Postprint of Research on Combined Power and Cooling Cogeneration Systems for Recovering Valuable Components from Turbine Exhaust Gas

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Abstract

This paper proposes and investigates a novel combined power and cooling system that recovers effective components from turbine exhaust. This system achieves dual coupling of energy and material between the power cycle and refrigeration cycle through cascade utilization of thermal energy and recovery of effective components from high-temperature, high-pressure superheated ammonia-water vapor. The concentration of effective components in the turbine exhaust after partial condensation is significantly increased, and recovering these components can effectively increase the system' s cooling capacity. The equivalent work efficiency of the novel system is 20.19%, which represents a 44.32% improvement over conventional separate production systems, with a relative energy saving rate of 31.61%. Compared with conventional open-type and closed-type combined power and cooling systems, the improvement in equivalent work efficiency of the novel system is 7.28% and 17.04%, respectively. The energy saving mechanism of the system is elucidated through investigating the energy transfer and conversion processes. A new concept of average energy grade difference ΔA is proposed to analyze the utilization of input exergy and the potential for further improvement of thermodynamic performance. Additionally, the coupling characteristics between the power cycle and refrigeration cycle in the system are preliminarily explored.

Full Text

Preamble

Recovery of Active Ingredients from Turbine Exhaust in a Power/Refrigeration Cogeneration System

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Abstract: This paper proposes and investigates a novel power/refrigeration cogeneration system that recovers active ingredients from turbine exhaust vapor. Through cascade utilization of thermal energy and recovery of active ingredients from high-temperature, high-pressure superheated ammonia-water vapor, the system achieves dual coupling of energy and materials between the power generation cycle and refrigeration cycle. After partial condensation, the concentration of active ingredients in the turbine exhaust increases significantly, and recovering these ingredients effectively enhances the system' s refrigeration capacity. The equivalent work efficiency of the proposed system is 20.19%, representing a 44.32% improvement over conventional separated systems, with a relative thermal energy saving ratio of 31.61%. Compared with conventional open and closed power/refrigeration cogeneration systems, the equivalent work efficiency improvements are 7.28% and 17.04%, respectively. The energy-saving mechanism is elucidated through analysis of energy transfer and conversion processes. A new concept of "average energy level difference" (ΔA) is proposed to analyze the utilization of input exergy and the potential for further thermodynamic performance improvement. Additionally, the coupling characteristics between the power generation and refrigeration cycles are preliminarily explored.

Keywords: power/refrigeration cogeneration; cycle coupling; thermodynamic analysis; active ingredient recovery; cascade utilization

0 Introduction

In traditional power generation technologies such as gas turbines and internal combustion engine generator sets, approximately two-thirds of the energy input is discharged as heat to the environment [1]. Recovering this waste heat can effectively reduce fossil fuel consumption and improve energy utilization efficiency. While using recovered waste heat for direct heating through heat exchangers is straightforward, converting waste heat into work or cooling is relatively more complex, and standalone waste heat power generation or refrigeration typically exhibits low efficiency.

Power/refrigeration cogeneration systems can efficiently convert thermal energy into both work and cooling through organic coupling of power cycles and refrigeration cycles [2]. These systems can serve as bottoming cycles to recover waste heat from conventional power cycles (e.g., gas turbines, internal combustion engines) or operate as independent systems utilizing industrial waste heat, solar energy, or geothermal energy. Goswami et al. [3] proposed a cogeneration system that replaces the condenser and throttling valve in an ammonia-water absorption refrigeration cycle with an expander, enabling simultaneous work and cooling production from a single cycle, known as the Goswami cycle. Zheng Danxing et al. [4] proposed a novel absorption power/refrigeration combined cycle based on the Kalina cycle, adding a condenser and evaporator between the flash tank and second absorber of the original process, and replacing the flash tank with a rectification column to increase ammonia concentration in the vapor phase. The low-temperature evaporation heat absorption using enriched ammonia vapor after condensation and throttling enables simultaneous work and cooling output. Zhang Na et al. [5] introduced a steam turbine in the dilute solution branch of an ammonia-water absorption refrigeration cycle to replace the throttling valve, utilizing medium-temperature waste heat to heat the dilute solution and generate steam, thereby achieving both work and refrigeration simultaneously. This system is called a parallel cycle, and series and mixed configurations were also proposed. Han Wei et al. [6, 7] investigated the coupling relationship between power cycles and refrigeration cycles, proposing open and closed cooling cogeneration cycles that improved waste heat utilization efficiency.

In power/refrigeration cogeneration systems using the same working fluid for both cycles, the streams can be connected and exchanged, enhancing overall thermodynamic performance through material coupling. In ammonia-water absorption refrigeration cycles, ammonia serves as the active refrigeration ingredient. In the turbine exhaust of ammonia-water mixture Rankine cycles, ammonia-water vapor at certain pressure undergoes partial condensation, significantly increasing ammonia concentration in the vapor phase and raising the partial pressure of the active refrigeration component, thereby enhancing refrigeration potential. Recovering this ammonia can increase the refrigerant flow rate in the reverse cycle, thus augmenting refrigeration capacity. Based on this principle, this paper proposes a novel open power/refrigeration cogeneration system that recovers active ingredients from turbine exhaust, evaluates its thermodynamic performance, and further analyzes the system' s energy-saving mechanism.

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1 System Process and Characteristics

1.1 Novel Power/Refrigeration Cogeneration System

The process flow of the proposed open power/refrigeration cogeneration system with active ingredient recovery from turbine exhaust is shown in [Figure 1: see original paper]. The system features two main characteristics: (1) Following the principle of "temperature matching and cascade utilization," the high-temperature portion of the external heat source is used for the power sub-cycle, while the low-temperature portion is used for refrigeration, with waste heat from the power sub-cycle also utilized in the refrigeration sub-cycle; (2)

The active refrigeration working fluid in the turbine exhaust of the power sub-cycle is recovered for further utilization in the reverse cycle, and the turbine exhaust pressure is not constrained by the rectification process pressure of the reverse cycle.

The system comprises two parts: a power cycle (dashed lines) and a refrigeration cycle (solid lines), with both energy and material coupling between the sub-cycles. The concentrated solution (stream 1) flowing from the absorber bottom splits into two streams: one entering the power sub-cycle and the other entering the absorption refrigeration sub-cycle. The stream entering the power sub-cycle (stream 2) is pressurized by a high-pressure solution pump, heated by the higher-temperature portion of flue gas in the vapor generator to form high-temperature, high-pressure ammonia-water mixture vapor (stream 4), and then expanded in the turbine to produce work. After passing through the reboiler and heat exchanger, the turbine exhaust undergoes partial condensation to form a gas-liquid mixture, which then enters a gas-liquid separator. The higher-concentration ammonia vapor at the top of the separator is compressed and sent to the rectification column, while the dilute solution at the bottom preheats the rectification column feed through the solution heat exchanger before merging with the dilute solution from the rectification column bottom and entering the absorber.

The stream entering the absorption refrigeration sub-cycle (stream 13) is first pressurized by a low-pressure solution pump and then splits into streams 15 and 18. Stream 15 is preheated through the solution heat exchanger, recovers lower-temperature heat from flue gas in the flue gas heat exchanger, and then enters the rectification column. Stream 18 sequentially recovers heat from turbine exhaust through the low-pressure solution heat exchanger and preheater before entering the rectification column. In the rectification column, the concentrated ammonia vapor from the compressor (stream 9) and the ascending ammonia vapor generated in the reboiler undergo sufficient heat and mass transfer with the preheated concentrated solutions (streams 17 and 20) to complete the rectification process, ultimately separating into ammonia vapor at the column top (stream 25) and dilute solution at the column bottom (stream 21).

The top ammonia vapor (stream 25) enters the condenser to form liquid ammonia (stream 26), which then enters the subcooler to recover partial cooling capacity from the ammonia vapor (stream 29) from the evaporator, forming subcooled liquid ammonia (stream 27). After throttling through the expansion valve (from condensation pressure to evaporation pressure), stream 27 forms a wet vapor that evaporates and absorbs heat in the evaporator, producing the refrigeration effect. The ammonia gas after refrigeration becomes superheated ammonia vapor (stream 30) through the subcooler and then enters the absorber. The column bottom dilute solution (stream 21) is cooled by the concentrated solution from the absorber through the solution heat exchanger, merges with the dilute solution from the gas-liquid separator bottom, is depressurized through the pressure reducing valve (from generation pressure to absorption pressure), and enters the absorber. In the absorber, the dilute solution absorbs ammonia gas to become concentrated solution, initiating the next cycle. The absorption heat released during this process is removed by cooling water circulation.

1.2 Reference System

The energy input to the aforementioned power/refrigeration cogeneration system is medium-low temperature heat, with outputs of work and cooling. To reasonably evaluate and analyze its thermodynamic performance, this study employs a conventional separated system as the reference system. The separated power cycle uses a simple Rankine cycle with water as the working fluid, while the separated refrigeration cycle uses a single-stage ammonia-water absorption refrigeration cycle, as shown in [Figure 2: see original paper]. The heat source enters the waste heat boiler of the power cycle and the reboiler of the refrigeration cycle independently, with the two cycles operating separately to output work and cooling capacity, respectively. The reference system has the same types of energy input and output as the cogeneration system and is simulated under identical thermal boundary conditions.

In this study, both the proposed system and reference system are simulated using Aspen Plus [8, 9]. The model is established based on mass conservation, energy conservation, and component conservation, with relative errors not exceeding 0.01%. In the simulation of the proposed system, the PSRK (Predictive Soave-Redlich-Kwong) equation of state is used to calculate thermodynamic parameters of NH -H O solutions [10, 11], while the Steam-TA equation of state is used for H O thermodynamic parameters. These property methods yield calculation results in good agreement with IIR (International Institute of Refrigeration) property data [12].

The external heat source suitable for the new system is waste heat from distributed energy systems, such as exhaust from internal combustion engines or gas turbines. Gas internal combustion engines are widely used in distributed energy systems; therefore, this study uses gas engine exhaust as the system external heat source.

Based on relevant research literature [13, 14], basic parameter settings are established as shown in . To facilitate key issue investigation, the following simplifications are made to the system: (1) The system operates under steady-state conditions; (2) Pressure losses in pipelines and general heat exchangers are neglected; (3) Heat losses from components and pipelines are neglected; (4) Working fluids at reboiler, absorber, and condenser outlets are in saturated states; (5) The isentropic efficiency of the NH compressor is calculated according to the pressure ratio Rp as follows [15]:

2 System Simulation and Evaluation

2.1 Simulation Parameters

The isentropic efficiency of the compressor is given by: = -

2.2 Evaluation Criteria

Considering the different grades of work and cooling outputs, traditional first-law efficiency is not well-suited for multi-product systems. For reasonable system evaluation, the output cooling capacity can be equivalently converted to work output using the coefficient of performance of a mechanical compression refrigeration cycle (COPC), which together with the original work output constitutes the system product. Therefore, equivalent work efficiency is defined as:

$$W_{,eq} = (W_{TUR} - W_{COMP} - W_P + Q_C/COP_C) / Q_f$$

where WTUR represents turbine output work, WCOMP and WP represent mechanical compressor and solution pump power consumption, QC represents cooling capacity in the evaporator, and Qf represents thermal energy carried by flue gas.

To intuitively represent the advancement of the studied system, the relative thermal energy saving ratio (TESR) is also adopted as an evaluation metric. TESR examines the difference in thermal energy consumption between cogeneration and conventional separated systems based on the same work and cooling outputs. TESR is defined as the ratio of thermal energy consumption reduction in the cogeneration system to the energy consumption of the reference system:

$$TESR = (Q_{in,ref} - Q_{in}) / Q_{in,ref}$$

where Qin is the heat consumption of the cogeneration system; P and QC are the work and cooling outputs of the separated system, respectively; th is the thermal power generation efficiency of the separated work-producing system; and COPA is the performance coefficient of the separated waste heat refrigeration system. These efficiencies are calculated under the same thermal boundary conditions as the cogeneration system.

2.3 Exergy Analysis Method

Exergy analysis based on the second law of thermodynamics can reveal the magnitude of irreversible losses in each process, thereby elucidating the reasons for system performance improvement. The exergy balance expression for any control volume is:

$$\Sigma(m_{in} \cdot e_{in}) + \Sigma(1 - T_0/T_j) \cdot Q_j + W = \Sigma(m_{out} \cdot e_{out}) + E_{dest}$$

where subscripts in and out denote streams entering and leaving the control volume, W represents work flow across the control volume boundary, Qj represents heat flow across the control volume boundary at temperature Tj, e represents specific exergy accompanying material streams, and T0 is the environmental temperature (298.15 K) [16]. Specific exergy can be calculated as:

$$e = (h - h0) - T0 \cdot (s - s0)$$

(1) Exergy Efficiency

This paper also employs exergy efficiency to represent the utilization rate of input exergy, expressed as:

$$ex = (Wnet + EC) / Ef$$

where Wnet is the system net work output; EC is the system output cooling exergy, calculated based on the exergy change of refrigerant at the evaporator inlet and outlet:

$$EC = m\_ref \cdot [(h\_EV,in - h\_EV,out) - T0 \cdot (s\_EV,in - s\_EV,out)]$$

where m_ref is the refrigerant mass flow rate, and h_EV,in, s_EV,in and h_EV,out, s_EV,out are the specific enthalpy and entropy of refrigerant entering and leaving the evaporator, respectively.

In Eq. (6), Ef represents the system input heat exergy, calculated based on the enthalpy and entropy of flue gas heat source entering the system and at ambient conditions:

$$Ef = m\_f \cdot [(h\_f - h0) - T0 \cdot (s\_f - s0)]$$

(2) Exergy Destruction Coefficient

Exergy efficiency cannot directly indicate the distribution of exergy losses within the system or reveal weak links. Correspondingly, the proportion of exergy loss to input exergy can reveal the location and magnitude of exergy degradation, complementing exergy efficiency. This paper defines the exergy destruction coefficient for each component as:

$$i = E\_dest,i / Ef$$

(3) Key Process Energy Level Analysis

Conventional exergy analysis effectively characterizes irreversible loss magnitude but has limitations when analyzing system performance improvement potential. For instance, large exergy losses in a heat transfer process may result from large heat transfer quantities rather than process sub-optimization, potentially misleading conventional exergy analysis. Accordingly, this paper proposes the concept of average energy level difference ΔA, defined as:

$$\Delta A = \Delta E / \Delta H$$

where ΔE represents exergy loss in an energy transfer or conversion process, and ΔH represents the enthalpy change between the energy release and acceptance sides.

As a dimensionless parameter, the average energy level difference ΔA characterizes the irreversible loss per unit of energy transfer or conversion. A larger ΔA indicates poorer energy grade matching and greater potential for thermodynamic performance improvement.

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3 System Thermodynamic Performance

Under design conditions, the initial temperatures of heat source flue gas and cooling water are 350°C and 30°C, respectively, with refrigeration evaporation temperature set at -15°C. The waste heat boiler hot-end temperature difference is set at 30°C, with other parameters as assumed in . Simulation calculations yield parameters such as temperature, pressure, mass flow rate, enthalpy, and entropy at various system points, and further calculate the load of each main component as shown in .

The results are presented in . When the flue gas heat input to the cogeneration system is 358.83 kW, the turbine work output is 35.18 kW, compressor and pump power consumption are 3.06 kW and 3.64 kW, respectively, producing 28.48 kW net work and 127.01 kW cooling capacity (work-to-cooling ratio of 0.224), with an equivalent work efficiency of 20.19%. Under the same thermal boundary conditions, the conventional separated system requires 524.67 kW flue gas heat input to produce the same 28.48 kW net work and 127.01 kW cooling capacity, yielding an equivalent work efficiency of 13.80%. Specifically, the steam Rankine cycle consumes 210.60 kW flue gas heat, producing 30.97 kW turbine work (partially used for pump consumption), while the absorption refrigeration cycle consumes 314.07 kW flue gas heat, delivering 127.01 kW evaporator cooling capacity with a COP of 0.404.

Compared with the separated system, the cogeneration system' s equivalent work efficiency increases by 6.39 percentage points, achieving a relative thermal energy saving ratio of 31.61%.

The proposed open cogeneration system is compared with previously published conventional closed [6] and conventional open [7] power/refrigeration cogeneration systems. lists the thermodynamic performance parameters of the conventional systems under identical thermal boundary conditions. With the same heat source input, the new system's cooling capacity slightly decreases compared with the conventional open system, from 133.65 kW to 127.01 kW (equivalent to 2.30 kW reduction in equivalent work). However, net work output increases by 33.71% (7.18 kW). Consequently, the work-to-cooling ratio increases, and the equivalent work efficiency improves by 7.28% over the conventional open system, with the relative energy saving ratio increasing by 4.51 percentage points. Compared with the conventional closed system, the new open system shows even greater performance improvements, with equivalent work efficiency and relative energy saving ratio increasing by 17.04% and 68.77%, respectively.

4 Analysis

4.1 System Energy Transfer and Conversion Process Analysis

The main heat transfer and energy conversion processes in the proposed open power/refrigeration cogeneration system are illustrated in [Figure 3: see original paper]. The higher-temperature portion of heat source flue gas (from 350°C to 170.8°C, 202.71 kW) heats high-pressure ammonia-water solution from 37.1°C to 320°C (state points 3-4) in the power cycle waste heat boiler, generating high-temperature, high-pressure superheated ammonia-water vapor. This vapor undergoes deep cascade utilization in both power and refrigeration cycles. In the flue gas heat exchanger, the lower-temperature portion of flue gas (from 170.8°C to 117.9°C, 58.46 kW) preheats rectification column feed in the refrigeration cycle, achieving heat recovery.

The high-temperature, high-pressure superheated ammonia-water vapor first expands through the turbine to produce work (state points 4-5). The turbine enthalpy drop is 35.18 kW, of which 6.70 kW satisfies compressor and solution pump consumption, leaving 28.48 kW as net system work output. State points 4-5' in [Figure 3: see original paper] represent the turbine expansion in the conventional open cogeneration system. To directly feed turbine exhaust into the refrigeration cycle rectification column, the exhaust pressure must be increased to match rectification pressure, resulting in relatively high exhaust pressure and temperature (1.356 MPa, 173.01°C). Under these conditions, ammonia-water vapor expansion is insufficient, and large heat transfer temperature differences in the rectification column increase irreversible losses. In contrast, the new system turbine exhaust can expand to lower pressure levels (0.6 MPa or lower), achieving larger specific enthalpy drops and significantly improved work capacity.

With turbine exhaust temperature and pressure of 141°C and 0.6 MPa, respectively, its condensation heat is highly suitable for driving the absorption refrigeration cycle. As shown in [Figure 3: see original paper], turbine exhaust sequentially recovers heat through the refrigeration cycle reboiler and preheater, cooling from 141°C to 120°C (state points 5-6, 84.34 kW) and finally to 95°C (state points 6-7, 38.07 kW). At 0.6 MPa turbine exhaust pressure, ammonia concentration in the vapor phase increases rapidly during condensation, as shown in [Figure 4: see original paper]. The ammonia component can be recovered and utilized in the refrigeration sub-cycle to increase system cooling capacity. The ammonia mass concentrations in the vapor phase of the gas-liquid mixture at state points 5, 6, and 7 are 0.396, 0.697, and 0.889, respectively, with increasing value for refrigeration cycle recovery. The new system separates the high-concentration ammonia vapor at state point 7 (state point 7' in [Figure 4: see original paper]) through the gas-liquid separator, compresses it to the absorption refrigeration cycle rectification pressure, and feeds it to the rectification column for further purification to meet refrigeration requirements. Since this involves rectification of superheated vapor, no additional heat consumption is required, only appropriate cooling load.

The dilute solution from the gas-liquid separator bottom preheats the rectification column feed solution. Throughout this process, high-temperature, high-pressure superheated ammonia-water vapor generated from flue gas waste heat first expands through the turbine to produce work, then its condensation heat is cascade-utilized in the refrigeration cycle, and finally its active ingredients are recovered to increase cooling capacity, achieving dual coupling of energy and materials between power and refrigeration cycles. The proposed open power/refrigeration cogeneration system has three characteristics: (1) Turbine exhaust pressure is not constrained by the refrigeration cycle, substantially increasing work capacity; (2) Following the principle of "temperature matching and cascade utilization," the higher-temperature portion of flue gas is utilized in the power cycle, while lower-temperature portions and turbine exhaust are cascade-utilized in the refrigeration cycle; (3) After partial condensation, active ingredient concentration in the exhaust increases significantly, and recovering these ingredients only requires work for pressure elevation, not heat consumption, effectively increasing system cooling capacity.

4.2 System Exergy Analysis Results

Exergy analysis reveals the exergy balance of the proposed open system and separated system, as shown in . Simulations ensure identical exergy outputs (48.04 kW) for both systems, comprising 28.48 kW mechanical exergy and 19.56 kW cooling exergy. Under these conditions, exergy inputs (all thermal exergy) are 112.52 kW and 164.77 kW, respectively, with the separated system requiring 46.44% more input. Internal mechanical power consumption is provided by the turbine, assuming no mechanical losses in work transmission. In the cogeneration system, total exergy loss in all components accounts for 57.31% of input exergy, 13.53 percentage points lower than the reference system. The exergy efficiencies of the cogeneration and separated systems are 42.69% and 29.16%, respectively.

System exergy losses can be divided into four main categories. [Figure 5: see original paper] and [Figure 6: see original paper] illustrate the distribution of exergy losses among specific components within each category and present the corresponding average energy level differences.

The higher exergy efficiency of the new system primarily results from reduced exergy losses in the first category (working fluid-to-heat source heat transfer process). In the cogeneration system, exergy loss in this portion accounts for only 15.18% of input exergy, less than half of that in the reference system (32.38%). Comparing [Figure 5: see original paper] and [Figure 6: see original paper] reveals that this is mainly because the reference system directly uses 350°C flue gas to drive the absorption refrigeration cycle, creating large temperature differences in the reboiler heat transfer process with an exergy destruction coefficient as high as 26.13% and correspondingly high average energy level difference ΔA (0.194), indicating significant energy-saving potential. In the new system, heat source flue gas drives the ammonia-water power cycle, effectively reducing heat transfer temperature differences, with ΔA in both waste heat boiler and flue gas heat exchanger at low levels (0.063 and 0.073, respectively), indicating good thermal grade matching.

In the second category, exergy loss in heat exchange with heat sinks in the separated system shows a flue gas loss exergy destruction coefficient of 9.93% with high ΔA (0.167). In the new system, the variable-temperature evaporation characteristics of ammonia-water mixture are fully utilized to effectively reduce exhaust gas temperature (117.9°C for cogeneration system vs. 181.0°C for SRC), reducing ΔA for exhaust gas loss to 0.075. Additionally, in both new and separated systems, the partial condenser shows low exergy destruction coefficient but high ΔA (0.145 and 0.139), suggesting potential for using this condensation heat to preheat rectification column feed and reduce irreversible losses.

The third category involves internal heat transfer process exergy losses. Although the cogeneration system has more internal heat exchange components, all maintain good energy matching (low ΔA), resulting in low total exergy destruction coefficient (9.43%) and minimal irreversible losses. The higher-temperature condensation heat carried by turbine exhaust provides thermal load for rectification in the reboiler, while lower-temperature condensation heat preheats rectification column feed, achieving cascade utilization. The fourth category comprises pressure regulation process exergy losses. Components in this category are mature mechanical equipment designed for specific pressure changes, with very limited potential for further exergy loss reduction and are generally not considered.

The analysis shows that most components in the proposed power/refrigeration cogeneration system exhibit low ΔA values, indicating good grade matching. Particularly, exergy losses in working fluid-to-heat source heat transfer and exhaust gas processes are substantially reduced compared with the separated system. The next potential improvement for the new system lies in recovering condensation heat from the partial condenser.

4.3 Cycle Coupling Characteristics Analysis

The study reveals that turbine exhaust pressure (p_TUR,in) and partial condensation temperature at the gas-liquid separator inlet (t_PC) serve as coupling point parameters between power and refrigeration cycles, significantly affecting active ingredient concentration and recovery rate at the gas-liquid separator vapor outlet. [Figure 7: see original paper] shows the effect of partial condensation temperature t_PC on system performance under different turbine exhaust pressures. For a given turbine exhaust pressure, equivalent work efficiency initially increases then decreases with rising t_PC, exhibiting an optimal value. The optimal t_PC corresponding to maximum equivalent work efficiency varies with exhaust pressure. Specifically, higher turbine exhaust pressure yields higher optimal t_PC. When turbine exhaust pressures are 0.4 MPa, 0.6 MPa, and 0.8 MPa, the optimal t_PC values are 75°C, 95°C, and 100°C, respectively. At p_TUR,in = 0.6 MPa and t_PC = 95°C, the proposed system achieves maximum equivalent work efficiency of 20.19%.

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5 Conclusions

This paper establishes a model for a novel open power/refrigeration cogeneration system with active ingredient recovery from turbine exhaust, evaluates its basic thermodynamic performance, and elucidates the energy-saving mechanism and potential for further performance improvement through energy transfer/conversion process analysis and exergy analysis. Additionally, the coupling characteristics between system cycles are preliminarily explored, providing theoretical basis for system optimization. The main conclusions are:

1) Simulation results show that under flue gas heat source temperature of 350°C, cooling water temperature of 30°C, and refrigeration evaporation temperature of -15°C, the proposed open power/refrigeration cogeneration system achieves an equivalent work efficiency of 20.19%, representing a 44.32% improvement over conventional separated systems with a relative thermal energy saving ratio of 31.61%. Compared with conventional open and closed power/refrigeration cogeneration systems, the equivalent work efficiency improvements are 7.28% and 17.04%, respectively.

2) Energy transfer and conversion process analysis reveals that the new system achieves dual coupling of energy and materials between power and refrigeration cycles through cascade utilization of thermal energy and active ingredient recovery from high-temperature, high-pressure superheated ammonia-water vapor. The higher- and lower-temperature portions of flue gas are cascade-utilized in power and refrigeration cycles, respectively. Turbine exhaust first undergoes cascade thermal energy utilization, followed by active ingredient recovery in the refrigeration cycle. After partial condensation, active ingredient concentration in the exhaust increases significantly. Recovering these ingredients only requires work for pressure elevation, not heat consumption, effectively increasing system cooling capacity.

3) Based on exergy analysis, the novel concept of average energy level difference ΔA is proposed. As a dimensionless parameter, ΔA characterizes irreversible loss per unit of energy transfer or conversion. Larger ΔA indicates poorer energy grade matching and greater potential for thermodynamic performance improvement. Comparative exergy analysis using ΔA clarifies the advantages of the new system in exergy utilization and its potential for further performance enhancement.

4) Turbine exhaust pressure and partial condensation temperature t_PC at the gas-liquid separator inlet, as coupling point parameters between power and refrigeration cycles, significantly affect active ingredient concentration and recovery rate. Higher turbine exhaust pressure corresponds to higher optimal t_PC. When turbine exhaust pressures are 0.4 MPa, 0.6 MPa, and 0.8 MPa, the optimal t_PC values are 75°C, 95°C, and 100°C, respectively.

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Note: Figure translations are in progress. See original paper for figures.