of performance (COP) of the wastewater source heat pump improved by 3.16%. In terms of economy, the initial investment of the system is reduced by 11.40%, and the annual operation cost is decreased by 1.80%. The composite system can reduce carbon dioxide emissions by 706.15 tons each year. This study provides theoretical and technical foundations for multi-objective optimization of waste heat recovery systems in public buildings, particularly addressing the operational demands of educational institutions.

Keywords: solar energy; wastewater source heat pump; orthogonal experiment; multifactorial analysis

1. Introduction

The accelerated development of the global economy has intensified energy shortages and environmental degradation, rendering the efficient exploitation of renewable energy technologies crucial for addressing energy crises [1]. Solar energy, as an abundant and inexhaustible clean resource, has garnered significant attention due to its immense potential in building hot water supply systems [2,3]. However, its inherent intermittency and



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spatiotemporal variability (e.g., diurnal radiation fluctuations and seasonal efficiency variations) challenge the reliability of standalone solar thermal systems [4]. In this context, heat pump technology emerges as an ideal supplementary solution, which enables efficient heat transfer from low-temperature sources to higher temperature sinks through the reverse Carnot cycle [5]. The synergistic integration of solar collectors and heat pumps establishes a complementary heating system where solar energy provides baseline thermal load while heat pumps compensate for supply deficiencies during insufficient irradiation periods, thereby enhancing system reliability. Pioneering studies have validated this coupling mechanism. Sterling et al. [6] demonstrated the economic superiority of indirect solar-assisted heat pump systems through life-cycle cost analysis. Yang et al. [7] identified the parallel configuration of solar and air-source heat pumps as exhibiting optimal heating stability and coefficient of performance (COP), consistent with Anderson et al. [8] and Liu et al. [9] findings on multi-source synergy.

Waste heat recovery from public bathhouses presents substantial energy-saving potential [10]. In China, these centralized hot water consumption facilities are widely distributed across educational institutions and industrial complexes [11]. With economic growth and improved living standards, bathing water demand has surged, reaching 780 million cubic meters annually in urban areas [12]. According to the Code for Design of Building Water Supply and Drainage (GB 50015-2019), bathing water typically requires 40–45 °C service temperature, while post-discharge wastewater retains approximately 30 °C thermal energy. This low-contaminant effluent offers cost-effective heat recovery potential through minimal pretreatment [13]. Nevertheless, most public bathhouses currently discharge untreated wastewater into sewage systems, resulting in substantial thermal energy wastage. Wastewater source heat pumps (WSHPs) provide a viable solution by recovering low-grade heat from bathing effluent. Baek et al. [14] achieved an average annual COP of 4.8 using sauna wastewater, meeting full hot water demands except during extreme winter conditions. Kahraman et al. [15] further revealed that WSHP COP increases from 3.36 to 3.69 as wastewater temperature rises from 20 °C to 40 °C, confirming temperature's critical impact on system efficiency. However, WSHP operation necessitates temporary wastewater storage in collection tanks, introducing inherent operational hysteresis between water consumption and heat production phases [16,17]. These temporal constraints compromise the reliability of single-source systems, driving growing interest in dual/multi-source configurations.

Previous studies have explored multi-energy systems for hot water supply, with common configurations employing electric boilers in parallel with solar collectors or air-source heat pumps (ASHPs) for staged heating, where electric heating serves as backup during adverse meteorological conditions [18–20]. An emerging alternative integrates tri-source systems combining solar collectors, ASHPs, and wastewater-source heat pumps (WSHPs) [21]. However, these dual/tri-source configurations lack in-depth discussion on synergistic optimization among subsystems, particularly regarding operational coordination and parameter interdependencies.

In summary, although current research on heat pump-coupled solar water heating systems has achieved some progress, significant limitations remain: (1) Parametric analyses remain predominantly confined to single-variable investigations, lacking systematic exploration of multifactorial synergy mechanisms; (2) Current multi-energy configurations inadequately utilize solar energy, evidenced by insufficient collector operational duration and suboptimal solar conversion efficiency; (3) Existing wastewater heat recovery studies neglect institution-specific operational constraints—particularly the significant reduction in bathhouse operation during academic vacations—failing to address critical temporal impacts on multi-source system performance.

Therefore, this study conducts the following research:

- (1) Taking university bathhouses in Changchun as the research subject, this study analyzes bathing water consumption patterns through questionnaire surveys to calculate annual daily hot water loads. Considering the university-specific operational schedule (including academic breaks) and the demand for high-volume hot water supply within short durations, a schematic design of solar—WSHP hybrid system is developed. Based on the severe cold climate characteristics of the region, an air-source heat pump (ASHP) was selected as the suitable auxiliary heat source.
- (2) A TRNSYS (Transient System Simulation Tool) simulation model of the hybrid system is established to verify its ability to meet annual hot water demands through comprehensive operational simulations.
- (3) Through systematic analysis, four key parameters are selected as influencing factors: the heating capacity of the WSHP, the output of the air-source heat pump (ASHP), the volume of the heat storage tank, and the area of solar collectors. A five-level orthogonal experimental design is implemented using the simulation model, with annualized cost as the evaluation criterion, to determine parameter influence hierarchy and optimize system configuration.
- (4) The optimized system undergoes annual operational simulations to validate its performance under diverse seasonal and meteorological conditions. A comparative analysis with the baseline system is conducted to assess optimization effectiveness.

2. Materials and Methods

2.1. Bathhouse Water Demand Analysis

2.1.1. Bathing Behavior Survey

This study focuses on a public bathhouse serving a university campus in Jingyue Development Zone, Changchun City, Jilin Province, located in a severe cold region of China. The annual solar radiation reaches approximately 5000 MJ/m^2 [22]. In 2024, the average high temperature was 12.3°C and average low temperature 3.16°C (https://tianqi.2345.com/wea_history/54161.htm, accessed on 6 July 2025). The bathing facility originally operated a solar water heating system comprising 746 flat-plate collectors, yielding a total collector area of 1492 m^2 . Despite this capacity, the system remains unable to meet hot water demand and requires supplementation by externally sourced hot water. The bathhouse accommodates over 16,000 full-time students and operates daily from 10:00 to 20:00 in a two-story building. The geographical location, weather conditions, and energy characteristics of the bathing facility are presented in Figure 1. The shower temperature is set to 43°C , and the post-discharge wastewater temperature averaged 36.2°C (measured repeatedly using a TP101 electronic probe thermometer), demonstrating significant waste heat recovery potential. A questionnaire survey was conducted to analyze the bathing patterns of students, including bathing frequency per day, average shower duration, and common bathing time periods during summer, winter, and transitional seasons on both weekdays and weekends. A total of 521 valid questionnaires were collected. Table 1 summarized the survey results of student bathing behavior.

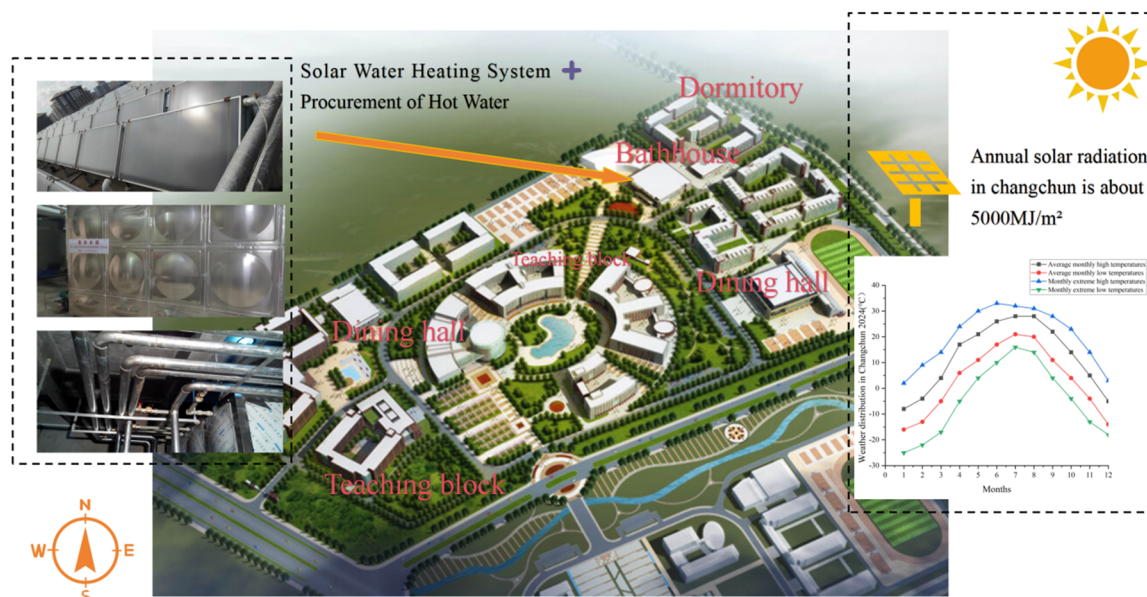


Figure 1. Basic conditions for bathhouses (geography, weather, energy).

Table 1. Survey of bathing situation of students in school.

Season	Weekday Bathing Frequency	Weekend Bathing Frequency	Average Shower Duration (s)	Water Consumption per Capita (L)	Daily Water Consumption—Weekday (L)	Daily Water Consumption—Weekend (L)	Total Campus Population
Summer	2.68	1.06	590	78.6	674,073.6	666,528	16,000
Winter	1.75	0.86	698	93.2	521,920	641,216	16,000
Transitional	2.13	0.93	634	84.5	575,952	628,680	16,000
Summer vacation	1.67		664	88.5	63,340.71		3000

2.1.2. Annual Daily Hot Water Load Calculation

The hybrid system heats tap water, which is subsequently mixed with unheated supply water to meet students' bathing demands. As the temperature of tap water (acting as the cold source) fluctuates with seasonal ambient

variations, these temperature changes must be incorporated into water demand calculations. The monthly average tap water temperature ($T_{i,n}$) in Changchun is determined through Equations (1) and (2) [23]:

$$T_{i,n} = \frac{T_{i,max} + T_{i,min}}{2} - \frac{T_{i,max} - T_{i,min}}{2} \cos\left(2\pi \frac{n-2}{12}\right) \quad (1)$$

$$T_{i,max} = 2T_{i,ave} - T_{i,min} \quad (2)$$

where $T_{i,n}$ is the monthly average cold water temperature for calculation month n ($^{\circ}\text{C}$); $T_{i,max}$ is the warmest monthly average cold water temperature; $T_{i,min}$ is the coldest monthly average cold water temperature; n is the calculation month index (1–12); $T_{i,ave}$ is the annual average cold water temperature.

The annual daily hot water load (Q) is calculated via Equation (3) [24]:

$$Q = \frac{C_p(t_e - t_{i,n})M\rho}{3600} \quad (3)$$

where Q is the daily hot water load (kW); C_p is the constant-pressure specific heat of water ($\text{kJ}/(\text{kg}\cdot^{\circ}\text{C})$); ρ is the water density (kg/L); t_e is the shower setpoint temperature ($^{\circ}\text{C}$); M is the daily water consumption (L).

2.2. Hybrid System Configuration Design

2.2.1. Solar Collector Area Calculation

Given the university's location in a severe cold region, an indirect solar thermal collection system is adopted. The required collector area is determined through Equations (4) and (5) [25]:

$$A_{IN} = A_C \left(1 + \frac{F_R U_L A_C}{U_{IN} A_{hx}}\right) \quad (4)$$

$$A_C = \frac{Q_w \rho_w C_w (t_{end} - t_0) f}{J_T \eta_{cd} (1 - \eta_L)} \quad (5)$$

where A_{IN} is the total area of indirect solar collectors (m^2); A_C is the collector area for direct solar system (m^2); F_{RUL} is the total heat loss coefficient of indirect collectors ($\text{W}/(\text{m}^2\cdot^{\circ}\text{C})$); U_{hx} is the heat exchanger coefficient ($\text{W}/(\text{m}^2\cdot^{\circ}\text{C})$); A_{hx} is the heat exchanger area (m^2); Q_w is the daily water consumption (L); ρ_w is the water density (kg/L); C_w is the water specific heat at constant pressure ($\text{kJ}/(\text{kg}\cdot^{\circ}\text{C})$); t_{end} is the design terminal temperature of storage tank ($^{\circ}\text{C}$); t_0 is the design cold water temperature of storage tank ($^{\circ}\text{C}$); J_T is the solar irradiance in collector plane ($\text{kJ}/(\text{m}^2\cdot\text{d})$); f is the solar fraction; η_{cd} is the average collector efficiency; η_L is the thermal loss rate of piping.

2.2.2. Heat Storage Tank Calculation

During the operation of the heat storage tank, lower-temperature water exits from the bottom of the tank through a circulation pump, enters the solar collector for heating, and then re-enters the tank from the top. This study adopts a single-tank system where all solar-heated water is stored in the heat storage tank to meet the hot water demand of the campus bathhouse. Therefore, the volume of the heat storage tank must consider both the solar collector area and the bathhouse water load. The volume of the heat storage tank is calculated using Equation (6):

$$V = A_{IN} \times B_1 \quad (6)$$

where V is the volume of the heat storage tank (m^3); B_1 is the daily hot water production per unit collector area ($\text{L}/(\text{m}^2\cdot\text{d})$).

2.2.3. Wastewater Source Heat Pump (WSHP)

In the WSHP system, post-bathing wastewater is first filtered to remove suspended solids such as hair, then enters the evaporator where the refrigerant absorbs heat from the wastewater. The recoverable heat is calculated using Equations (7) and (8) [26]:

$$Q = \frac{\mu \rho C G_0 \Delta t}{3600} \quad (7)$$

$$Q_0 = \frac{Q}{T_0} \quad (8)$$

where Q is the waste heat recovery load from bathing wastewater (kW); μ is the wastewater collection coefficient; ρ is the wastewater density (kg/L); C is the specific heat capacity of wastewater (kJ/(kg·°C)); T_0 is the operating time of the WSHP (h); Δt is the temperature difference of the wastewater before and after passing through the heat pump unit (°C); G_0 is the volume of bathing wastewater (L).

2.2.4. Solar Water Heating System Circulation Pump

In solar water heating systems, circulation modes are categorized into forced circulation and natural circulation based on operational principles. Natural circulation relies on water level gradients to drive hot and cold water flow without external energy input, while forced circulation requires pump pressurization to overcome insufficient hydraulic head. The solar subsystem in this study employs a pressurized configuration, necessitating a forced circulation design with dedicated circulation pumps. The pump flow rate is determined by Equation (9):

$$q_x = A_{IN} \times q_z \quad (9)$$

where q_x is the circulation pump flow rate (L/s); q_z is the working fluid flow rate per unit collector area (L/(s·m²² of collectors, q_z is selected as 0.060 L/(s·m²) following industry standards for pressurized solar thermal installations.

2.2.5 Collector Tilt Angle

In practical engineering, there are generally three principles for determining the optimal inclination angle of the solar energy: maximizing the total annual solar radiation, minimizing the annual auxiliary heating amount, and the modified formula of latitude. For practical engineering with a large hot water load, the size of the total annual solar radiation is mainly considered, with the total heat gain as the main factor. The total solar radiation incident on the tilted surface of the collector, denoted as H_T , is equivalent to the sum of the beam radiation component (H_{bT}), the sky-diffuse radiation component (H_{dT}), and the ground-reflected radiation component ($H_{\rho T}$), as expressed by Equation (10):

$$H_T = H_{bT} + H_{\rho T} + H_{dT} = H_b R_b + H_d R_d + \rho H R_\rho \quad (10)$$

where H_b is the direct solar radiation on the horizontal plane; R_b is the ratio of the radiation on the inclined surface outside the atmosphere to that on the horizontal plane; H_d is the scattered solar radiation on the horizontal plane; R_d is the ratio of the scattered sky radiation to the radiation on the horizontal plane; ρ is the surface reflectivity; H is the total solar radiation on the horizontal plane; R_ρ is the ratio of the ground reflected radiation to the radiation on the horizontal plane.

2.3. Solar-Wastewater Source Heat Pump Coupled System

The integrated system combines three subsystems: a solar hot water system, a WSHP system, and an ASHP system. These subsystems are interconnected through pumps and piping networks to heat water in the storage tank for bathing demands. The solar thermal subsystem includes solar collectors, a plate heat exchanger, circulation pumps, and heat exchanger pumps; the WSHP subsystem consists of a wastewater source heat pump, filters, and circulation pumps; the ASHP subsystem comprises an air-source heat pump and circulation pumps. The schematic diagram of the hybrid system is presented in Figure 2.

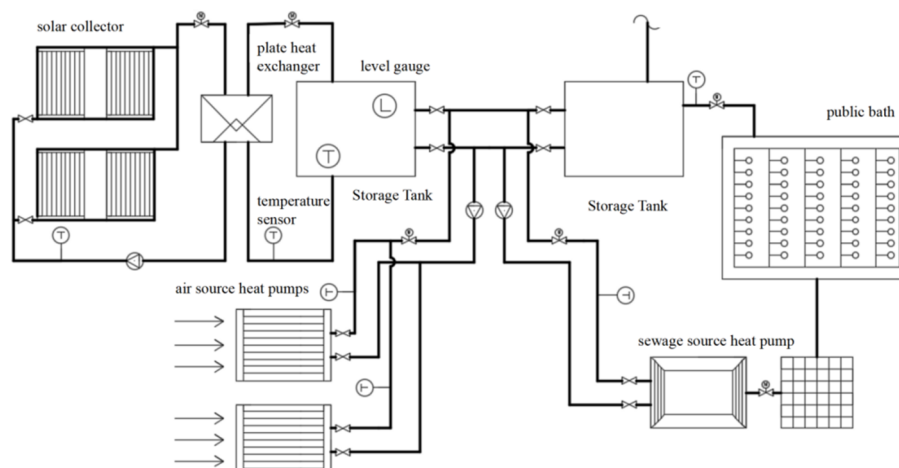


Figure 2. Schematic diagram of the solar-wastewater source heat pump hybrid system.

Operational Control Strategy for Hybrid System

To optimize the utilization of solar energy resources and achieve coordinated operation among multiple heat sources, this study employs both temperature differential control and temperature-setpoint control strategies for the activation and deactivation of the system's three heat sources. This is achieved by monitoring the collector outlet temperature and the temperature at the top layer of the solar storage tank. The control strategy encompasses the following three aspects:

- (1) **Solar Water Heating System:** The solar water heating system is activated and operated via temperature differential control. The solar collector pump initiates operation when the temperature difference between the solar collector and the storage tank exceeds 8 °C. Conversely, the pump ceases operation when this temperature difference falls below 2 °C.
- (2) **Water-Source Heat Pump (WSHP) System:** The operational time window for the WSHP system is set daily from 11:00 to 22:00. Within this window, the WSHP is activated one hour after the bathhouse opens, provided the sewage flow rate and temperature have stabilized and the temperature at the top layer of the storage tank is below 49 °C. The WSHP is deactivated once the temperature at the top layer of the storage tank reaches 58 °C [27,28] (indicating that the solar system alone can meet the bathhouse hot water demand at this point).
- (3) **Air-Source Heat Pump (ASHP) System:** The operational time window for the ASHP system is set daily from 16:00 to 22:00. Within this window, activation is governed by a temperature-setpoint control: the ASHP starts only if the WSHP system is already operating and the temperature at the top layer of the storage tank remains below 45 °C. The ASHP is deactivated when the temperature at the top layer of the storage tank exceeds 58 °C (indicating that the combined operation of the solar system and the WSHP can meet the bathhouse hot water demand).

The system adopts a unified terminal temperature setpoint (58 °C), simplifying control logic and avoiding safety risks caused by frequent heat pump cycling. By monitoring only two critical temperature points, it effectively manages the heating operation of all three heat sources, resulting in a control strategy that is clearer, easier to implement, and simpler to debug [29].

2.4. Orthogonal Experimental Optimization

2.4.1. Range Analysis

This study employs orthogonal experimental design to optimize the constructed hybrid system. Orthogonal experimental design, proposed by Japanese mathematician Genichi Taguchi, is a multi-factor, multi-level optimization method rooted in Galois theory. This approach selects representative parameter combinations from full factorial experiments to identify optimal solutions while minimizing experimental iterations and workload, ensuring accuracy and efficiency.

The range analysis method (R-method) evaluates experimental data through Equations (11) and (12) [30]:

$$\overline{K_{jm}} = \frac{K_{jm}}{S} \quad (11)$$

$$R_j = \max(\overline{K_{j1}}, \overline{K_{j2}}, \overline{K_{jm}}) - \min(\overline{K_{j1}}, \overline{K_{j2}}, \overline{K_{jm}}) \quad (12)$$

where K_{jm} is the sum of experimental results for level m in column j ; S is the number of factor levels; $\overline{K_{jm}}$ is the average value of K_{jm} ; R_j is the range of factor j . The range R_j reflects the influence magnitude of factor j on the target variable, with larger values indicating higher significance. Factor prioritization is determined by descending R_j values.

2.4.2 Annualized Cost Evaluation

The dynamic annualized cost method evaluates hybrid system economics by converting capital investments into equivalent annual costs using a recovery factor, combined with annual operational expenses. This approach applies to scenarios with identical/diverse service lifetimes. Annualized cost (A_C) is calculated via Equation (13) [31]:

$$A_C = C_i \left[\frac{i(1+i)^n}{(1+i)^n - 1} \right] + C_k \quad (13)$$

where A_C is the annualized cost; C_i is the initial investment; C_k is the annual operational cost; i is the recovery factor; n is the service life.

3. Annual Operational Simulation Analysis of Hybrid System

3.1 TRNSYS Simulation Model Development

Based on the system schematic (Figure 1) and control logic, a comprehensive simulation model of the solar-wastewater source hybrid energy system was developed using the TRNSYS platform. The model prioritizes synergistic utilization of solar and wastewater heat, incorporating key modules such as solar collectors, a heat storage tank, circulation pumps, a WSHP, and an auxiliary ASHP. Critical parameters for each module were detailed in Table 2, with the solar collector tilt angle optimized according to local latitude.

Table 2. TRNSYS model parameter settings for composite systems.

Component	Value	Unit
Solar collector area	2000	m ²
Solar collector tilt angle	40	°
Collector fluid specific heat	3.57	kJ/kg·°C
Heat storage tank volume	120	m ³
Tank heat loss coefficient	0.4	W/m ³ ·K
WSHP heating capacity	1667.35	kW
Fluid specific heat capacity	4.19	kJ/kg·K
ASHP heating capacity	1280.94	kW
Solar pump design flow rate	108	m ³ /h
Plate exchanger pump flow rate	108	m ³ /h

The TRNSYS modeling strictly adheres to the system schematic and control logic through modular programming, as structurally detailed in Figure 3. First, the solar collector subsystem was constructed using Type 109 modules, integrating local meteorological data (solar irradiance, ambient temperature) from the weather database. Second, the WSHP was modeled via Type 609 modules, incorporating real-time wastewater temperature data for dynamic simulation. Finally, subsystems were interconnected through piping and control logic modules to form the integrated simulation model.

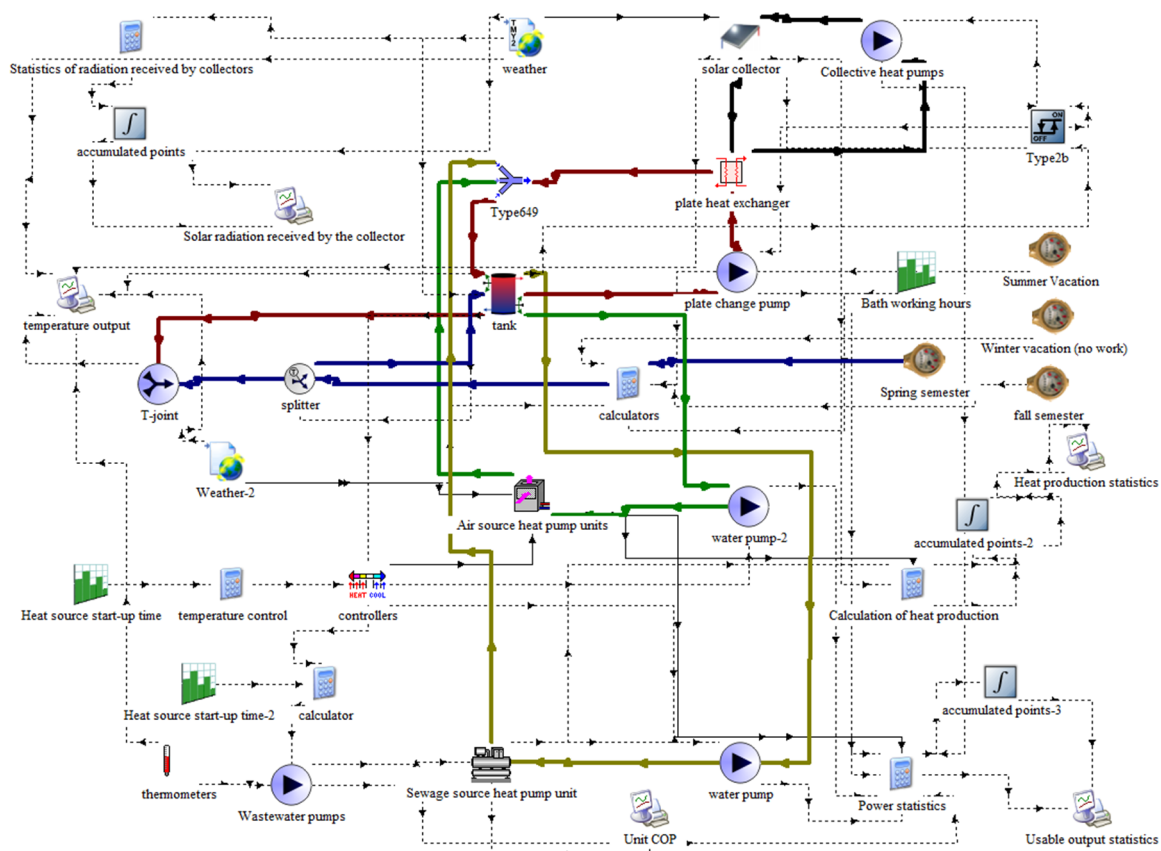


Figure 3. TRNSYS simulation model diagram.

3.2. Model Description and Validation

This study adopts the solar water heating system design presented in Figure 3. The parameter configuration is based on Kalogirou's setup [32], including the use of local Nicosia weather data, a 2.7 m² solar collector, and a 150 L storage tank. The tank temperature was selected as the output variable for validation against Kalogirou's experimental measurements. Comparative results were shown in Table 3, where a mean percentage difference of 2.66% between simulated and measured values demonstrated the model's reliability.

Table 3. Comparison of Measured and Simulated Results.

Test Day	Actual Data (Tank Temperature)	Modelled Data (Tank Temperature)	Absolute % Difference
360 h	49.0	45.0	8.16%
384 h	48.7	48.3	0.82%
408 h	42.1	41.0	2.61%
420 h	42.4	41.8	1.42%
1248 h	49.5	49.9	0.81%
1272 h	54.2	56.8	4.80%
1296 h	49.7	52.3	5.23%
1752 h	55.2	56.4	2.17%
1776 h	57.4	58.1	1.22%
1800 h	56.8	57.9	1.94%
1824 h	54.4	57.6	5.88%
1848 h	56.5	57.5	1.77%
2400 h	67.2	67.3	0.15%
2424 h	67.8	66.8	1.47%
2448 h	66.7	66.3	0.60%
2472 h	64.4	67.7	5.12%
2496 h	65.4	65.1	0.46%
4056 h	64.2	65.9	2.65%
4080 h	61.7	64.7	4.86%
4104 h	63.6	63.1	0.79%
4128 h	60.7	63.0	3.79%
4152 h	67.4	65.8	2.37%
8520 h	43.7	44.9	2.75%
8544 h	48.5	46.8	3.51%
8568 h	44.8	45.1	0.67%
Mean percentage difference			2.64%

3.3. Annual Operation of Hybrid System

This study conducted annual dynamic simulation analysis of a multi-source composite system, comprehensively evaluating its operational characteristics and energy supply capabilities under varying climatic conditions and hot water supply demands throughout the year. In the TRNSYS simulation, the Type15-2 weather module was precisely integrated with the Type1b solar collector module. The weather module dynamically incorporates local meteorological data-including solar irradiance, ambient temperature, and wind speed-to drive the dynamic performance simulation of the solar collector. Built-in calculator modules standardized units to accurately output critical parameters such as collector thermal power and annual effective heat collection. The simulation spanned a full year (0–8760 h) with a 0.25-h timestep, balancing computational accuracy and resource efficiency. Post-simulation analysis provided the annual thermal energy and energy output of the multi-source composite system, as shown in Figure 3.

Analysis of Figure 4a revealed pronounced seasonal fluctuations in solar heat collection. Summer exhibited peak efficiency due to high solar irradiance and extended daylight hours, with May reaching the annual maximum of 108,245 kWh. During the spring semester (March–June), monthly solar heat collection averaged 96,241.6 kWh, while autumn semester values (September–December) decreased to 69,603 kWh owing to lower solar altitude angles and increased cloud cover. Notably, reduced solar radiation in winter coincided with the students' winter vacation period when bathhouse hot water supply was unnecessary, aligning perfectly with campus bathhouse demand patterns. This supply-demand synchronization highlighted the thermo-economic synergy of solar systems as stable auxiliary heat sources, effectively reducing off-peak energy consumption.

Figure 4b,c demonstrated climate- and control-driven operational patterns of water-source heat pump (WSHP) and air-source heat pump (ASHP). The heating power of the WSHP during spring and autumn transitional

seasons significantly exceeded summer levels. This occurs because lower ambient temperatures and solar irradiance in transitional seasons reduce solar heating efficiency, requiring the WSHP to shoulder a greater portion of the hot water load compared to summer. Conversely, the ASHP operates far less frequently than the WSHP, dictated by the composite system's control strategy: The ASHP only runs when the temperature at the top layer of the heat collection tank is below 45 °C. It deactivates when this temperature exceeds 58 °C (indicating the WSHP alone can meet bathhouse demand). Consequently, the ASHP bore no hot water load during summer when favorable weather conditions yielded high solar irradiance. Figure 4c further demonstrated that during summer vacation, reduced hot water demand resulted in ASHP remaining inactive for extended periods. However, near 4893 h, abrupt ASHP activation occurred frequently due to meteorological variations: a sudden decrease in solar thermal output during this period caused insufficient heating supply from both solar and WSHP systems, thus triggering ASHP activation.

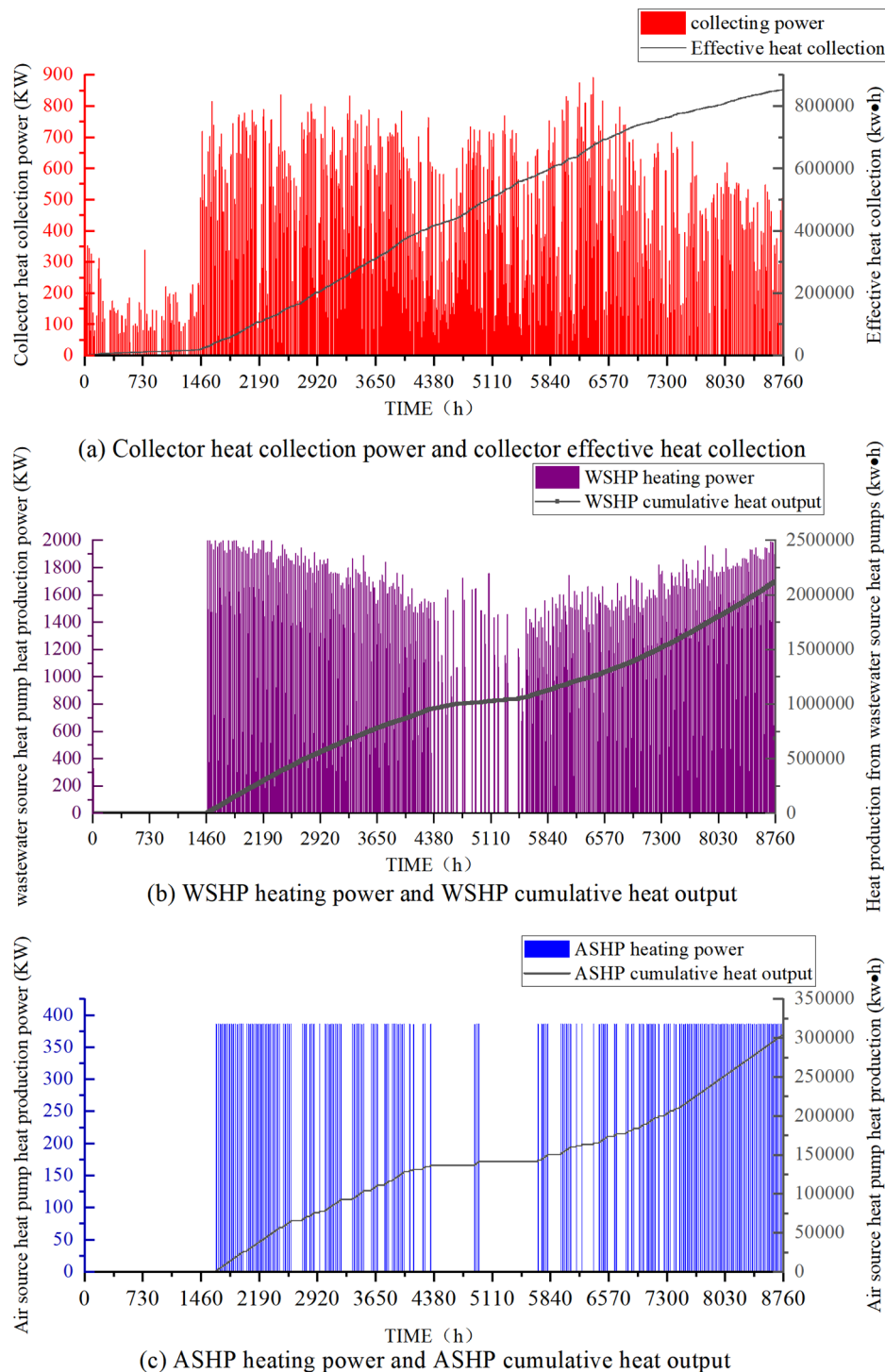


Figure 4. Annual thermal energy and energy output of the multi-source composite system.

3.4. Annual Heat Production of Hybrid System

To visually analyze the annual heat production of the hybrid system, bar charts were generated to display monthly contributions from solar collectors, WSHP, and ASHP, as shown in Figure 5.

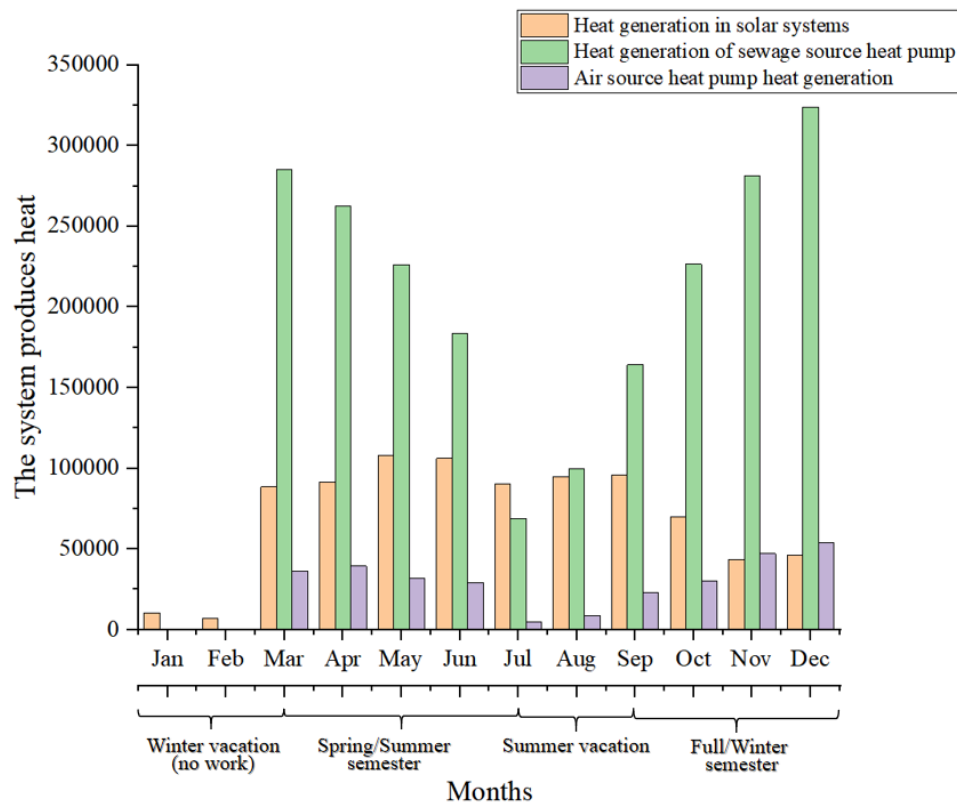


Figure 5. Annual heat output distribution of the composite system.

Figure 5 illustrated the monthly heat production shares of the three heat sources. Due to the university's winter vacation in January and February, when the bathhouse was closed, both the WSHP and ASHP systems were not in operation. Annually, WSHP accounted for 65.70% of total heat production, solar contributed 26.00%, and ASHP provided 9.30%, aligning with the system's design philosophy of prioritizing wastewater heat recovery.

Monthly variations showed solar heat production peaks from May to September, while WSHP and ASHP outputs inversely correlated with solar availability. This pattern stems from control logic: when solar collection suffices, WSHP heating load decreases, and ASHP operation ceases entirely.

3.5. Hot Water Supply Reliability Analysis

The hybrid system underwent annual operational analysis through TRNSYS simulations. During modeling, the outlet temperature parameter from the Type4c stratified heat storage tank module and Changchun's ambient temperature data from the Type109-TMY2 weather module were connected to the output terminal of the Type65a data recorder module. Simulation parameters were configured with a duration of 0–8760 h and a timestep of 0.25 h. Post-simulation, the outlet temperature profile of the hybrid system was plotted as shown in Figure 6.

In the simulation, the tank outlet temperature represents the actual supply temperature for bathhouse use. Figure 6 demonstrated that the annual outlet temperature remained above 43 °C. During periods 0–1440 h and 4344–5558 h, corresponding to the university's winter and summer vacations, the WSHP and ASHP systems operated at reduced capacity or were suspended. Consequently, the tank outlet temperature was slightly higher-consistent with expectations. Critically, all outlet temperatures exceeded 43 °C, confirming the system met bathhouse hot water requirements. The average outlet temperature of approximately 50 °C also revealed significant energy-saving potential.

A thermostatic valve was installed at the bathhouse point of use to prevent overheating. Annual operational results were exported and plotted as the temperature at the point of use (Figure 6). As seen in Figure 7, usage temperatures predominantly clustered around 44 °C, fully satisfying bathhouse demands.

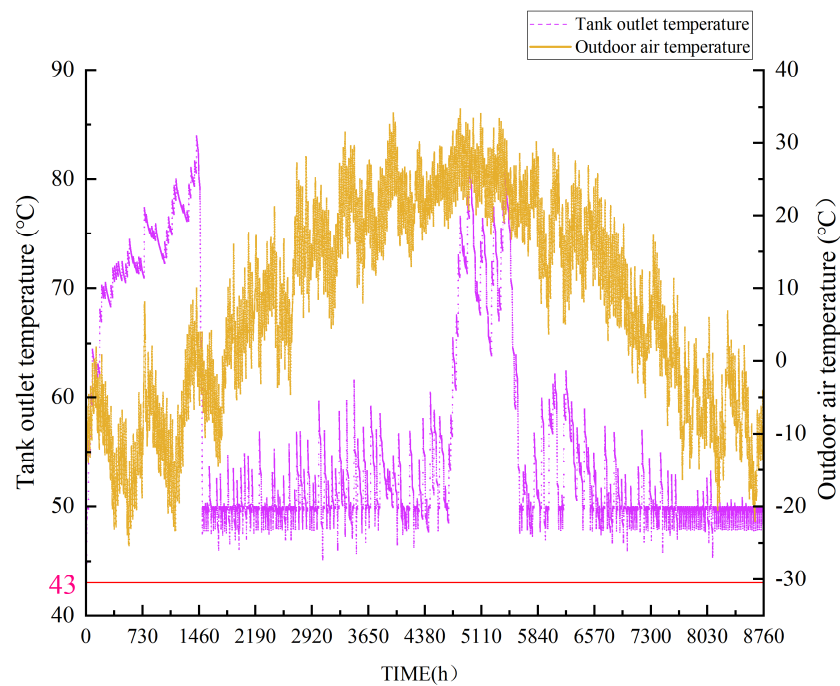


Figure 6. Storage tank outlet temperature versus ambient air temperature.

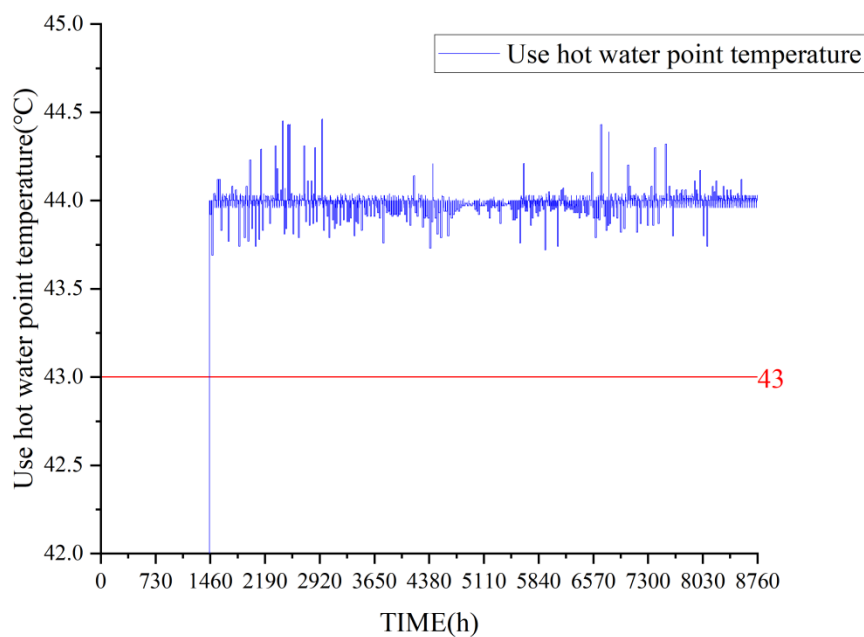


Figure 7. Use hot water point temperature.

3.6. The Key Challenges

Annual simulations conducted via TRNSYS confirm the integrated system's capability to fully satisfy the university bathhouse's hot water demand. However, under current parameters, Figure 6 shows tank outlet temperatures predominantly clustered around 50 °C while Figure 7 indicates bathing water temperatures concentrated near 44 °C, revealing significant optimization potential. The system design must therefore balance fulfilling thermal requirements with minimizing initial investment and operational costs. Given the integrated system's composition-incorporating solar thermal, sewage-source heat pump (WSHP), and air-source heat pump (ASHP) subsystems-synergistic interaction among multiple factors necessitates optimization. Orthogonal array testing provides an effective experimental methodology for this multi-factor, multi-level collaborative optimization, enabling parameter prioritization and identification of optimal configurations.

4. System Optimization Design

4.1. Orthogonal Experiment Factors and Level Selection

In optimizing the solar-heat pump hybrid system, previous research has identified several key factors, including solar collector area and storage tank volume [33]. Building upon this foundation, this study incorporates the heating capacities of the Wastewater source heat pump (WSHP) and the Air-Source Heat Pump (ASHP) as two additional experimental factors, resulting in four factors in total. Given that the system is located in Changchun, which falls under Solar Resource Region III, and adhering to the recommended solar fraction range of 30% to 50% specified in the Technical standard for solar water heating system of civil buildings (GB 50364-2018) [22], this orthogonal experiment selected five distinct solar fraction levels (30%, 35%, 40%, 45%, 50%). Using Equations (4) to (8), the selection calculation for the solar collector area, storage tank volume, WSHP heating capacity, and ASHP heating capacity were performed to ensure the selected variables remain within reasonable bounds. The four experimental factors and their corresponding five levels are detailed in Table 4.

Table 4. Table of Factor Levels.

Level	Solar Fraction	Collector Area (m ²)	Tank Volume (m ³)	WSHP Heating Capacity (kW)	ASHP Heating Capacity (kW)
		A	B	C	D
1	30%	1714	102	1839.68	970.22
2	35%	2000	120	2152.42	1135.16
3	40%	2285	137	2518.33	1328.14
4	45%	2571	154.3	2946.45	1553.92
5	50%	2857	170	3447.34	1818.09

4.2. Orthogonal Experiment

The L25(5⁴) orthogonal array was employed to conduct 25 experimental runs. Annual energy consumption was determined through TRNSYS simulations, and dynamic annualized costs were calculated via Equation (10) to evaluate economic impacts. Range analysis results are presented in Table 5.

Table 5. Range Analysis Results.

Column (Ten Thousand Yuan)	Solar Collector Area	Heat Storage Tank Volume	WSHP Heating Capacity	ASHP Heating Capacity
K1	309.60	312.63	286.57	293.70
K2	304.27	306.55	293.59	297.06
K3	303.90	302.97	300.97	303.47
K4	303.02	301.47	309.85	310.56
K5	303.53	301.13	329.68	301.13
$\overline{K1}$	61.92	62.526	57.314	58.74
$\overline{K2}$	60.854	61.31	58.718	59.412
$\overline{K3}$	60.78	60.594	60.194	60.674
$\overline{K4}$	60.604	60.294	61.97	62.112
$\overline{K5}$	60.706	60.226	65.936	60.226
R	1.316	2.3	8.622	3.372

A detailed range analysis of Table 4 provided critical insights into the economic impacts of key factors on the hybrid system. Among the four parameters—solar collector area, heat storage tank volume, WSHP heating capacity, and air-source heat pump (ASHP) heating capacity—the WSHP heating capacity exhibited the most significant influence, with $R = 8.622$. This indicated that WSHP output played a pivotal role in system economics. As the core energy conversion component, WSHP heating capacity directly determines energy consumption and operational costs. Higher WSHP output enhanced low-grade thermal energy utilization from wastewater, reducing reliance on auxiliary sources like ASHP and lowering costs. Conversely, insufficient WSHP output necessitated greater ASHP engagement, increasing expenses.

The ASHP heating capacity ranked second in economic impact, with $R = 3.372$. As a backup heat source, appropriately sized ASHP minimizes energy use during solar/WSHP shortfalls. However, improper sizing leads to energy waste and cost escalation.

While solar collector area (range: 2.3) and tank volume (range: 1.316) showed smaller impacts, their roles remain vital. Collector area governs solar harvesting efficiency, reducing auxiliary energy demand, while tank

volume affects thermal storage capacity, optimizing heat pump runtime. Their relatively lower influence stemmed from the dominance of heat pump dynamics in operational economics.

Prioritized by annualized cost criteria, the factor hierarchy was: WSHP heating capacity > ASHP heating capacity > tank volume > collector area, clarifying their relative economic significance.

The factor-level versus performance metric plot (Figure 8) visually reinforced these findings, illustrating annualized cost trends across parameter variations. This visualization validated the range analysis and guided targeted optimizations, such as recalibrating WSHP-ASHP capacity ratios or adjusting storage thresholds to mitigate thermal oversupply.

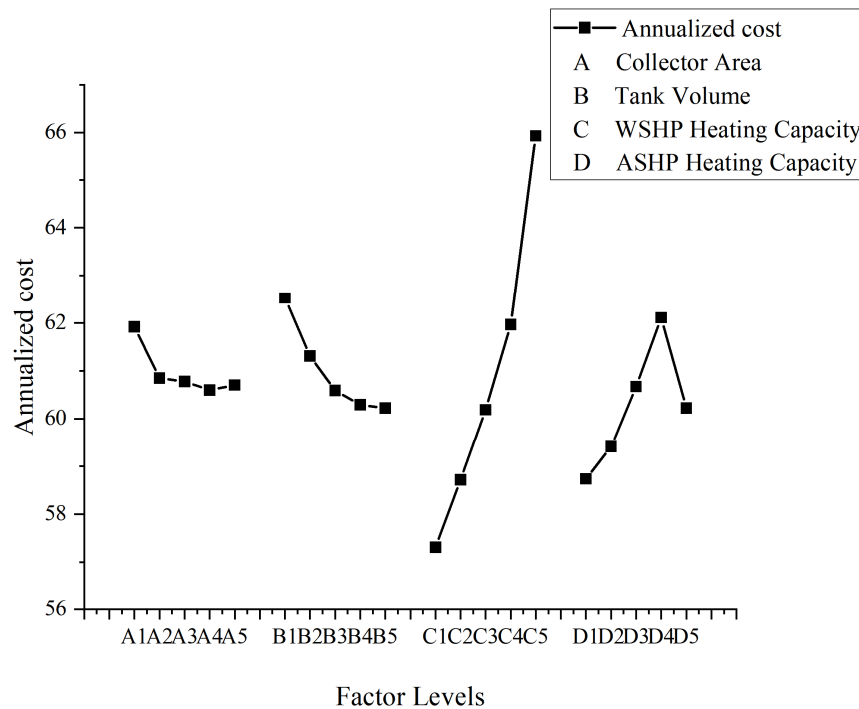


Figure 8. Factor level and annualized costs.

Figure 8 visually demonstrated the influence of factor levels on the annualized cost. Using the annualized cost as the evaluation criterion, lower values indicate better economic performance of the system. Therefore, the optimal parameter combination was selected as the one with the minimum annualized cost among the four factors. According to Figures 3–5, the optimal level combination obtained from this orthogonal experiment was A4B5C1D1, corresponding to a solar collector area of 2571 m², a heat storage tank volume of 170 m³, the WSHP heating capacity of 1839.68 kW, and the ASHP heating capacity of 970.22 kW.

4.3. Collector Tilt Angle Optimization

The radiation intensity received by the solar collector is closely related to the inclination angle of the solar collector, and there is an absolute positive correlation between the radiation intensity and the effective collection amount of the collector. However, as the surface temperature of the collector rises, the heat collection efficiency of the collector will decrease accordingly. In previous designs, the inclination angle of the solar collector is usually set as the local latitude as the optimal angle. But the monthly optimal inclination angle of the solar collector has seasonal variations, changing with the seasons. When changing from summer to winter, the monthly optimal inclination angle will gradually increase, while when changing from winter to summer, it will gradually decrease. Therefore, setting only the local latitude as the optimal inclination angle does not fully conform to the actual situation, and the inclination angle of the solar collector needs to be optimized according to specific circumstances.

In this section, the inclination angle of the solar collector is optimized based on the principle of maximizing the total annual solar radiation. Through the simulation of TRNSYS, the inclination angle of the solar collector was taken as the variable, with a change step of 5°, and the change range from 0° to 90°. The cumulative heat collection amount of the solar energy was simulated and calculated, as shown in Figure 9.

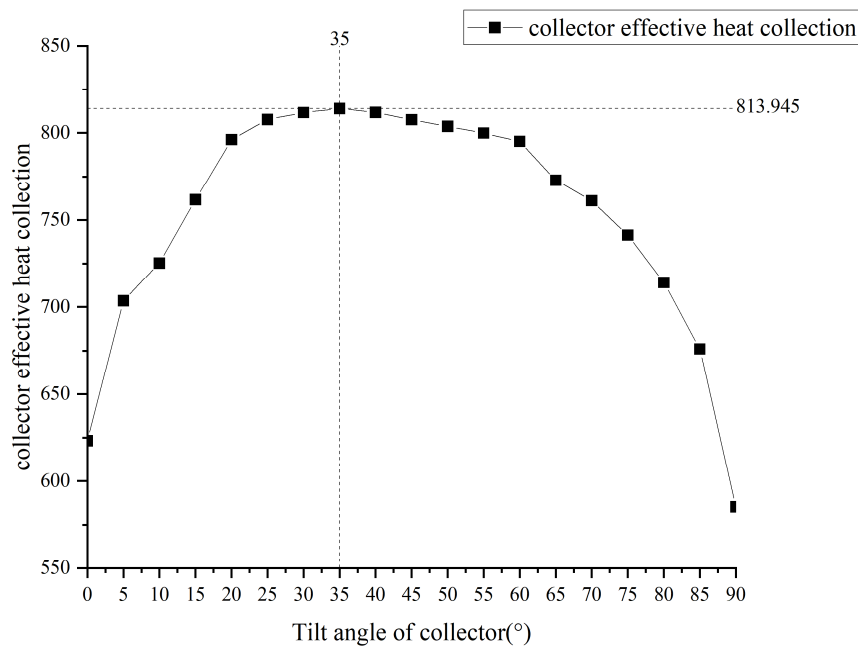


Figure 9. Cumulative heat collection of solar collectors at different inclination angles.

As could be seen from Figure 8, the cumulative heat collection amount of the solar energy first increased and then decreased with the inclination angle of the solar collector, and reached its peak when the inclination angle was 35°.

This study did not consider the potential impact of snow accumulation in the design of the collector tilt angle for Changchun, a severe cold region. The reason is that the system constructed in this study aims to maximize the annual total solar heat gain of the solar collection system to support the hot water demand during the bathhouse's actual operational periods. The months with the heaviest snow accumulation in the system's location are January and February, during which the university is on winter vacation and the bathhouse is closed. Therefore, the final 35° tilt angle selected in this paper, although lower than Changchun's latitude of 45°, was determined without incorporating the impact of snow accumulation into the calculation of the optimal collector tilt angle.

4.4. Experimental Design Optimization

The optimal parameter combination derived from orthogonal experiments—solar collector area of 2571 m², heat storage tank volume of 170 m³, WSHP heating capacity of 1839.68 kW, and ASHP heating capacity of 970.22 kW—was implemented in the TRNSYS simulation model. A comparative analysis of the optimized design parameters versus the initial design parameters (Table 2) is presented in Table 6

Table 6. Comparison of design scheme and optimized parameters.

Parameter	Pre-Optimization Design	Optimized Design	Optimization Percentage
Solar collector area (m ²)	2000	2571	+28.55
Heat storage tank volume (m ³)	120	170	+41.6
Solar heat collection (kWh)	852,655	965,551	+13.22
Annual WSHP heating (kWh)	2,121,913	2,051,753	−3.31
Annual ASHP heating (kWh)	304,939	291,915	−4.3
Annual system energy consumption (kWh)	518,078	461,020	−11.01
WSHP COP	4.459	4.6	+3.16
Total system investment	228.84	202.75	−11.40
Annual operational cost	27.19	26.70	−1.8
Annualized cost	57.28	50.85	−11.23

The optimized design showed a 28.55% increase in solar collector area, a 41.6% increase in tank volume, and a 14.5% reduction in both WSHP and ASHP heating capacities. Simulation results indicated a 3.3% decrease in annual WSHP heating output and a 4.3% decrease in ASHP heating output, while solar heat collection increased by 13.22%. The WSHP COP improved from 4.459 to 4.6. The increase in solar heat collection after optimization reduced the operating frequency of the WSHP and its corresponding energy consumption, leading to an improvement in the system COP.

Economically, the total system investment decreased by 11.40% (from 2,288,400 RMB to 2,027,500 RMB), annual operational costs decreased by 1.8% (from 271,900 RMB to 267,000 RMB), and the annualized cost decreased by 11.23% (from 572,800 RMB to 508,500 RMB). These improvements confirmed the enhanced economic performance of the optimized system.

4.5. Hot Water Supply Reliability Analysis of Optimized System

Annual simulation in TRNSYS using optimized parameters yielded tank outlet and point-of-use temperatures shown in Figures 10 and 11. Figure 10 demonstrated that the post-optimization tank outlet temperature averaged approximately 47 °C. Throughout the annual operation, only isolated instances fell below 43 °C, confirming the optimized system reliably supplied adequate hot water to the bathhouse. Figure 11 revealed greater temperature fluctuations at the point of use during the spring semester, though all values remained above 43 °C. The sole exception occurred at 2420 h (42.89 °C). These results verified that the optimized system met bathhouse hot water requirements.

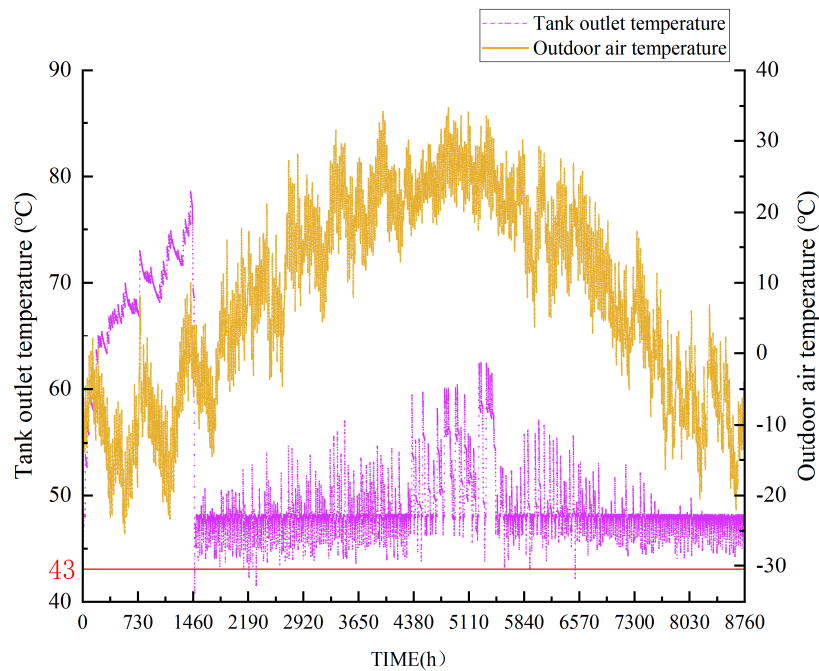


Figure 10. Optimized system: Outlet temperature versus ambient air temperature.

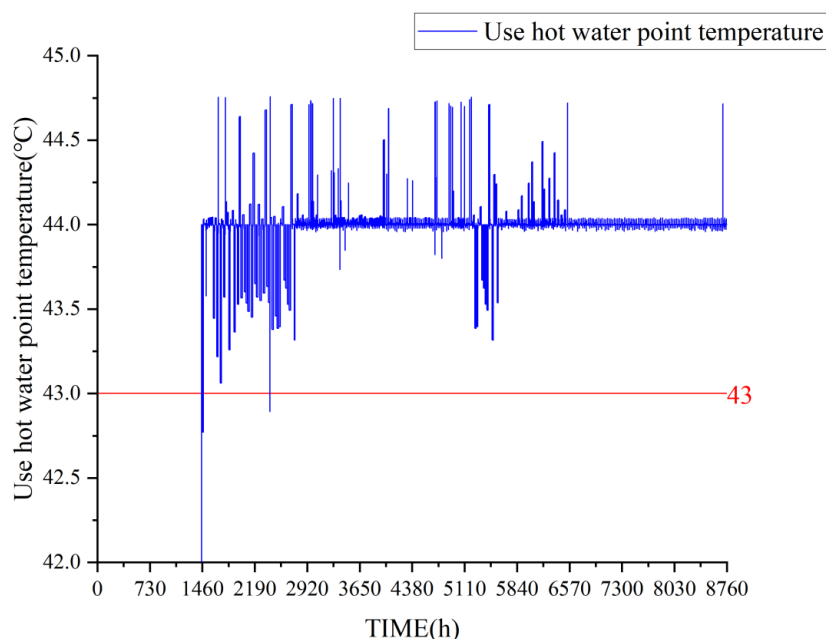


Figure 11. Optimized system: Use hot water point temperature.

5. Comprehensive Benefit Evaluation of the Optimized System

5.1. Energy Consumption Comparison of Hot Water Preparation Schemes

There are many options for preparing hot water to meet bathhouse demand. Conventional energy sources include gas-fired boilers, coal-fired boilers, and diesel boilers; renewable energy options include electric boilers, solar water heating systems, air-source heat pumps (ASHPs), and water-source heat pumps (WSHPs). Through comparative analysis of energy consumption across several hot water preparation systems, Table 7 presented the theoretical calorific values and thermal efficiencies of each system [34–36].

Table 7. Actual Calorific Values and Thermal Efficiencies of Various Hot Water Systems.

Hot Water Systems	Energy Source	Theoretical Calorific Value	Unit	Thermal Efficiency	Actual Calorific Value
Gas-fired Boiler	Natural Gas	8600	Kcal/m ³	70%	6020
Coal-fired Boiler	Coal	7000	Kcal/kg	60%	4200
Diesel Boiler	Diesel	10,200	Kcal/kg	85%	8670
Electric Boiler	Electricity	860	Kcal/(kWh)	98%	842.8
Air-source Heat Pump	Electricity	860	Kcal/(kWh)	250%	2150

Based on the bathhouse's total annual hot water load and actual calorific values (from Table 7), the annual energy consumption of gas-fired boilers, coal-fired boilers, and diesel boilers was determined. Combining this with the energy consumption of the composite system yielded the energy consumption calculation for each scheme, as presented in Table 8.

Table 8. Calculation of Energy Consumption for Each Scheme.

Hot Water System	Energy Source	Annual Energy Consumption (kWh)
Gas-fired Boiler	Natural Gas	453,793.8
Coal-fired Boiler	Coal	650,695.8
Diesel Boiler	Diesel	315,215.9
Electric Boiler	Electricity	3,242,670.3
composite system	Electricity	518,070

Using primary energy conversion coefficients (natural gas: 1.2143, diesel: 1.4571, electricity equivalent: 0.1129), the annual energy consumption of each hot water system was converted into tons of standard coal equivalent (tce) for comparative analysis. The resulting energy consumption comparison was presented in Figure 12.

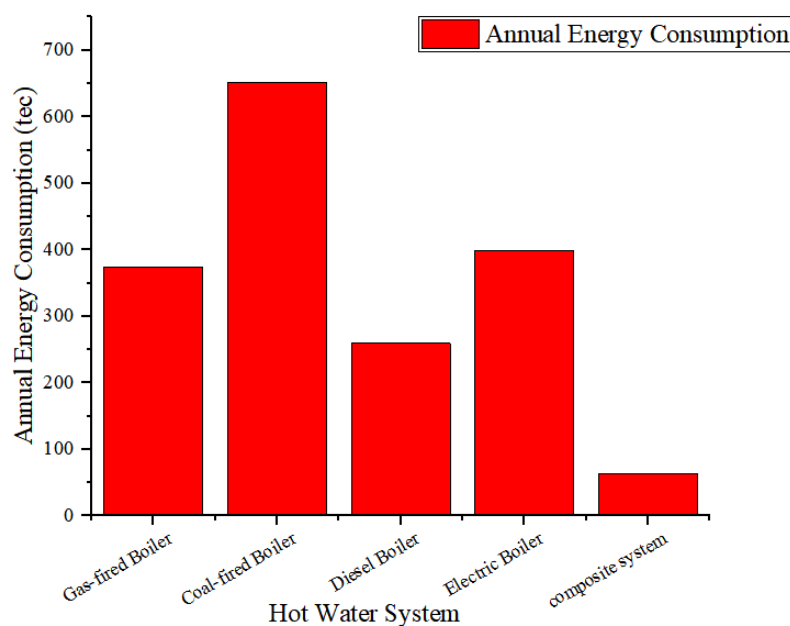


Figure 12. Comparison of annual energy consumption of hot water system.

Figure 12 and Table 8 demonstrated the annual energy consumption comparison across hot water systems. The composite system designed in this study exhibited the lowest annual energy consumption, at only 63.63 tce. The diesel boiler, gas-fired boiler, and electric boiler systems followed, with annual energy consumptions of 259.586 tce, 373.71 tce, and 398.624 tce, respectively. The coal-fired boiler system had the highest annual energy consumption, reaching 650.695 tce. This clearly indicated that the composite system adopted in this study achieved pronounced energy-saving effects.

5.2. Environmental Analysis of the Composite System

5.2.1. Annual Conventional Energy Substitution of the Composite System

The energy savings of the composite system primarily manifest as reductions compared to conventional energy sources. The annual conventional energy substitution (Q_{bm}) is calculated using Equations (14) and (15) [37]:

$$Q_{bm} = Q_{ce} - Q_{sys} \quad (14)$$

$$Q_{ce} = \frac{Q_h}{q_{ce} \times \eta_{ce}} \quad (15)$$

where Q_{bm} is the annual conventional energy substitution (tce); Q_{ce} is the Standard coal consumption for equivalent hot water production using conventional energy (tce); Q_{sys} is the annual primary energy consumption of the composite system (tce); Q_h is the total annual bathhouse hot water load (MJ); q_{ce} is the calorific value of standard coal equivalent (MJ/tce); η_{ce} is the thermal efficiency of standard coal.

Based on questionnaire surveys, the total annual bathhouse hot water load Q_h was calculated as 3,279,507 kWh (equivalent to 11,806,225.2 MJ). Using the standard coal equivalent calorific value q_{ce} of 7000 kcal/kg (equivalent to 29,309 MJ/tce) and a standard coal thermal efficiency η_{ce} of 65%, Q_{ce} was determined to be 619.72 tce. Since the composite system exclusively utilizes renewable energy, Q_{sys} equals the system's annual energy consumption. With the optimized system's annual energy consumption being 461,020 kWh and applying an electricity-to-primary-energy conversion factor of 0.31 kgce/c, Q_{sys} was calculated as 142.916 tce. Substituting Q_{ce} and Q_{sys} into Equation (14) yielded an annual conventional energy substitution Q_{bm} of 476.804 tce for the system.

5.2.2. Pollutant Reduction by the Composite System

The composite system designed in this study utilizes three heat sources—solar energy, water-source heat pump (WSHP), and air-source heat pump (ASHP)—all representing renewable energy technologies. This configuration significantly reduces conventional energy consumption and eliminates harmful gas emissions during operation, thereby substantially mitigating environmental pollution.

For environmental benefit quantification, the system's annual conventional energy substitution (476.804 tce) was applied to Equations (16)–(19) to calculate reductions in CO₂, SO₂, NO_x, and particulate matter [38]:

$$E_{CO_2} = Q_{bm} \cdot F_{CO_2} \quad (16)$$

$$E_{SO_2} = 2\%Q_{bm} \quad (17)$$

$$E_{NO_x} = Q_{bm}(1 - P)0.725\% \quad (18)$$

$$Q_{ycp} = 1\%Q_{bm} \quad (19)$$

where E_{CO_2} is the CO₂ reduction over system lifespan (kg); F_{CO_2} is the CO₂ emission factor; E_{SO_2} is the SO₂ reduction over system lifespan (kg); E_{NO_x} is the NO_x reduction over system lifespan (kg); P is the NO_x desulfurization rate (80%); Q_{ycp} is the particulate matter reduction over system lifespan (kg).

As detailed in Table 9, the composite system implemented in university settings achieves annual emission reductions of 706.15 tons of CO₂, 9.54 tonnes of SO₂, 0.69 tons of NO_x, and 4.77 tons of particulates during hot water production. Compared to coal-fired boilers, this system demonstrates substantial advantages in mitigating environmental pollution.

Table 9. Pollutant Reduction by the Composite System.

	CO ₂ (t)	SO ₂ (t)	NO _x (t)	Particulate Matter (t)
Pollutant Reduction	706.15	9.54	0.69	4.77

6. Conclusions

This study investigates a solar-wastewater source heat pump hybrid system for a university bathhouse in severe cold regions. The system integrates three heat sources: solar thermal collectors, a wastewater source heat pump (WSHP), and an auxiliary air-source heat pump (ASHP). Through questionnaire-based analysis of bathing demand and design specifications, the system configuration and control strategies were established. TRNSYS simulations yielded the following conclusions:

- (1) The annual simulation of the hybrid system demonstrated effective performance: solar heat collection reached 852,655 kWh, WSHP heating output totaled 2,121,913 kWh, and ASHP contributed 304,939 kWh. The WSHP achieved a COP of 4.459. While outlet temperatures remained above 43°C with reliability factors averaging 1.2, indicating full demand fulfillment, these results revealed excessive heat production, highlighting optimization potential.
- (2) Orthogonal experiments with four factors (solar collector area, tank volume, WSHP/ASHP capacities) and five levels identified WSHP capacity as the most influential parameter on annualized costs (priority order: WSHP > ASHP > Tank volume > Collector area).
- (3) The optimized system configuration achieved through orthogonal experiments comprises a solar collector area of 2571 m², a heat storage tank volume of 170 m³, the WSHP heating capacity of 1839.68 kW, and the ASHP heating capacity of 970.22 kW. Post-optimization results demonstrated a 13.2% increase in solar heat collection, a 3.1% improvement in WSHP COP (from 4.459 to 4.6), an 11.4% reduction in initial investment, a 1.8% decrease in annual operational costs, and an 11.2% reduction in annualized costs. Furthermore, compared to coal-fired boilers, this system demonstrated significant advantages in mitigating environmental pollution. During hot water production, the composite system achieved annual emission reductions of 706.15 tonnes of CO₂, 9.54 tonnes of SO₂, 0.69 tonnes of NO_x, and 4.77 tonnes of particulates.

This study examines the applicability of solar-assisted heat pump systems in bathhouse settings. The university bathhouse analyzed is located within the Jingyuetan Tourist Reserve in Northeast China, where environmental regulations mandate clean energy sources. Consequently, an air-source heat pump (ASHP) was directly selected as the auxiliary heat source without comparative analysis of alternatives. This site-specific constraint limits the universal applicability of the system design. Practical implementations should therefore determine auxiliary heat sources according to local conditions. Future research should also address two critical aspects: the impact of sewage impurities on water-source heat pump (WSHP) performance, and validation of the optimized solution's applicability using additional empirical engineering data.

Author Contributions

Y.W.: writing—original draft preparation, conceptualization, methodology; X.Y.: data curation; C.P.A.-Y.: supervision, validation; B.T.C.: visualization, investigation; M.Q.: writing—reviewing and editing, funding acquisition, supervision. All authors have read and agreed to the published version of the manuscript.

Funding

This work was supported by the Talent Introduction Program of Xihua University (Grant No. ZX20250024).

Data Availability Statement

Not applicable.

Conflicts of Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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