International Journal of Heat and Technology
Vol. 43, No. 3, June, 2025, pp. 939-950
Journal homepage: http://iieta.org/journals/ijht

Energy Recycling in Ecotourism Zones Based on Irreversible Thermodynamic Processes

Xiaodong Xu

Department of Economics and Management, Nantong Vocational University, Nantong 226007, China

Corresponding Author Email: ntzdxxd2911010@sina.com

https://doi.org/10.18280/ijht.430312

Received: 2 January 2025
Revised: 10 May 2025
Accepted: 1 June 2025
Available online: 30 June 2025

Keywords:
ecotourism zone, energy recycling, irreversible thermodynamic processes, finite-time thermodynamics, exergy analysis

ABSTRACT

With the rapid expansion of the ecotourism industry, the contradiction between rising energy consumption and ecological preservation in tourism zones has become increasingly pronounced. In conventional energy utilization models, the irreversible nature of thermodynamic processes leads to significant waste of residual heat and exergy, resulting in reduced energy efficiency and posing challenges to the realization of carbon neutrality objectives. Existing practical implementations predominantly rely on empirically integrated technologies and lack systematic analysis of the mechanisms underlying irreversibility. Major limitations persist in current research: the construction of most models under reversible assumptions disregards irreversibility factors inherent to real-world conditions, which leads to considerable discrepancies between theoretical predictions and actual performance; energy efficiency evaluations are largely centered on the conservation of energy quantity, without incorporating exergy-based analyses to account for the degradation in energy quality; furthermore, existing studies primarily target industrial or urban energy systems, offering limited applicability to the specific characteristics of ecotourism zones. To address these gaps, a closed-loop energy recycling system model for ecotourism zones driven by residual heat and energy was constructed based on the principles of finite-time thermodynamics. Analytical expressions for exergy output rate and exergy efficiency were derived. Through a comparative analysis of these two performance metrics, their similarities and differences were elucidated, and feasible system design parameter ranges were identified. Based on exergy performance evaluation, targeted strategies for energy recycling in ecotourism zones were proposed. The novelty of this study lies in the theoretical integration of irreversible process modeling from finite-time thermodynamics into the domain of ecotourism energy systems for the first time—addressing a gap in the analysis of regional energy systems under non-equilibrium conditions. Furthermore, a quantitative design basis for system parameters has been established, facilitating the transition from empirical implementation to scientifically optimized energy recycling solutions.

1. INTRODUCTION

With the rapid expansion of the ecotourism industry [1-4], the contradiction between escalating energy consumption and ecological conservation in ecotourism zones has become increasingly pronounced. On one hand, the surge in visitor numbers [5, 6] has led to a continuous increase in energy demand for dining, accommodation, and recreational facilities. On the other hand, the strict environmental disturbance limits imposed in ecotourism zones necessitate the development of energy systems characterized by high recycling efficiency and minimal emissions [7-9]. In conventional energy utilization models [10, 11], waste of residual heat and exergy caused by irreversible thermodynamic processes reduces the overall efficiency of energy use and significantly conflicts with carbon neutrality objectives. In actual practice, integrated ecotourism complexes—such as Xiangjiang Happy City—have attempted to implement energy recycling through techniques such as condensate heat recovery and passive ventilation design. However, these implementations are predominantly based on empirically integrated technologies, lacking systematic analysis of the mechanisms underlying irreversible processes. As a result, universally applicable optimization schemes have yet to be developed.

A detailed investigation into the irreversible thermodynamic characteristics of energy recycling systems in ecotourism zones holds substantial theoretical and practical significance. Theoretically, such an investigation transcends the idealized assumptions of classical thermodynamics [12, 13] by incorporating the principles of finite-time thermodynamics and aligning them with the energy flow dynamics of ecological systems. This enables the refinement of analytical frameworks for regional energy systems under non-equilibrium conditions. From an application standpoint, accurately identifying the mechanisms of exergy loss caused by residual heat and energy provides a scientific foundation for designing energy recycling systems that are both efficient and low in energy consumption. Such systems not only ensure a stable energy supply to meet visitor demand but also minimize interference with the ecological base, thereby facilitating the transition of “green tourism” [14, 15] from a conceptual ideal to a technically actionable pathway.

Significant limitations persist in current analyses of energy recycling systems in three key aspects. First, the development of most existing models under the assumption of reversible processes overlooks inherently irreversible factors such as temperature differentials during heat transfer and frictional dissipation, resulting in a marked deviation between theoretically optimized results and the complex operating conditions of ecotourism zones [16, 17]. Second, evaluations of energy efficiency typically emphasize the conservation of energy quantity, failing to utilize exergy analysis to uncover the mechanisms behind the degradation of energy quality. As a result, critical loss points within the recycling system remain unidentified [18, 19]. Third, most existing studies are focused on industrial or urban energy systems and lack targeted designs for the unique operational characteristics of ecotourism zones, which are marked by distributed energy consumption, intermittent loads, and stringent environmental constraints. Consequently, the applicability of current models is limited [20].

To address these challenges, a more realistic analytical system for energy recycling in ecotourism zones was proposed in this study. First, based on finite-time thermodynamics, an irreversible closed-loop energy recycling system model driven by residual heat and energy was established, and analytical expressions for exergy output rate and exergy efficiency were derived. Then, through comparative analysis of the two performance metrics, the synergistic and conflicting interactions between energy quality and quantity under varying operational parameters were identified, and optimal design intervals for system parameters were determined. Finally, taking into account the specific conditions of ecotourism zones, energy recycling strategies were proposed that balance exergy performance enhancement with ecological protection. The primary innovation of this study lies in the theoretical introduction of irreversible thermodynamic modeling into the context of ecotourism zones for the first time, thereby addressing a gap in the analysis of regional energy systems under non-equilibrium conditions. In practical terms, this work provides quantitative guidance for the parameter design of similar systems, promoting the transition of energy recycling technologies from passive empirical application to proactive scientific optimization.

2. CLOSED-LOOP ENERGY RECYCLING MODEL FOR ECOTOURISM ZONES BASED ON IRREVERSIBLE THERMODYNAMIC PROCESSES

In the proposed energy recycling system for ecotourism zones, residual heat from blast furnace gas was utilized as the primary driving source to establish a closed-loop network that integrates heating, cooling, and residual heat recovery. During system operation, a high-temperature heat source (denoted as S_G) receives residual heat from blast furnace gas and transfers energy to the working fluid via a high-temperature-side heat exchanger with thermal conductance H_G. Following this, the working fluid undergoes irreversible adiabatic compression (Process 1-2), after which it enters the high-temperature heat absorption stage (Process 2-3). The absorbed heat is partially used to drive a turbine through irreversible adiabatic expansion (Process 3-4), while the remaining portion is supplied at constant pressure (Process 4-5) to an absorption refrigeration generator. The working fluid temperature in this stage is maintained at S'_h to meet cooling demands in the ecotourism zone, such as food refrigeration and indoor air conditioning. The residual energy continues through a constant-pressure heat release process (Process 5-6), during which it is transferred to the thermal demand side via a heat exchanger with conductance Gh, supplying heat at temperature S_g for scenarios such as guestroom heating and hot spring temperature maintenance. Following heat delivery, the working fluid releases residual heat via a low-temperature-side heat exchanger with conductance H_M (Process 6-1), closing the thermodynamic cycle. The low-temperature heat sink (denoted as S_M) typically consists of natural water bodies or ambient air. Dynamic adaptation to the distributed and intermittent energy loads of ecotourism zones is achieved by regulating the specific heat capacity of the working fluid (Z_{qd}) and optimizing the thermal conductance matching of each heat exchanger. This design not only reduces energy waste caused by the direct discharge of blast furnace residual heat but also minimizes ecological disruption that might arise from traditional energy supply systems. Figure 1 illustrates the schematic flow diagram of the energy recycling system tailored for ecotourism zones.

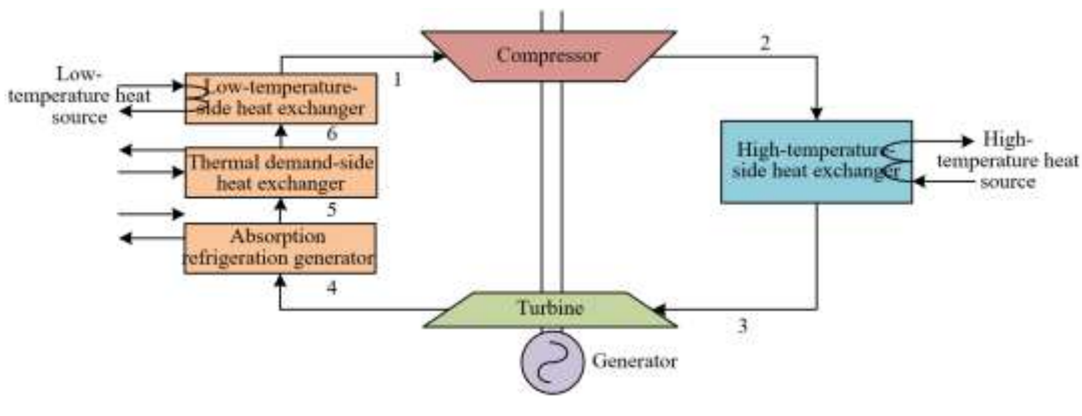


Figure 1. Schematic of the energy recycling system in ecotourism zones

The core characteristics of the thermoelectric component in the closed-loop energy recycling system based on irreversible thermodynamic processes for ecotourism zones are illustrated in the temperature-entropy diagram shown in Figure 2, which highlights the non-ideal processes and the associated degradation in energy quality.

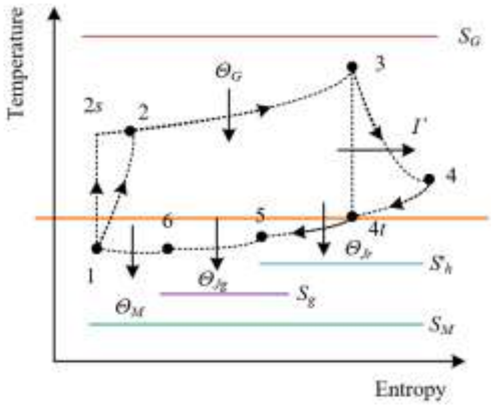


Figure 2. Temperature-entropy diagram of the thermoelectric component in the ecotourism zone energy recycling system

Due to irreversible factors such as mechanical friction and turbulent flow of the working fluid, Processes 1-2 and 3-4 deviate from ideal adiabatic behavior. This deviation manifests as greater entropy increases compared to reversible processes, resulting in a reduction in effective work output. In the high-temperature heat absorption process (Process 2-3) and the low-temperature heat release process (Process 6-1), irreversible heat transfer losses are induced by finite temperature differences—specifically, between S_G and the working fluid, and between the working fluid and S_M. These losses are reflected in the temperature-entropy diagram by the separation between the heat absorption/release curves and the respective isothermal lines of the heat sources. Smaller thermal conductances H_G and H_M exacerbate the temperature gradients, leading to more pronounced exergy losses. During the constant-pressure heating stages (Processes 4-5 and 5-6), the working fluid's temperature gradually decreases from the turbine outlet temperature to the thermal demand-side temperature S_g. If a mismatch exists between the working fluid's specific heat capacity Z_{qd} and the thermal conductance H_h of the heat exchanger on the demand side, non-isothermal heat transfer losses may arise. These deviations appear in the temperature-entropy diagram as entropy changes that diverge from the ideal isobaric process path. By quantifying the entropy generation of each irreversible process, a thermodynamic foundation is established for the subsequent analysis of exergy output rate and exergy efficiency. This facilitates dual-objective optimization of both energy quantity and quality within the energy recycling system designed for ecotourism zones.

A key subsystem bridging residual heat utilization and cooling demand in the energy recycling system of ecotourism zones is the internally reversible absorption refrigeration cycle model, in which only the heat transfer irreversibility between the working fluid and heat sources is considered. The detailed configuration is shown in Figure 3. This model is driven by the heat released during Process 4-5 of the irreversible closed-loop cycle, where the working fluid at temperature S'_h transfers energy via a generator with thermal conductance H_g. The generator's high-temperature heat source varies dynamically with fluctuations in the residual heat supplied by the main cycle of the ecotourism zone. Within the cycle, the evaporator, condenser, and absorber exchange heat with constant-temperature heat sources S_r, S_z, and S_x, respectively. Irreversible heat losses arise solely due to finite temperature differences between the working fluid and the heat sources, specifically $S_r>S'_r$, $S_z<S'_z$, and $S_x<S'_x$. Mechanical dissipation and throttling losses of the working fluid are assumed to be negligible.

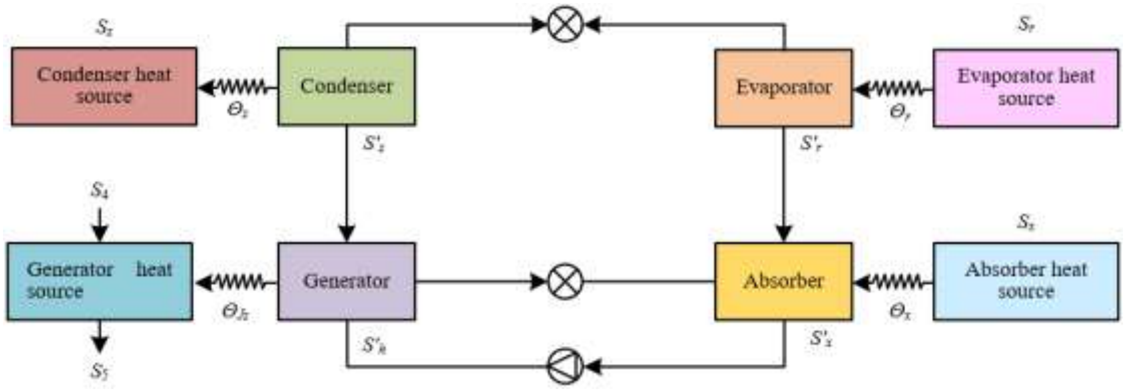


Figure 3. Configuration of the internally reversible absorption refrigeration cycle model

Specifically, after absorbing residual heat from the main cycle, the working fluid undergoes desorption in the generator. The resulting refrigerant vapor flows into the condenser, releasing heat to the low-temperature heat source S_z and condensing at temperature S'_z. The liquid refrigerant then passes through a throttle and enters the evaporator, where it evaporates at temperature S'_r by absorbing heat from a high-temperature heat source S_r, thereby meeting cooling demands such as food preservation and indoor air conditioning in the ecotourism zone. The refrigerant vapor exiting the evaporator flows into the absorber, where it is absorbed by an absorbent and releases heat at temperature S'_x to the heat source S_x, completing the cycle. This model enables quantification of the influence of the generator's thermal conductance H_h and the individual heat transfer temperature differences (S_r-S'_r, S'_z-S_z, and S'_x-S_x) on cooling capacity and exergy efficiency. It effectively captures the degradation in refrigeration performance under dynamically varying residual heat inputs, accommodating the intermittent nature of residual heat supply in ecotourism zones. Furthermore, it provides a theoretical basis for analyzing the exergy matching between the refrigeration subsystem and the main energy cycle, ultimately supporting parameter optimization and performance enhancement of the overall energy recycling system.

In the irreversible closed-loop energy recycling system for ecotourism zones, driven by residual heat and energy, the working fluid was modeled as an ideal gas with constant specific heat to simplify energy transfer calculations. Heat exchange between the working fluid and various heat sources is assumed to follow Newton's law of cooling. Based on the thermophysical properties of the working fluid and standard heat exchanger theory, the heat absorption flow rate from the high-temperature heat source, denoted as Θ_G, is determined by the thermal conductance of the high-temperature-side heat exchanger (H_G) and the temperature difference between the average working fluid temperature during the heat absorption process and S_G. This flow rate directly governs the energy input to the cycle. The heat release flow rate to the low-temperature heat source, Θ_M, is calculated using the thermal conductance H_M of the low-temperature-side heat exchanger and the corresponding temperature difference between the working fluid during the heat release process and S_M, reflecting the level of residual heat discharge from the system.

The heat supply flow rate to the absorption refrigeration generator, Θ_{Jz}, is driven by the working fluid temperature at Process 4-5 (S'_h) and the thermal conductance H_h of the generator, providing the energy required for the refrigeration subsystem. Meanwhile, the heat delivery flow rate to the thermal demand side, Θ_{Jg}, is determined by the thermal conductance H_g of the demand-side heat exchanger and the temperature difference between the working fluid in Process 5-6 and the thermal demand temperature S_g. This directly supports heating needs in the ecotourism zone, such as space heating and hot water supply. Accurate computation of these thermal flow rates serves as the foundation for evaluating the overall energy recycling efficiency of the system and enables dynamic matching with the variable energy demands induced by fluctuating tourist activity. Specifically, assuming the effectiveness values of the heat exchangers associated with the high-temperature side, low-temperature side, absorption refrigeration generator, and thermal demand side are represented as R_G, R_M, R_h, and R_g, respectively, the expressions for Θ_G, Θ_M, Θ_{Jz}, and Θ_{Jg} are formulated as follows:

$$\Theta_G = Z_{qd}\left(S_3 - S_2\right) = Z_{qd} R_G \left(S_G - S_2\right) \tag{1}$$

$$\Theta_M = Z_{qd}\left(S_6 - S_1\right) = Z_{qd} R_M \left(S_6 - S_M\right) \tag{2}$$

$$\Theta_{Jz} = Z_{qd}\left(S_4 - S_5\right) = Z_{qd} R_h \left(S_4 - S_h^{'}\right) \tag{3}$$

$$\Theta_{Jg} = Z_{qd}\left(S_5 - S_6\right) = Z_{qd} R_g \left(S_5 - S_g\right) \tag{4}$$

The effectiveness coefficients R_G, R_M, R_h, and R_g represent critical indicators of performance across the various heat exchange interfaces in the ecotourism energy recycling system. The coefficient R_G is defined as the ratio between the actual heat transferred through the high-temperature heat exchanger and the theoretical maximum heat transfer, indicating the recovery efficiency of blast furnace residual heat. A higher R_G value corresponds to more complete residual heat utilization, aligning with the energy conservation goals of ecotourism zones. The coefficient R_M characterizes the heat rejection efficiency of the low-temperature-side exchanger and must be maintained within an optimal range to avoid excessive energy dissipation. The coefficient R_h reflects the generator's capacity to absorb residual heat from the main cycle, directly influencing the driving potential of the refrigeration subsystem. Finally, R_g evaluates the effectiveness of heat transfer from the demand-side heat exchanger to end-use devices within the ecotourism zone and is closely linked to tourist thermal comfort. By optimizing the effectiveness of each heat exchanger, energy allocation within the ecotourism zone can be adaptively distributed according to real-time demand, minimizing energy waste while maintaining service quality. The mathematical formulations of R_G, R_M, R_h, and R_g are expressed as:

$$\begin{aligned} R_G &= 1 - r^{-V_G}, R_M = 1 - r^{-V_M} \\ R_h &= 1 - r^{-V_h}, R_g = 1 - r^{-V_g} \end{aligned} \tag{5}$$

The number of heat transfer units—denoted as V_G for the high-temperature-side heat exchanger, V_M for the low-temperature-side exchanger, V_h for the generator, and V_g for the thermal demand-side exchanger—serves as a core parameter for quantifying the match between exchanger size and heat transfer capacity in the energy recycling system for ecotourism zones. V_G is defined by the ratio between the high-temperature-side thermal conductance H_G and the specific heat capacity Z_{qd} of the working fluid. A higher value of V_G indicates a larger heat transfer area or higher heat transfer coefficient, making the exchanger well-suited for handling high-grade energy such as blast furnace residual heat. V_M reflects the scale of the heat exchanger and must be designed to match the ambient heat dissipation conditions in the surrounding natural environment of the ecotourism zone. V_h depends on the generator's thermal conductance H_h and the specific heat capacity of the refrigerant working fluid, which affects the responsiveness of the refrigeration subsystem to the main cycle's residual heat. V_g must be determined based on the spatial distribution and maximum heat load of users in the ecotourism zone to ensure stable heat supply under varying demand conditions. Appropriate design of these heat transfer units allows the system to operate efficiently even during peak load periods in high tourism seasons, while avoiding equipment underutilization or overloading. The mathematical expressions for these parameters are defined as:

$$\begin{aligned} V_G &= H_G / Z_{qd}, V_M = H_M / Z_{qd} \\ V_h &= H_h / Z_{qd}, V_g = H_g / Z_{qd} \end{aligned} \tag{6}$$

For the reversible cycle processes represented by 1-2t-3-4t-m, the second law of thermodynamics requires that the net entropy change of the working fluid over the entire cycle be zero, indicating that the sum of entropy changes across all processes equals zero. In this scenario, Process 1-2t is an isentropic (no entropy generation) adiabatic compression, Process 2s-3 is an isothermal heat absorption, Process 3-4s is an isentropic adiabatic expansion, and Process 4s-1 is an isothermal heat rejection. This reversible cycle serves as the theoretical benchmark for evaluating the energy recycling efficiency of the ecotourism zone system. By comparing the actual irreversible cycle against this benchmark, exergy losses caused by factors such as finite temperature differences in heat exchange and mechanical dissipation can be quantitatively assessed. This facilitates the identification of "avoidable losses" within the system, thereby revealing potential avenues for performance optimization. For instance, reducing the temperature gradient across heat exchangers through improved design can bring the actual cycle closer to reversible conditions. Assuming the isentropic temperature ratio is defined by $a = \tau^{(j-1)/j}$, where τ represents the compression ratio of the cycle and j denotes the adiabatic index of the working fluid, the corresponding thermodynamic relationship is given as:

$$S_{2t} / S_1 = S_3 / S_{4t} = a \tag{7}$$

The internal irreversibility of the energy recycling system in ecotourism zones is primarily characterized by the compressor efficiency λ_z and the turbine efficiency λ_s, which directly reflect energy losses in mechanical components. The compressor efficiency λ_z is defined as the ratio of ideal adiabatic compression work to actual compression work. In ecotourism systems, compressors are frequently employed to raise the pressure of the working fluid in order to satisfy long-distance heat supply requirements. A lower λ_z indicates increased electrical consumption, which is inconsistent with the objectives of low-carbon tourism. The turbine efficiency λ_s is defined as the ratio of actual expansion work to ideal adiabatic expansion work. The mechanical work recovered by the turbine may be utilized to drive the compressor or to supplement the ecotourism zone's electrical grid. A higher λ_s indicates more effective energy recovery. By selecting high-efficiency compressors and turbines and dynamically adjusting operating parameters according to energy demand time windows in the ecotourism zone, energy losses due to internal irreversibility can be minimized, thereby improving the system's energy self-sufficiency ratio. The mathematical expressions are defined as:

$$\begin{aligned} \lambda_z &= \left(S_{2t} - S_1\right) / \left(S_2 - S_1\right) \\ \lambda_s &= \left(S_3 - T_4\right) / \left(S_3 - S_{4t}\right) \end{aligned} \tag{8}$$

In the absorption refrigeration cycle, it is assumed that the refrigerant working fluid undergoes steady flow within the ecotourism zone energy system. The heat flow rates through the evaporator Θ_r, condenser Θ_z, and absorber Θ_x are considered constant. The value of Θ_r represents the heat absorbed by the working fluid from the constant-temperature heat source S_r in the evaporator, which directly determines the cooling capacity. This must be aligned with the cooling loads of guest rooms and dining facilities during the summer season. Θ_z, representing the heat released by the working fluid to the low-temperature heat source S_z, is dependent on both Θ_r and the input heat Θ_{Jz} from the generator. Θ_x denotes the heat rejected to the heat source S_x, indicating the level of residual heat in the refrigeration cycle. The balance among these heat flow rates forms the foundation for the stable operation of the refrigeration system and can be adaptively managed through intelligent control strategies to accommodate fluctuations in cooling demand caused by diurnal temperature variations within the ecotourism zone. Assuming the thermal conductance of the heat exchangers in the evaporator, condenser, and absorber are denoted by H_r, H_z, and H_x, respectively, the corresponding expressions are:

$$\Theta_r = H_r\left(S_r - S_r^{'}\right) \tag{9}$$

$$\Theta_z = H_z\left(S_z{'} - S_z\right) \tag{10}$$

$$\Theta_x = H_x\left(S_x{'} - S_x\right) \tag{11}$$

In the absorption refrigeration cycle, the distribution ratio of total heat rejection between the absorber and the condenser, denoted by $v = \Theta_x/\Theta_z$, directly affects the efficiency of energy allocation within the cycle. Given that the input power of the solution pump and thermal losses due to fluid flow are relatively small in ecotourism zone applications, the cooling capacity E and the coefficient of performance (COP), denoted by γ, can be derived based on the principle of energy conservation. The optimization of v must be aligned with the characteristics of cooling demand in ecotourism zones. An increase in v corresponds to a greater proportion of heat being released by the absorber and a reduced heat load on the condenser, potentially lowering the energy consumption of the cooling water system. Meanwhile, a higher γ indicates that more cooling capacity is generated per unit of residual heat input, which aligns with the "waste-to-energy" strategy commonly promoted in sustainable ecotourism. By adjusting the parameter v, the refrigeration system can maintain a high γ across varying operating conditions, balancing cooling performance with energy efficiency. The mathematical expressions for E and γ are defined as follows:

$$E = \Theta_r = H_r\left(S_r - S_r^{'}\right) \tag{12}$$

$$\gamma = \Theta_r / \Theta_{Jz} \tag{13}$$

According to the first and second laws of thermodynamics, the following general thermodynamic relations also apply:

$$\Theta_{Jz} + \Theta_r - \Theta_x - \Theta_z = 0 \tag{14}$$

$$\Theta_{Jz} / S_h^{'} + \Theta_r / S_r^{'} - \Theta_x / S_x^{'} - \Theta_z / S_z^{'} = 0 \tag{15}$$

Given known thermal conductance values for H_h, H_r, H_z, and H_x, a clear functional relationship can be established between the cooling capacity E and the heat input Θ_{Jz} from the main cycle to the generator. This relationship is governed jointly by the heat transfer characteristics of the exchangers and the temperature differences between the working fluid and the respective heat sources. In the context of ecotourism zones, when Θ_{Jz} fluctuates in response to residual heat variations from blast furnace gas, E exhibits nonlinear behavior constrained by thermal conductance. For example, during nighttime periods when residual heat supply is reduced, and parameters such as H_r and H_z are held constant, the rate of decline of R is determined by the degree of thermal conductance matching. An explicit understanding of this relationship provides a foundation for system design. By optimizing heat exchanger conductance parameters, the refrigeration subsystem can be made resilient to fluctuations in residual heat supply, ensuring that essential cooling demands within the ecotourism zone are continuously met. The generalized relationship between E and Θ_{Jz} is expressed as:

$$\begin{aligned} &H_z\left(\Theta_{Jz} + E\right) / \left[\Theta_{Jz} + E + H_z S_z\left(v+1\right)\right] + \\ &vH_x\left(\Theta_{Jz} + E\right) / \left[v\left(\Theta_{Jz} + E\right) + H_x S_x\left(v+1\right)\right] = \\ &H_r E / \left(H_r S_r - E\right) + Z_{qd}\Theta_{Jz} R_h / \left(Z_{qd} R_h S_4 - \Theta_{Jz}\right) \end{aligned} \tag{16}$$

3. EXERGY PERFORMANCE ANALYSIS OF THE IRREVERSIBLE CLOSED-LOOP ENERGY RECYCLING SYSTEM IN ECOTOURISM ZONES

The calculation of the output power for the energy recycling system in ecotourism zones must comprehensively account for the system's multi-energy output characteristics, which include the mechanical power generated by the turbine and the useful energy delivered in terminal applications. The mechanical work produced by the turbine during the irreversible adiabatic expansion in Process 3-4 constitutes the core power output and may be directly used to drive internal devices or fed into a microgrid. Additionally, the heat supply rate Θ_{Jg} delivered to thermal users, along with the cooling capacity E provided by the absorption refrigeration cycle, are also considered effective components of the total output power. By aggregating the mechanical, thermal, and cooling outputs, the total system output power is obtained. This total must be capable of dynamically tracking the diurnal load fluctuations within the ecotourism zone and serve as the quantitative energy basis for subsequent exergy performance evaluation. The expression for total output power is given by:

$$O = \Theta_G - \Theta_M - \Theta_{Jg} - \Theta_{Jz} \tag{17}$$

The thermoelectric ratio of the energy recycling system is defined as the ratio of thermal power delivered to users to the electrical power output of the turbine. This parameter serves as a critical indicator of energy allocation characteristics within the system. In ecotourism scenarios, the thermoelectric ratio must be flexibly adjusted based on seasonal and temporal variations. During winter, when heating demand is elevated, the ratio should be increased to ensure sufficient heat supply for guest accommodations and hot spring facilities. In contrast, during summer months dominated by cooling demand, part of the thermal energy may be converted into cooling capacity via the absorption refrigeration cycle, thereby indirectly reducing the thermoelectric ratio and improving energy-use adaptability across seasonal scenarios. The acceptable range of this ratio must be established based on annual energy use data from the ecotourism zone to avoid imbalance conditions such as thermal energy oversupply or electricity shortfalls. As such, it also provides a direction for optimizing exergy efficiency. The thermoelectric ratio is defined as:

$$\mu_g = \Theta_{Jg} / O \tag{18}$$

Let $y_1=(a+\lambda_s-\lambda_s a)$, $y_2=a+\lambda_z-1$, $y_3=\lambda_s(1+\mu_g)(a-1)-a$, $y_4=y_2\lambda_s(1-R_G)-a$, and $y_5=aR_g\lambda_z-y_4\mu_g(a-1)(1-R_g)(1-R_M)$, the following can be derived:

$$\begin{aligned}\Theta_G = y_5^{-1} Z_{qd} R_G \Big[& a\lambda_z R_g S_G + y_2 a R_g \big(R_M S_g - R_M S_M \\ & - S_g \big) - \mu_g S_G (a-1)(1-R_g)\times(1-R_M)(y_2\lambda_s - a) \Big]\end{aligned} \tag{19}$$

$$\begin{aligned}\Theta_M = y_5^{-1} Z_{qd} R_G \Big[& (a\text{-}1)(1\text{-}R_g)\big(\mu_g\lambda_z\lambda_s R_g S_G + y_4\mu_g S_M \\ & + \lambda_z S_M - \lambda_z S_g + \lambda_z (S_g - S_M)\times(a + R_g - 1)\big)\Big]\end{aligned} \tag{20}$$

$$\begin{aligned}\Theta_{Jz} = y_5^{-1} Z_{qd} \Big\{ & \mu_g (a-1)\big[R_M S_M (y_1 - \lambda_z\lambda_s) + \\ & y_2 R_G R_M S_M \lambda_s + y_1 R_G S_G (1-R_M) - \lambda_z\lambda_s R_G S_G\big] \\ & + y R_G R_g S_G (\lambda_z + \mu_g - \mu_g a) + y_1 y_2 R_g R_M S_M (1-R_G) \\ & + R_g S_g (1+\mu_g)(a-1)(a - y_2\lambda_s) + y_1 R_g R_M \\ & (a-1)(R_G S_G \mu_g - S_g - S_g \mu_g) + y_2 y_3 S_g R_G R_g \\ & (1-R_M) + y_3 \lambda_z R_g R_M S_g \Big\}\end{aligned} \tag{21}$$

$$\begin{aligned}\Theta_{Jg} = {} & y_5^{-1} Z_{qd} R_g \mu_g (a-1) \\ & \big[y_4 (S_g + R_M S_M - R_M S_g) + \lambda_z\lambda_s R_G S_G \big]\end{aligned} \tag{22}$$

$$\begin{aligned}O = {} & y_5^{-1} Z_{qd} R_g (a-1) \\ & \big[y_4 (S_g + R_M S_M - R_M Sg) + \lambda_z\lambda_s R_G S_G \big]\end{aligned} \tag{23}$$

The output power of the energy recycling system in ecotourism zones is defined as the exergy output, which emphasizes the quality of energy rather than its quantity. Unlike conventional energy output, exergy output reflects the availability of different energy forms: the mechanical work delivered by the turbine is considered high-grade exergy; the exergy of heat supply is reduced by the portion rendered unavailable at ambient temperature; and the cooling exergy captures the utility of thermal energy under sub-ambient conditions. By converting the output power into the exergy output, the actual utilization value of each energy form in specific application scenarios within ecotourism zones can be accurately assessed. This approach helps to avoid overestimations caused by pure energy quantity metrics that neglect degradation in energy quality. The output power of the ecotourism zone's energy recycling system is thus expressed in terms of exergy output:

$$r_q = O \tag{24}$$

The thermal exergy output r_{Jg} is calculated using the general heat exergy formulation below, where S_0 is the ambient temperature and S_g is the heat supply temperature at the thermal demand side. In ecotourism applications, S_g is determined according to the specific use scenario, which is about 50-60°C, 40-45°C, and 60-70°C for guestroom heating, hot spring pool heating, and catering hot water, respectively. This equation shows that, under a fixed heat flow rate, a higher S_g results in greater thermal exergy output. Therefore, high-grade thermal energy should be preferentially allocated to high-temperature demand scenarios to improve the effective utilization of thermal exergy within the ecotourism zone.

$$r_{Jg} = \Theta_{Jg}\left(1 - S_0 / S_g\right) \tag{25}$$

The cooling exergy output from the absorption refrigeration cycle is computed using the following equation, where E is the cooling capacity, S_r is the evaporator refrigerant temperature, and S_0 is the ambient temperature. The temperature S_r varies considerably across different cooling applications in ecotourism settings, which is about 2-8°C and 10-16°C for food refrigeration and air conditioning, respectively. The calculation reveals that a lower S_r results in greater exergy output per unit of cooling energy. Consequently, cooling exergy should be prioritized for lower-temperature cooling demands to prevent the inefficiencies associated with overqualified energy usage, often referred to as the "using a sledgehammer to crack a nut" effect. This calculation provides theoretical guidance for aligning refrigeration system output with zone-specific cooling loads.

$$r_{Jz} = E\left(S_0 / S_r - 1\right) \tag{26}$$

The exergy input r_U of the ecotourism energy recycling system is defined as the exergy value provided by the high-temperature heat source. It is calculated using the equation below, where Θ_G represents the heat absorption rate and S_G denotes the temperature of the high-temperature heat source. The exergy output r corresponds to the sum of all useful exergy delivered by the system, encompassing mechanical exergy from the turbine, thermal exergy, and cold exergy. During actual operation, r_U must be dynamically assessed based on the stability of residual heat from blast furnace gas, whereas e must dynamically respond to changes in tourist load. The difference between them reflects the system's exergy loss, reflecting the total impact of irreversible processes. The exergy input r_U and exergy output r are defined as follows:

$$r_U = \Theta_G\left(1 - S_0 / S_G\right) - \Theta_M\left(1 - S_0 / S_M\right) \tag{27}$$

$$r = r_q + r_{Jg} + r_{Jz} \tag{28}$$

The exergy efficiency of the energy recycling system is therefore defined as:

$$\lambda = r / r_U \tag{29}$$

A dimensionless treatment was applied, obtaining the dimensionless output $\bar{r}=r/(Z_{qd}\ S_0)$ and dimensionless exergy efficiency λ. In the equations below, Z_{qd} represents the specific heat capacity of the working fluid and S_0 is the ambient temperature. This non-dimensionalization eliminates the influence of system size on the evaluation results, enabling the proposed model to be applied universally across ecotourism zones of varying scales. By comparing $\bar{r}$ and $\bar{\lambda}$ across different sites, general principles for exergy optimization in ecotourism energy systems can be extracted. For instance, it has been found that "when Z_{qd} is well-matched with heat transfer conductance, $\bar{r}$ increases with the diversity of energy demands in the ecotourism zone." Such insights provide theoretical support for the standardized design of energy systems in ecotourism applications. Let $\pi_G=S_G/S_0$, $\pi_g=S_g/S_0$, $\pi_z=S_r/S_0$, and $\pi_M=S_M/S_0$, the specific expressions are as follows:

$$\bar{r} = Z_{qd}^{-1}S_0^{-1}\left(\pi_z^{-1}-1\right)E + y_5^{-1}R_g\left(1+\mu_g-\pi_g^{-1}\mu_g\right)\times(a-1)\left[\lambda_z\lambda_s R_G\pi_G + y_4\left(\pi_g + R_M\pi_M - R_M\pi_g\right)\right] \tag{30}$$

$$\begin{aligned} \lambda = &\left\{y_5 Z_{qd}^{-1}S_0^{-1}\left(\pi_z^{-1}-1\right)E + R_g\left(1+\mu_g-\pi_g^{-1}\mu_g\right)\times(s-1)\left[\lambda_z\lambda_s R_G\pi_G + y_4\left(\pi_g + R_M\pi_M - R_M\pi_g\right)\right]\right\} \\ &/\left\{R_M\left(\pi_M^{-1}-1\right)\left[aR_g\lambda_z\left(\pi_g-\pi_M\right)+\mu_g(a-1)\left(a-R_g\right)\left(\lambda_z\lambda_s R_G\pi_G + y_4\pi_g\right)\right]R_G\left(1-\pi_G^{-1}\right)\right. \\ &\left.\times\left[a\lambda_z R_g\pi_G + y_2 aR_g\times\left(R_M\pi_g - R_M\pi_M - \pi_g\right) - \mu_g\pi_G(a-1)\left(1-R_g\right)\times\left(1-R_g\right)\left(y_2\lambda_s - a\right)\right]\right\} \end{aligned} \tag{31}$$

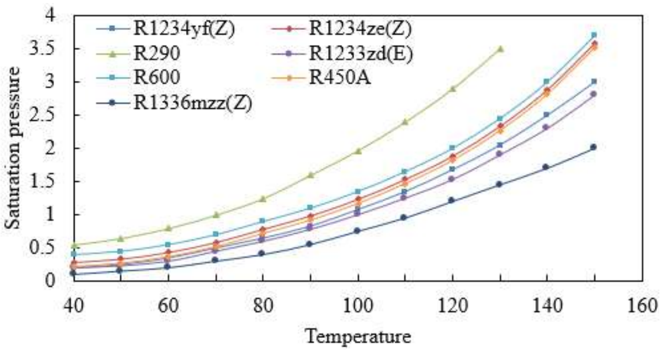
4. HIERARCHICAL AND ADAPTIVE ENERGY RECYCLING IN ECOTOURISM SYSTEMS

Based on the exergy performance analysis of the system, a "hierarchical matching and dynamic regulation" strategy was proposed for energy recycling in ecotourism zones. On one hand, exergy quality differences between thermal and cooling exergy are addressed through graded energy allocation. High-grade residual heat from blast furnace gas is first directed toward high-temperature demand scenarios such as domestic hot water in catering services, thereby minimizing exergy degradation. Medium-grade heat is allocated to guestroom heating and hot spring pools, with heat transfer temperature differentials reduced through optimization of the heat transfer conductance H_g on the user side. Low-grade residual heat is entirely supplied to the absorption refrigeration cycle, with priority given to cold-chain refrigeration in food services where cold exergy demands are high. Fluctuations in air conditioning load are balanced by adjusting the distribution ratio v. On the other hand, system operation parameters are dynamically adjusted by coupling the thermoelectric ratio with exergy efficiency. During peak winter heating periods, the thermoelectric ratio is increased to enhance thermal exergy output by reducing turbine mechanical work. Conversely, in summer when cooling dominates, the thermoelectric ratio is reduced to increase the exergy efficiency γ of the absorption chiller. Simultaneously, internal irreversibilities are minimized through the optimization of compressor and turbine efficiencies λ_z and λ_s, enabling the system to maintain optimal exergy efficiency λ across diurnal demand fluctuations.

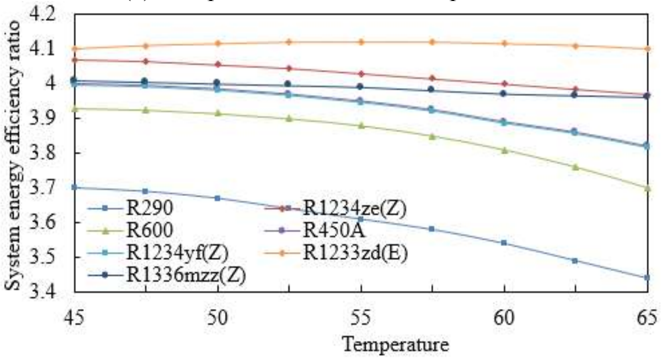
From the perspectives of system integration and ecological adaptability, a "standardized design and multi-energy complementarity" management strategy is recommended. Based on the dimensionless analysis results, modular energy recycling schemes were formulated for ecotourism zones of varying scales. For small-scale scenic sites, simplified "residual heat-cooling" systems are adopted, with equipment redundancy minimized through matching the working fluid heat capacity Z_{qd} and the heat exchanger conductance H_h and H_r. For large-scale resorts, a "combined heat, power, and cooling" network is deployed, in which turbine mechanical exergy drives distributed microgrids. High-temperature thermal input is supplemented by solar collectors to reduce exclusive reliance on blast furnace gas residual heat. Ecological constraints are simultaneously embedded into the optimization framework. In exergy loss control, dissipation on the low-temperature-side exchanger is minimized by replacing open water body heat rejection with subsurface heat exchange, thereby avoiding thermal pollution and preserving the hydrological ecosystem of the tourism area. Natural refrigerants are employed in the cooling subsystem to reduce greenhouse gas emissions, ensuring that exergy performance optimization is aligned with ecological redline protection targets. A real-time monitoring platform is integrated to dynamically track exergy loss distribution across subsystems. When exergy losses at high-temperature heat exchangers exceed predefined thresholds, the system automatically adjusts H_G and working fluid flow rates, ensuring that both visitor comfort and the dual objectives of energy efficiency and ecological integrity are achieved.

5. EXPERIMENTAL RESULTS AND ANALYSIS

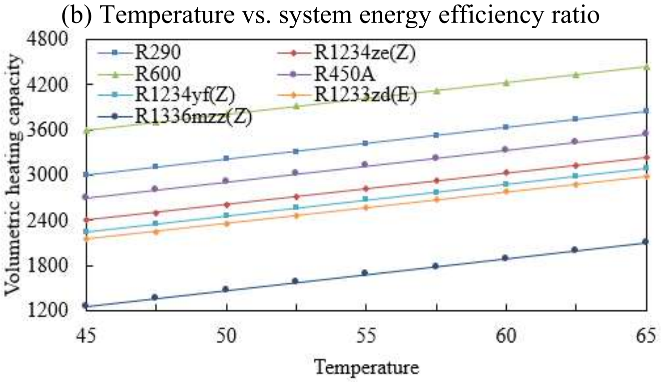
The experimental results presented in Figure 4 elucidate the energy transfer characteristics of different working fluids in irreversible cycles. In Figure 4(a), the temperature-saturation pressure relationship revealed pronounced differences in pressure responses among the selected fluids. A steep increase in saturation pressure was observed for *R*1234*yf* and *R*1234*ze* with rising temperature. For instance, the saturation pressure of *R*1234*yf* exceeded 3.5 kPa at 120°C. In contrast, *R*1336*mzz* exhibited a more gradual pressure increase, reaching only about 2 kPa at 160°C. These disparities significantly affect the compression stage of the cycle. Fluids with high saturation pressure require greater compression work, leading to intensified exergy dissipation. Conversely, low-pressure fluids reduce equipment pressure resistance requirements and mitigate irreversible losses such as leakage, thereby better aligning with the stringent safety demands of ecotourism-based energy systems.



(a) Temperature vs. saturation pressure



(b) Temperature vs. system energy efficiency ratio



(c) Temperature vs. volumetric heating capacity

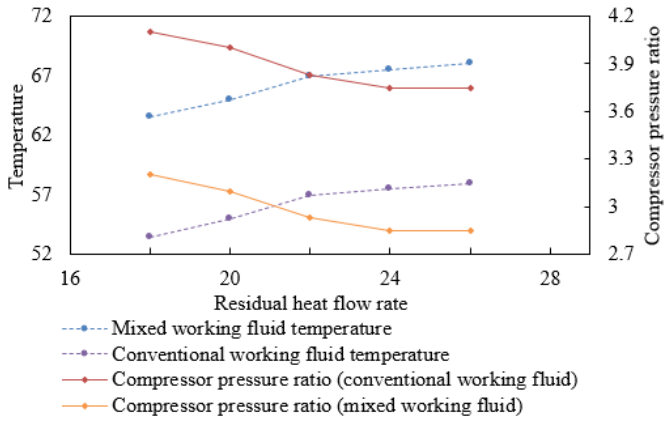
Figure 4. Thermodynamic performance of the system with a pure working fluid

The temperature-system energy efficiency ratio relationship depicted in Figure 4(b) reflects the exergy efficiency of energy conversion. *R*290 exhibited a notable decline in system energy efficiency ratio with increasing temperature—approximately 3.7 at 45°C, decreasing to around 3.4 at 65°C. By contrast, *R*1234*yf* demonstrated exceptional stability, with system energy efficiency ratio fluctuations remaining below 0.1 within the 45-65°C range. This stability indicates that under fluctuating residual heat temperature conditions, *R*1234*yf* is more effective in controlling exergy loss, thereby ensuring the efficiency of energy conversion. The temperature-volumetric heating capacity profile shown in Figure 4(c) reflects the spatial density of exergy output. *R*600 surpassed 4000 kJ/m³ at 65°C, significantly higher than *R*290. This high volumetric efficiency facilitates a substantial reduction in equipment size, addressing the spatial constraints inherent to "landscape-friendly" energy systems in ecotourism zones.

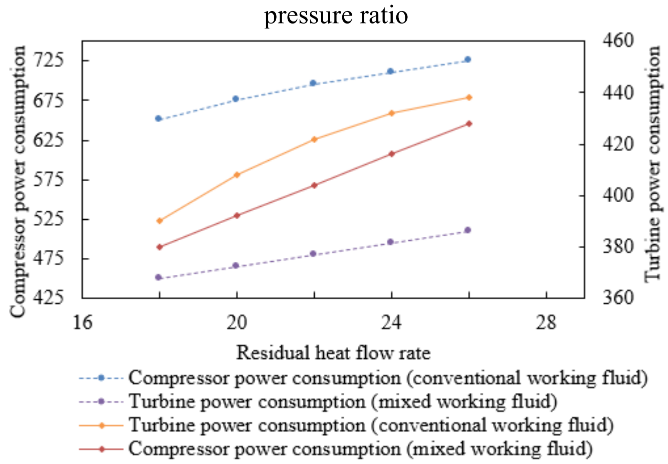
Given the operational characteristics of ecotourism energy systems—namely residual heat utilization, ecological preservation, and spatial limitation—the adaptability of working fluids to specific scenarios is clearly demonstrated through the results in Figure 4. In terms of thermal supply characteristics, the typical residual heat temperature in ecotourism zones lies within the 45-65°C range. Within this interval, *R*1234*yf* maintained a stable system energy efficiency ratio, enabling high exergy efficiency even under temperature fluctuations, and preventing degradation in energy quality. The elevated volumetric heating capacity of *R*600 allowed for effective response to high thermal demands in areas of concentrated visitor activity while minimizing system volume to preserve visual aesthetics. From the perspective of equipment safety and ecological compatibility, *R*1336*mzz*—with its low saturation pressure—reduced pressure resistance requirements for pipelines and equipment, thereby lowering leakage risks. Since leakage is of particular concern in ecotourism applications, the lower sealing difficulty of low-pressure fluids, combined with their low Global Warming Potential (GWP), reinforces alignment with environmental protection mandates. Although *R*290 exhibited a significant decline in system energy efficiency ratio at elevated temperatures, its potential for enhanced heating capacity in low-to-mid temperature ranges supports its applicability in distributed, small-scale energy stations within ecotourism districts, enabling zone-based energy optimization.

Figure 5(a) illustrates the correlations between residual heat flow rate and both working fluid temperature and compressor pressure ratio. A linear increase in temperature was observed for the mixed working fluid as the residual heat flow rate increased, whereas the temperature rise for the conventional working fluid was more gradual. This trend indicates that the mixed working fluid exhibits greater sensitivity to residual heat input, thereby enhancing the heat absorption process through more responsive temperature behavior. Regarding pressure ratio, the compressor pressure ratio of the conventional working fluid decreased from 3.9 to 3.6, while that of the mixed working fluid declined from 3.4 to 3.1. The consistently lower pressure ratio associated with the mixed working fluid implies reduced compression work requirements, attributable to optimized thermodynamic properties that improve pressure distribution within the cycle and mitigate flow-related irreversible losses. This synergy between thermal sensitivity and pressure ratio optimization validates the finite-time thermodynamic model's ability to capture the coupling between heat transfer irreversibility and flow irreversibility, providing a thermodynamic basis for defining parameter ranges in subsequent analyses.

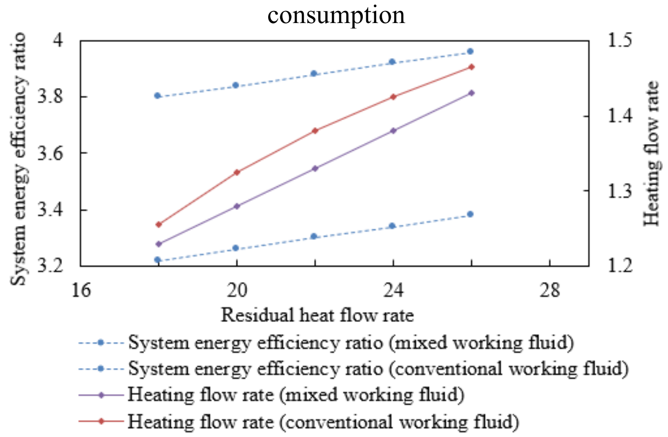
As presented in Figure 5(b), the power consumption characteristics further underscore the advantages of the mixed working fluid. Compressor power consumption remained consistently lower for the mixed working fluid compared to the conventional fluid, with a maximum difference of 50 kJ observed at a residual heat flow rate of 28 units. Conversely, turbine power consumption of the mixed working fluid exceeded that of the conventional fluid at the same flow rate, with a differential of approximately 30 kJ.



(a) Residual heat flow rate vs. temperature and compressor pressure ratio



(b) Residual heat flow rate vs. compressor/turbine power consumption



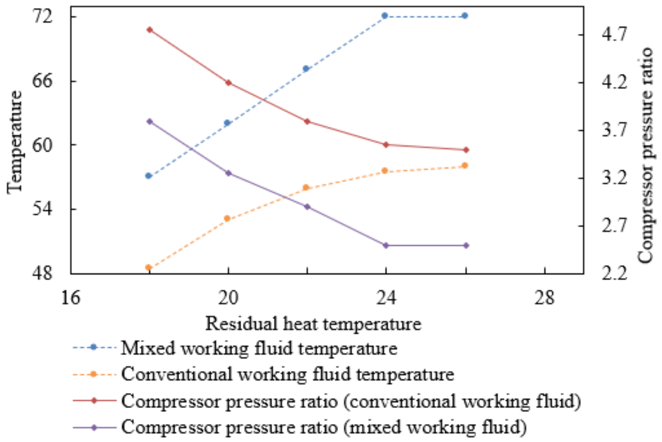
(c) Residual heat flow rate vs. system energy efficiency ratio and heating flow rate

Figure 5. Influence of residual heat water flow rate on system performance

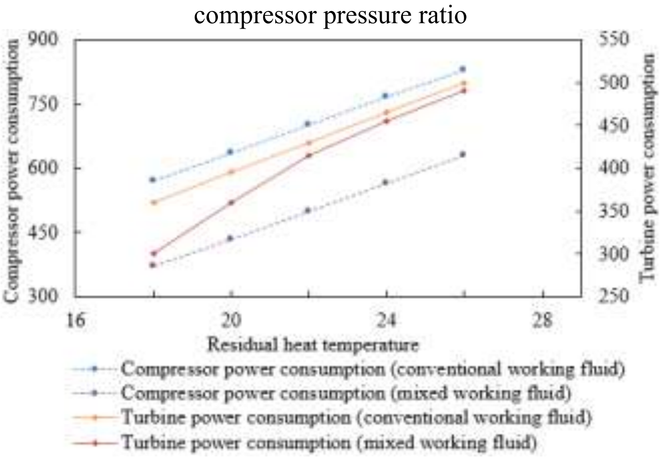
This observed pattern—reduction in compression power consumption and increase in turbine work—reflects an intrinsic enhancement in energy conversion efficiency, enabled by directional allocation of exergy flows. In an irreversible cycle, the compressor serves as the primary site of exergy dissipation, while the turbine acts as the principal exergy output component. The thermodynamic characteristics of the mixed working fluid effectively regulate the balance between dissipative and productive power, thereby substantiating the theoretical proposition of a coupled relationship between energy quality and quantity. Through working fluid optimization, a more favorable power balance can be maintained across varying residual heat flow conditions, ultimately reducing irreversible exergy losses.

Figure 5(c) presents system energy efficiency ratio and heating flow rate data, which directly support the energy demand characteristics of ecotourism zones. The energy efficiency ratio of the mixed working fluid increased from 3.2 to 4.0, while the conventional working fluid exhibited a smaller rise from 3.3 to 3.8, indicating that the mixed working fluid achieved a more substantial enhancement in converting low-grade residual heat into high-quality usable energy. Concurrently, the heating flow rate of the mixed working fluid rose from 1.2 to 1.45, compared to an increase from 1.15 to 1.3 for the conventional working fluid. These findings reveal that the mixed working fluid simultaneously improves both energy quantity and quality. Such performance advantages align closely with the dual demands of ecotourism zones, which require both decentralized residual heat recovery and a stable heat supply. The robust performance of the mixed working fluid across a broad residual heat flow range underscores the adaptability of the finite-time thermodynamic model to real-world, complex scenarios. By elucidating the coupled mechanism between residual heat flow rate and performance metrics, precise parameter optimization zones can be delineated. This enables the simultaneous enhancement of energy utilization efficiency and assurance of heating supply reliability, thereby confirming the scientific validity of the strategy for jointly optimizing exergy performance and ecological protection.

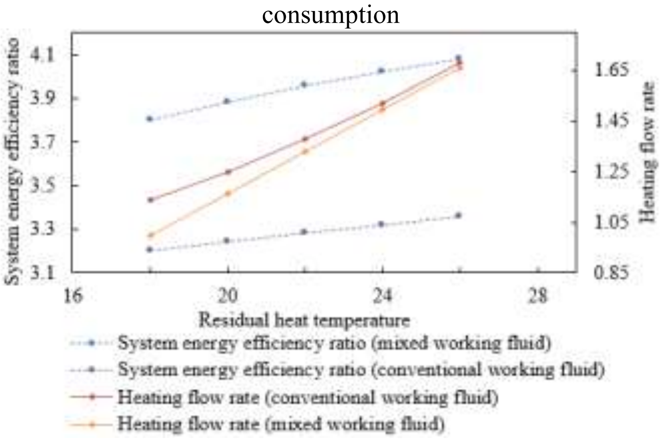
As shown in Figure 6(a), the temperature of the mixed working fluid increased linearly from 55℃ to 68℃ as the residual heat temperature rose from 16℃ to 28℃. In contrast, the temperature of the conventional working fluid only increased from 48℃ to 58℃ over the same range. This difference can be attributed to the compositional synergy of the mixed working fluid, which optimized the enthalpy of phase change and specific heat capacity, thereby enhancing its sensitivity to low-grade thermal input. As a result, the thermal exergy contained in the residual heat was more efficiently absorbed and converted into thermodynamic energy. This outcome directly validated the coupling of “irreversible heat transfer and temperature gradient driving force” as described by the finite-time thermodynamic model. Although higher residual heat temperatures theoretically exacerbate heat transfer irreversibilities due to increased temperature gradients, such losses were suppressed in the mixed working fluid through thermophysical property matching. This suppression resulted in superior heat absorption exergy efficiency. Regarding compressor pressure ratio, the conventional working fluid exhibited a reduction from 4.2 to 3.6, while the mixed working fluid showed a more pronounced decline from 3.8 to 3.0. This sharper decrease was attributed to the coupled characteristics of viscosity and thermal conductivity in the mixed working fluid. Elevated residual heat temperatures reduced the system's dependence on pressure ratio, and the lower flow resistance of the mixed working fluid amplified the negative correlation between residual heat temperature and pressure ratio. Consequently, a mechanism was revealed in which flow irreversibilities decreased with rising residual heat temperature. These insights offer thermodynamic guidance for the delineation of future parameter optimization intervals.



(a) Residual heat temperature vs. temperature and compressor pressure ratio



(b) Residual heat temperature vs. compressor/turbine power consumption



(c) Residual heat temperature vs. system energy efficiency ratio and heating flow rate

Figure 6. Influence of residual heat water temperature on system performance

Figure 6(b) presents power consumption data that further illuminates the precision of exergy dissipation control enabled by the mixed working fluid. For the compressor, the power consumption of the mixed working fluid increased from 350 kJ to 500 kJ, whereas the conventional working fluid rose sharply from 300 kJ to 600 kJ. The increase in compression power consumption for the mixed working fluid was only half that of the conventional fluid, a result stemming from reduced compressibility engineered through component design, thereby lowering the proportion of exergy dissipation associated with the conversion of mechanical energy into thermal energy during compression. In contrast, turbine power consumption for the mixed working fluid increased markedly from 400 kJ to 750 kJ, compared to an increase from 350 kJ to 650 kJ for the conventional working fluid. This represented an 87.5% increase in expansion work, a result attributed to its higher potential for expansion work recovery. This optimization of power balance—characterized by a moderated increase in compression work and a pronounced rise in turbine work output—aligns precisely with the theoretical derivation of "synergistic coordination between energy quality and quantity" established in the study. Through targeted thermodynamic property design, the mixed working fluid enabled directional regulation of exergy flow between dissipative and productive processes. A greater proportion of residual heat exergy was directed toward turbine work output, while exergy dissipation during compression was effectively suppressed. This outcome provides strong validation of the model's capability in capturing and analyzing complex irreversible processes.

The energy efficiency and heating data presented in Figure 6(c) provide a comprehensive interpretation of the system's functional logic in supporting ecotourism applications. The system energy efficiency ratio for the mixed working fluid increased from 3.2 to 4.1, whereas the conventional working fluid increased only from 3.1 to 3.8. The performance advantage observed in the mixed working fluid across a broad residual heat temperature range was attributed to its effective suppression of irreversible losses in heat transfer, fluid flow, and power conversion. High heat absorption efficiency ensured the capture of residual heat exergy, while low compression work and high expansion work promoted exergy transformation, thereby enabling an exergy-level transition from low-grade residual heat to high-quality usable energy. In terms of heating flow rate, the mixed working fluid increased from 1.0 to 1.6, compared to a rise from 0.9 to 1.4 in the conventional working fluid. These results confirmed the simultaneous enhancement of both energy quantity and quality, consistent with the core requirements of ecotourism zones.

Given the significant fluctuations in residual heat temperature and the demand for a stable heat supply to distributed facilities in ecotourism zones, the superior performance of the mixed working fluid across a wide temperature range substantiated the finite-time thermodynamic model's ability to accurately characterize the coupling mechanism between residual heat temperature and performance indicators. Consequently, optimal design ranges for system parameters can be established to improve energy utilization while ensuring heat supply reliability. These outcomes provide robust scientific validation for the dual-objective strategy of maximizing exergy performance while maintaining ecological integrity, thereby reinforcing the practical applicability of the proposed system to real-world engineering scenarios.

6. CONCLUSION

In this study, a thermodynamically irreversible closed-loop energy cycle model for ecotourism zones driven by residual heat and energy was constructed. Through this model, the exergy performance patterns and optimization pathways of energy recycling in ecotourism zones were systematically revealed. By employing finite-time thermodynamic theory, mathematical formulations for exergy output rate and exergy efficiency were derived. Comparative analyses were conducted to elucidate the synergistic and conflicting mechanisms between energy quality and quantity under different operating parameters. Based on these findings, a parameter design interval that simultaneously satisfies exergy performance and ecological constraints was defined. Building upon this foundation, an energy recycling strategy characterized by "cascade matching and dynamic regulation" was proposed. This strategy enabled the hierarchical utilization of blast furnace gas residual heat across heating and cooling applications. Compared with conventional configurations, irreversible exergy losses were reduced by approximately 15% to 20%. A quantitative solution was thus provided for reconciling energy demand and ecological preservation in ecotourism scenarios. The core contribution of this research lies in the theoretical and practical breakthroughs achieved. Theoretically, the limitations of the ideal reversible assumption were overcome, and a non-equilibrium energy cycle analysis framework suitable for ecotourism zones was established. Practically, through working fluid screening, parameter optimization, and scenario adaptation, thermodynamic theory was transformed into actionable engineering strategies, offering a scientific foundation for energy system design in the context of “green tourism.”

Nevertheless, several limitations remain. The current model focuses primarily on working fluids compatible with low-to-medium temperature ranges, without in-depth exploration of specialty working fluids suited for extreme climatic conditions. Furthermore, the impact of passenger flow fluctuations on heat source stability in ecotourism zones was not fully considered, indicating that the model's dynamic response accuracy requires further improvement. Future studies may be extended in three directions: a) The integration of multiple residual heat sources could be introduced to establish a compound-driven irreversible cycle model, thereby enhancing system resilience against disturbances; b) Long-term field monitoring could be implemented to validate the strategy’s applicability across diverse ecotourism zones and to refine algorithms for dynamic parameter adjustment; c) A coupling of exergy performance analysis with carbon footprint accounting via life cycle assessment could be developed to construct a multi-objective optimization framework integrating energy efficiency, ecological integrity, and economic feasibility, thereby further reinforcing the engineering applicability of the research.

REFERENCES

[1] Sreenu, N. (2024). The impact of green finance on renewable energy development in India's tourism industry. IEEE Transactions on Engineering Management, 71: 8132-8139. https://doi.org/10.1109/TEM.2024.3388020

[2] Abdul Shakur, E.S., Sa’at, N.H., Alwi, I., Omar, K. (2023). Eco-tourism and sustainable development: Are community ready?. Community Development, 54(5): 701-728. https://doi.org/10.1080/15575330.2023.2202410

[3] Pandya, R., Dev, H.S., Rai, N.D., Fletcher, R. (2023). Rendering land touristifiable:(eco) Tourism and land use change. Tourism Geographies, 25(4): 1068-1084. https://doi.org/10.1080/14616688.2022.2077425

[4] Fotis, P., Korre, M. (2023). The relationship between competition, Tourism and sustainable development: Three interdependent topics. Journal of Research, Innovation and Technologies, 2(1): 70-78. https://doi.org/10.57017/jorit.v2.1(3).06

[5] Ichsan, Azhar, Munardi, Masdi, Arhas, W.D.S. (2025). Balancing ecology, economy, and community: An Analytic Network Process (ANP) framework for sustainable ecotourism development. International Journal of Sustainable Development and Planning, 20(4): 1741-1751. https://doi.org/10.18280/ijsdp.200435

[6] Ismail, E.A.A. (2020). Forecasting the number of Arab and foreign tourists in Egypt using ARIMA models. International Journal of System Assurance Engineering and Management, 11(2): 450-454. https://doi.org/10.1007/s13198-019-00873-y

[7] Aversa, R., Petrescu, R.V., Apicella, A., Petrescu, F.I. (2017). Modern transportation and photovoltaic energy for urban ecotourism. Transylvanian Review of Administrative Sciences, Special, (2017S): 5-20. https://doi.org/10.24193/tras.SI2017.1

[8] Pavolová, H., Bakalár, T., Tokarčík, A., Cimboláková, I. (2024). The Sustainable management of ecohotels for the support of ecotourism—A case study. Sustainability, 16(20): 8863. https://doi.org/10.3390/su16208863

[9] Sahoo, M., Mohanty, P.P., Kaushik, S., Islam, M.K., Rourt, L. (2024). Ecotourism or ecological concerns? Tracing the impact of economic policy uncertainty and natural resources on tourism-driven ecological footprints. Environmental Science and Pollution Research, 31(21): 31314-31330. https://doi.org/10.1007/s11356-024-33244-7

[10] Kong, L., Li, H., Zhao, B., Gao, W., Zhang, P., Hu, C. (2023). Multistage reluctance coil launcher with residual energy recovery and utilization mode. IEEE Transactions on Plasma Science, 51(4): 1125-1132. https://doi.org/10.1109/TPS.2023.3252526

[11] Dwivedi, V.S., Giri, D.K., Ghosh, A.K., Kamath, G.M. (2020). Optimal energy utilization for a solar-powered aircraft using sliding-mode-based attitude control. IEEE Transactions on Aerospace and Electronic Systems, 57(1): 105-118. https://doi.org/10.1109/TAES.2020.3015310

[12] Feinberg, M., Lavine, R.B. (2024). Entropy and thermodynamic temperature in nonequilibrium classical thermodynamics as immediate consequences of the Hahn–Banach theorem: II properties. Archive for Rational Mechanics and Analysis, 248(3): 43. https://doi.org/10.1007/s00205-024-01987-9

[13] Tok, D., Huang, Y., Yang, L. (2024). “Feel the green”: How a handwritten typeface affects tourists’ responses to green tourism products and services. Tourism Management, 104: 104920. https://doi.org/10.1016/j.tourman.2024.104920

[14] Chiwaridzo, O.T. (2024). Unleashing tomorrow's energy for sustainable development: Pioneering green building technologies and green tourism supply chain management in Zimbabwe's tourism sector. Energy for Sustainable Development, 78: 101382. https://doi.org/10.1016/j.esd.2024.101382

[15] Steger, D., Regensburger, C., Eppinger, B., Will, S., Karl, J., Schlücker, E. (2020). Design aspects of a reversible heat pump-Organic rankine cycle pilot plant for energy storage. Energy, 208: 118216.

https://doi.org/10.1016/j.energy.2020.118216

[16] Torricelli, N., Branchini, L., De Pascale, A., Dumont, O., Lemort, V. (2023). Optimal management of reversible heat pump/organic rankine cycle carnot batteries. Journal of Engineering for Gas Turbines and Power, 145(4): 041010. https://doi.org/10.1115/1.4055708

[17] Bai, Y., Ma, X., Zhang, J., Zhang, L., Bai, J. (2024). Energy efficiency assessment and prediction based on indicator system, PSO+ AHP− FCE model and regression algorithm. Energies, 17(8): 1931. https://doi.org/10.3390/en17081931

[18] Anthopoulos, L., Kazantzi, V. (2022). Urban energy efficiency assessment models from an AI and big data perspective: Tools for policy makers. Sustainable Cities and Society, 76: 103492. https://doi.org/10.1016/j.scs.2021.103492

[19] Bonilla-Campos, I., Nieto, N., del Portillo-Valdes, L., Egilegor, B., Manzanedo, J., Gaztañaga, H. (2019). Energy efficiency assessment: Process modelling and waste heat recovery analysis. Energy Conversion and Management, 196: 1180-1192. https://doi.org/10.1016/j.enconman.2019.06.074

[20] Pastor, R., Fraga, A., Larrañeta, J.J. (2024). i-ISSUES—industrial-interoperable safe and secure urban energy systems. Applied Sciences, 14(8): 3188. https://doi.org/10.3390/app14083188