Research Article

Completely Recuperative Supercritical CO₂ Recompression Brayton/Absorption Combined Power/Cooling Cycle: Performance Assessment and Optimization

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Received 8 March 2022; Revised 12 April 2022; Accepted 20 April 2022; Published 20 May 2022

1. Introduction

Global environmental pollution issues concerning ozone depletion, excess emission of carbon dioxide, and growing demand for energy consumption have aroused widespread focus on the efficient utilization of primary energy and the development of sustainable energy use technologies [1, 2]. Efficient and low-carbon or even zero-carbon energy conversion systems that can meet different renewable energy requirements for multiple heat source applications have become an important research subject [3, 4]. In recent years, Supercritical Carbon Dioxide (SCO₂) has attracted the attention of many scholars as an efficient and pollution-free clean cycle working fluid, and research on the SCO₂ Brayton cycle has gradually become a hotspot [5, 6]. For instance, concentrated solar power (CSP) and nuclear power integrated SCO₂ Brayton cycle for distributed power systems have been promising application prospects in the increasingly severe global circumstances of implementation in carbon neutrality and carbon peaking [7, 8].

For major clean energy power generation applications such as geothermal power plants, natural gas stations, concentrated solar power plants, fuel cells, generation IV nuclear power plants, and fusion reactors, the SCO₂ Brayton cycle is considered to be the most promising alternative power system due to its layout simplicity, investment economy, size compactness, and intrinsic safety [9–12]. The SCO₂ Brayton cycle, first proposed by Feher and Angelino...
in 1967, was operated at temperatures and pressures above the critical point (31.1°C and 7.39 MPa, respectively) for the entire cycle [13, 14]. The temperature and pressure at the inlet point of the compressor were slightly higher than the critical point, and the density of carbon dioxide was close to that of liquid and difficult to be compressed; therefore, the size and power consumption can be significantly reduced. In recent years, many researchers and institutions built various engineering prototypes and verified the feasibility of the system [3, 5].

A simple Brayton cycle, recompression cycle, intercooling cycle, and reheating cycle are typical operation conditions aimed at efficiency increase in power generation, and in these cycles, the exhausting heat in the precoolers is ultimately discharged to the environment; although it is low-grade heat at a temperature of about 150°C, the heat transfer capacity in the precoolers is generally 1.5 times that of the power generation capacity [15].

Although the Brayton cycles have many advantages in terms of performance and compactness, the high cost of heat exchangers (about 40% of the total investment cost) and large low-grade heat loss in the precooler (exceeding 55% of overall heat capacity provided) restricted their further development, promotion, and application [15–17]. The used heat exchangers which are called printed circuit heat exchangers (PCHE) have characteristics of high temperature, pressure resistance, and compact structure and are extremely expensive for the required typical manufacturing process involving chemical etching and diffusion bonding [18, 19]. In addition, when integrated with different solar concentration technologies such as solar power tower solar dish, parabolic trough collector, and linear Fresnel collector for regions with rich solar resources, the large amount of exhaust low-grade waste heat in low-temperature recuperator (LTR) ranging from 100 to 180°C is frequently cooled by open cooling water circulation in PRC, which not only caused the system efficiency for solar energy utilization to decrease but also limited its commercial engineering applications in arid inland regions for a lot of valuable water resources are consumed [20].

To improve the performance of the Brayton cycle, various combined systems with functional outputs are introduced to realize a further utilization of rejected heat; therefore, low boiling working medium power systems are employed as the bottom cycle [21]. Among them are two typical power cycles: the organic-temperature Rankine cycle and ammonia-water cycle [22]. The organic Rankine cycle which employed isobutane, isopentane, R114, etc. has the drawbacks of high cost and environmental pollution. The ammonia-water cycle, on the other hand, exhibited a variable temperature boiling process, which allowed maintenance of a fixed temperature difference with limited heat sources and less exergy loss. However, the variable temperature condensation process resulted in a low turbine expansion ratio and inefficiency. Kalina [23] introduced a power cycle which substituted the condensation process with an absorption process, attracting wide attention.

Since cooling/heating demand accounted for 40–50% of domestic power consumption, for the low-grade exhaust heat in the Brayton cycle, heat-driven absorption refrigeration also can be a promising alternative [24, 25]. Moreover, the operational temperature, pressure, and concentration have been complementary with ammonia-water power cycles; therefore, combined power/cooling systems have been widely concerned [26].

However, simply analyzing the thermal efficiency of the system can easily result in higher system unit costs [27, 28]. To accurately describe the impact of energy quality and unit cost on system performance, a thermoeconomic analysis methodology that takes into account the required capital investment, exergy inefficiencies, and the cost rates associated with these inefficiencies should be introduced as the main evaluation criterion in evaluating and optimizing thermal systems [29].

Optimization of thermal systems merely from the perspective of thermodynamics would always result in a high cost for the total system [30]. To accurately evaluate and promote the system performance, thermoeconomic analysis in a synthesis of exergy and economics perspectives should be advocated to perform a high-efficient and low-cost design and operation. To ameliorate this deficiency, several typical methodologies from the thermoeconomic point of view were developed and applied in recent years, thermoeconomic functional analysis (TFA), specific exergy costing approach (SPECO), and theory of the exergetic cost (TEC), for instance, and all proposed methodologies can be employed in the analysis of thermal systems principally.

The above efforts highlight the performance promotion potential of the Brayton combined systems, and it seems that the Brayton cycle integrating with low-grade heat utilization cycles is the most promising alternative for different heat source utilization occasions. Despite the large amounts of research in the case of the Brayton combined systems and to the best of the authors’ knowledge, no attempt has been paid to achieve complete utilization of rejected heat in the precoolers. Furthermore, the complex coupling relations between different subcycles and components were hardly elaborated.

The present work is an effort to tackle these challenges. For this purpose, a novel system that has the characteristics of high integration and completely recuperative Brayton cycle in combination with an absorption cycle capable of power/cooling supply is proposed and analyzed. To accurately evaluate and promote the system performance, thermoeconomic analysis in a synthesis of exergy and economics perspectives is conducted to perform a high-efficient and low-cost design and operation. In addition, a parametric sensitivity analysis concerning a set of decision variables such as maximum temperature, maximum pressure, concentration, and the temperature difference is performed to identify the influence on thermal efficiency, exergy efficiency, and unit cost of products. Moreover, in various factors affecting the system performance, for which different variation trends are indicated, multiple key factors which influence the entire system performance should be evaluated simultaneously. For multiparameter optimization purposes, various optimization procedures are available; among them are the direct search method, Monte Carlo
method, and Genetic Algorithm (GA) method [31]. Considering the strong nonlinear constraints of the proposed system, the direct search method is applied in this paper from the view of the thermoeconomic approach. It is hoped that the results can provide some help in the development of attractive and efficient energy utilization systems.

2. System Description and Assumptions

A schematic diagram of the considered power/cooling integrated system is shown in Figure 1. And the cycle functions as a combination of power and cooling outputs as the main and side products, respectively.

As depicted in the proposed layout, for the SCO₂ Brayton subcycle represented by a solid yellow line, SCO₂ stream 1 which is slightly above the critical point entered the MCP and is compressed into a high pressure state of about 25 MPa; then, the stream leaving the MCP flowed to the LTR and recovered heat; then, the exiting stream 3a joins stream 3b at the recompression compressor (RCP) outlet and merges into a whole stream 3; afterwards, stream 3 is heated in the high-temperature recuperator (HTR) before entering the intermediate heat exchanger (IHE), and then, the stream absorbs heat from the high-temperature source (HTS) through IHE; the outlet stream flows to the TUR1, fully expands, and produces electricity in the power generation unit 1 (PGU1). The expanded stream 6 enters the HTR and LTR to heat stream 3 and stream 2, respectively. Stream 8 exiting the LTR is split into two streams: stream 8a and stream 8b. Stream 8a enters the waste heat recovery exchanger 1 (WHE1), the GEN, and the waste heat recovery exchanger 2 (WHE2) to heat the ammonia-water solution streams, respectively. And stream 8b flows into the RCP and further joins stream 3a exiting the LTR. For the absorption power/cooling subcycle represented by a solid blue line, the saturated high-concentration ammonia-water solution (referred to as rich solution) stream 31 leaving the absorber 1 (ABS1) is throttled, and afterwards enters the condenser (CON); then, the saturated liquid ammonia flows into the subcooler (SUC) to heat the ammonia vapor leaving the SUC, is throttled, and afterwards enters the evaporator (EVA) to produce refrigeration effect; the ammonia vapor leaving the SUC joins with the pressurized basic solution stream 17 and is remixed into a rich solution, and therefore, the whole system completed.

To simplify the analysis, some reasonable assumptions are made in this paper:

(1) The steady-state operating conditions are assumed, and the possible leakage and pressure drops in the connection pipes are neglected [24, 32]

(2) Heat dissipation to the surroundings in the main devices and pipes is neglected [24, 31]

(3) The stream states at the CON, ABS1, and ABS2 exit are assumed to be saturated, and the cooling water enters each device above ambient temperature and pressure [24]

(4) Specific isentropic efficiencies are assumed in the pumps and turbines [32]

(5) The chilled water entering the EVA is assumed to be water under environmental conditions

3. Exergy and Exergoeconomic Analysis Model and Performance Evaluation Criteria of the System

3.1. Exergy Analysis Model and Performance Evaluation Criterion of the Novel System. The proposed system consists of various functional devices, turbines, compressors, and heat exchangers, for instance. And each component of the proposed system can be considered to be a controllable volume with inlet and outlet streams, energy transfer, and work interactions. Mass and exergy balance expressions based on the principle of the first and second law of each component at corresponding state points are provided for exergy analysis.

The total exergy rate of input heat and inlet exergy streams equals the total exergy rate of output work, outlet exergy streams, and irreversible exergy loss rate. Therefore, the exergy balance equation of an open, steady component with multiple inflow and outflow streams is summarized as follows [24]:

\[ E_{w,i} = E_{q,i} + \sum E_{in,i} - \sum E_{out,i} - E_{d,i}, \]  

where \( E_{w,i} \) is the exergy rate associated with output work, \( E_{q,i} \) is the exergy rate associated with input thermal energy, \( E_{in,i} \) and \( E_{out,i} \) are the exergy rates associated with inlet and outlet flows, respectively, and \( E_{d,i} \) is the relevant exergy destruction rate.

For the inflow and outflow streams of each component, ignore the impact of magnetic, nuclear, electrical, and surface effects and neglect the changes in mechanical energy;
the exergy value of a certain stream is the sum of physical and chemical exergies. And exergy rate expression is as follows:

\[ \dot{E} = \dot{E}_{ph} + \dot{E}_{ch}. \]  

(2)

And the physical exergy can be expressed as [26]

\[ \dot{E}_{ph} = \dot{m}(h - h_{ref}) - T_{ref}(s - s_{ref}), \]  

(3)

where the subscript ref stands for the reference state (surroundings) of CO\(_2\) at 298.15 K and 0.1 MPa. And the chemical exergy of the ammonia-water mixture at different concentrations can be expressed as [26]

\[ \dot{E}_{ch} = \dot{m} \left[ \left( \frac{X}{M_{NH_3}} \right) e_{ch,NH_3}^0 + \left( 1 - \frac{X}{M_{H_2O}} \right) e_{ch,H_2O}^0 \right], \]  

(4)

where \(X\) is the molar fraction of ammonia in the ammonia-water mixture and the \(M_{NH_3}\) and \(M_{H_2O}\) are the molar mass of ammonia and water, respectively. \(e_{ch,NH_3}^0\) and \(e_{ch,H_2O}^0\) are the standard reference chemical exergy values of ammonia and water, respectively [33].

To evaluate the proposed system from the first law perspective, the overall thermal efficiency of the combined system \(\eta_{en}\) can be expressed as [34]

\[ \eta_{en} = \frac{W_{\text{net,brayton}} + W_{\text{net,absorption}} + Q_{\text{refrigeration}}}{Q_{\text{input}}}, \]  

(5)

where \(W_{\text{net,brayton}}\), \(W_{\text{net,absorption}}\), and \(Q_{\text{refrigeration}}\) are, respectively, the net power output for the Brayton subcycle, the net power output for the absorption power/cooling subcycle, and the cooling capacity associated with the refrigeration output in EVA. \(Q_{\text{input}}\) is the supplied heat associated with the heat source in HTS.

To evaluate the proposed system from the second law perspective, the exergy efficiency of the combined system \(\eta_{ex}\) can be defined as [34]

\[ \eta_{ex} = \frac{W_{\text{net,brayton}} + W_{\text{net,absorption}} + \dot{E}_{\text{refrigeration}}}{\dot{E}_{\text{input}}}, \]  

(6)
where \( E_{\text{refrigeration}} \) is the exergy associated with the refrigeration effect, \( E_{\text{input}} \) is the exergy associated with the heat source in HTS. These are expressed as

\[
E_{\text{refrigeration}} = E_{25}^{t} - E_{26}^{t} = m_{25}(h_{25} - h_{26}) - T_{\text{ref}}(s_{25} - s_{26}),
\]

\[
E_{\text{input}} = Q_{\text{input}} \left( 1 - \frac{T_{\text{ref}}}{T_{\text{HTS}}} \right).
\]  

(7)

Instead of the conventional temperate effectiveness definition based on the constant isobaric specific heat capacity assumption in classical heat exchanger design guides, the physical property variation of \( \text{SCO}_2 \) in HTR and LTR is too dramatic to ignore; therefore, the definition of enthalpy effectiveness is adopted and expressed as [35]

\[
\epsilon_{\text{HTR}} = \frac{m_{6}(h_{6} - h_{7})}{m_{6}(h_{6} - h_{7_{\text{min}}}(P_{7}, T_{3}))} - \frac{m_{7}(h_{7} - h_{8})}{m_{7}(h_{7} - h_{8_{\text{min}}}(P_{7}, T_{3}))},
\]

\[
\epsilon_{\text{LTR}} = \frac{m_{8}(h_{8} - h_{9})}{m_{8}(h_{8} - h_{9_{\text{min}}}(P_{7}, T_{3}))} - \frac{m_{9}(h_{9} - h_{10})}{m_{9}(h_{9} - h_{10_{\text{min}}}(P_{7}, T_{3}))}.
\]  

(8)

3.2. Exergoeconomic Analysis Model and Performance Evaluation Criterion of the Novel System.

For system equipment that absorbs heat and generates electric, the total cost rate of output work and outlet exergy streams, and equipment expenditure is equal to the overall cost rate of input heat exergy, inlet exergy streams, and equipment expenditure is expressed as [35]

\[
C_{\text{w,total}} = \frac{\sum_{i=1}^{n_{f}} \gamma_{i} Z_{i} + \sum_{i=1}^{n_{p}} Z_{i}}{Z_{i}}.
\]  

(15)

where \( Z_{i} \) and \( Z_{i}^{\text{OM}} \) are the cost rate terms related to annu-

\[
Z_{i}^{\text{CL}} = \frac{\text{CRF}}{\tau} Z_{i},
\]

(12)

where CRF is the capital recovery factor and \( \tau \) is the annual system operation time 8000 h [36]. And the relevant expressions are expressed as follows:

\[
\text{CRF} = \frac{i_{r}(1 + i_{r})^{n}}{(1 + i_{r})^{n} - 1}.
\]  

(13)

And \( i_{r} \) is the annualized interest rate (10%) and \( n \) is the system effective operational period (20 years).

\[
Z_{i}^{\text{OM}} = \gamma_{i} Z_{i} + \omega_{i} E_{p,i} + R_{i},
\]  

(14)

where \( \gamma_{i} \) and \( \omega_{i} \) are the fixed and alterable operation and maintenance expenditure and \( R_{i} \) is the unanticipated operation and maintenance costs in addition to equipment investment and product exergy. In the last two terms in equation (14) which are profoundly smaller ones, they are frequently neglected in published works of literature, and the same strategy is adopted in the present work; consequently, the \( \gamma_{i} \) is fixed at 0.06 [37].

To evaluate the proposed system from the exergy economics point of view, the specific cost of the product exergy streams for the whole system \( c_{p,\text{total}} \) is expressed as

\[
c_{p,\text{total}} = \frac{\sum_{i=1}^{n_{f}} \gamma_{i} E_{p,i} + \sum_{i=1}^{n_{p}} Z_{i}}{\sum_{i=1}^{n_{p}} E_{p,i}},
\]

where \( n_{f}, n_{p}, \) and \( n_{p} \) are the entire terms related to the fuel exergy streams, system components, and product exergy streams for the whole system, respectively.

The detailed exergoeconomic balance equations and relevant auxiliary equations for each component in the proposed system are expressed in Table 6 in the appendix, and the relevant calculation models of \( Z_{i} \) are provided in Table 7 in the appendix. The components downstream of the throttle values and expansion value are regarded as an integrated component for that the relevant definition of throttling products is unclear. And the costs of the valves and SPL are ignored due to the relatively smaller proportion of total investment. It is worth pointing out that the heat exchangers most commonly adopted for the Brayton subcycle are printed circuit heat exchangers (PCHE) [10, 38]. Compared to typical bulky shell and tube exchangers, PCHE are novel compact microchannel plate exchangers which are assumed to have excellent pressure and temperature resistance but relatively much higher prices. To calculate the required equipment investment accurately, a material of SUS316L and the cost price of 50 $/kg of the bonded block core are assumed, and the typical sizes for etched channel diameter, channel spacing, and plate thickness of 1.5 mm,
0.5 mm, and 1.5 mm are adopted in the cost calculation, and the detailed schematic model of the cross section and geometry dimension feature of the PCHE is illustrated in Figure 14 in the appendix. Moreover, the definition of integral mean temperature difference (IMTD) and calculation of heat transfer area is illustrated in Figure 15 in the appendix.

Assume the specific exergetic cost of the entering cooling water stream of CON, ABS1, ABS2, and EVA to be 0 $/GJ [39]. Therefore, exergetic cost equations of relevant fuel and product exergy streams can be obtained, and the linear systems of equations listed in Table 6 in the appendix consist of 51 variables \( C_1, C_2, C_3, C_3a, C_3b, C_4, \ldots, C_8, C_8a, C_8b, \ldots, C_{47} \) and 51 equations and are solved using the open-source library package Scipy 1.7 under Python environment.

On the basis of the above mathematical analysis model, a simulation procedure is built and implemented on the Python 3.7 platform, and the fluid properties are calculated using the open-source library package CoolProp.

3.3. Model Validation. For model validation purposes, since there hardly exists a thermal system with consistent layout and parameter operation ranges in published pieces of literature, the \( \text{SCO}_2 \) Brayton subcycle and the absorption power/cooling subcycle within the proposed system are verified individually. And the detailed comparison results are listed in Tables 1 and 2.

To validate the established model built for the proposed system, the performance results of the \( \text{SCO}_2 \) recompression Brayton subcycle are compared with the available published literature data by Cheng et al. [40]. And the satisfying agreement is reached for the comparison results illustrated in Table 1.

For the absorption power/cooling subcycle, a published literature data by Zheng et al. [41] with the similar operation and functional characteristic is built, as the mass flow rate information for each state point is not provided and the mass flow rate of the working medium at the turbine inlet is set as 1 kg/s for the sake of contrastive analysis. As can be seen from Table 2, the thermal and exergy efficiency of the reference system is 24.24% and 37.28%, respectively; the corresponding energy and exergy efficiency of the built procedure is 24.0% and 38.7%, respectively. The main reasons for the deviations are the differences brought by the thermal physical property programs and rounding error.

In general, the comparison results in Tables 1 and 2 agree well with those reported in published literature; therefore, the accuracy of the built procedure is valid.

### 4. Results and Discussion

#### 4.1. Parametric Sensitivity Analysis

A parametric sensitivity analysis is carried out for the proposed combined power/cooling cycle to reveal the impacts on thermal efficiency, exergy efficiency, and the specific cost of the product exergy streams for the whole system. The key parameters concerned include TUR1 inlet temperature, TUR1 inlet pressure, WHE1 outlet degree of overheat, WHE1 pitch point difference, TUR2 inlet pressure, TUR2 inlet ammonia concentration, GEN outlet temperature, and ABS1 outlet ammonia concentration. And the basic initial input parameters for the base case system performance evaluation are summarized in Table 3.

The mass flow rate of the \( \text{SCO}_2 \) Brayton subcycle \( m_5 \) is assumed to be 2500 kg/s, and the molar fraction of ammonia vapor at the top exit of the RET \( x_{22} \) is 0.998, which can be treated as pure ammonia. And the detailed information for the thermodynamic properties of each state point is summarized in Table 8 in the appendix.

The effects of TUR1 inlet temperature \( T_5 \) on thermal efficiency, exergy efficiency, and the specific cost of the product exergy streams for the proposed system (represented by solid symbols) and typically \( \text{SCO}_2 \) recompression Brayton cycle (represented by hollow symbols) are depicted in Figure 2. With the increase in the \( T_5 \) for the Brayton power module, the generated electricity of the TUR1 is on the increase, the outflow enthalpy of the HTR is on the increase, and the enthalpy values of \( H_8, H_{8a}, \) and \( H_2 \) remain constant, due to that split ratio \( = 1 - (h_2 - h_8)/(h_{8a} - h_2) \); the split ratio is on the decrease; therefore, the power consumption of the MCP is on the increase, while that of the RCP is on the decrease, and the total consumption of the two compressors is on the decrease; the thermal efficiency, the exergy efficiency, and the net output capacity are on the increase. Moreover, compared to the increase rate of the cost of fuel exergy streams and the system components, the exergetic cost rate of the products increases significantly at relatively low heat source temperatures; therefore, the specific cost of product exergy streams declines sharply, and further benefits of increasing \( T_5 \) are negligible for inlet temperature above 800 K. For the thermal efficiency, the net power output, the refrigeration output, and the supplied heat increase with \( T_5 \); the relative increase in overall output capacity is greater than the thermal energy supply; the maximum thermal efficiency can reach up to 71.35% at the \( T_5 \) of 923.15 K. For the exergy efficiency, take the inlet temperatures of 646, 670, and 692 K for example, the overall exergy output is 67.00, 82.66, and 98.12 MW, the thermal exergy input is 162.67, 178.57, and 194.70 MW, and the relative increase in input exergy is 15.67, 15.90, and 16.13 MW, while the relative increase in output exergy is 15.92, 15.67, and 15.46 MW, respectively; therefore, the promotion effect of inlet temperature on
exergy efficiency is not so obvious under higher TUR1 inlet temperature; the maximum thermal efficiency can reach up to 67.31% at the $T_T$ of 923.15 K. For the specific cost of system products, although the cost of fuel exergy streams, system components, and the product exergy streams rise with $T_T$, the increase rate for the former two items is much smaller compared to that for the last item; hence, the variation for unit cost of system products is in a downward trend.

Figure 3 shows the effects of TUR1 inlet pressure $P_5$ on thermal efficiency, exergy efficiency, and the specific cost of the product exergy streams for the proposed system (represented by solid symbols) and typically SC$	ext{O}_2$ recompression Brayton cycle (represented by hollow symbols). As can be seen from Figure 3, compared to the recompression Brayton cycle, the thermal efficiency, the exergy efficiency, and the specific cost of the product exergy streams are enhanced significantly, and the influences of $P_5$ on system performance display a similar tendency to that of TUR1 inlet temperature $T_T$, and the reasons behind are almost the same with the variation of TUR1 inlet temperatures.

For the recompression Brayton cycle, with the inlet pressure $P_5$ rises, the split ratio decreases from 0.56 to 0.40, and the mass flow rate which flows to the WHE1 and WHE2 is on the increase; the generated power in the TUR1 increases from 156 MW to 300.9 MW, while the power consumption in the MCP and the RCP increases from 24.2 MW to 77.2 MW and 81.5 MW to 126.38 MW, respectively. And the benefits of increasing $P_5$ are negligible for inlet pressure above 26 MPa. The combined system exhibits optimal unit product cost performance of 11.58 $/\text{GJ}$ under the inlet pressure of 29 MPa, and the maximum thermal efficiency and exergy efficiency reach 69.20% and 65.00%, respectively.

The variation of thermal efficiency, exergy efficiency, and the specific cost of the product exergy streams with TUR2 inlet pressure $P_{11}$ is shown in Figure 4. As the TUR2 inlet pressure $P_{11}$ rises from 2.5 MPa to 4.0 MPa, the thermal efficiency is linearly proportional to the TUR2 inlet pressure, and the overall trend of exergy efficiency is increasing; however, the increase rate gradually decreases. The split ratio of the combined system has a significant influence on the system performance; for the thermal efficiency, the increase in the split ratio results in a decrease in power consumption for the MCP and a sharp increase in the power consumption for the RCP; therefore, the net power output of the Brayton subcycle and the refrigeration output capacity are on the decrease, the net power output of the absorption power/cooling subcycle is on the increase, the overall output energy and the input heat capacity are on the decrease, and the combined effect is a proportional increase; the maximum thermal efficiency reaches 69.09%. For the exergy efficiency, the decrease rate of overall output exergy exceeds the corresponding decrease rate of input exergy, and the maximum exergy efficiency reaches 63.23%.

For the specific cost of system products, within the TUR2 inlet pressure range investigated, both the fuel cost streams, the specific cost rate of system components, and the specific cost rate of product exergy streams are on the decrease, and firstly, the decrease rate of the product exergy streams is higher than that of the former two items; therefore, the combined effect is a proportional increase; afterwards, the decrease rate of the fuel cost streams and the system components is equal to that of product exergy stream

<table>
<thead>
<tr>
<th>Cycle parameters</th>
<th>Energy</th>
<th>Exergy</th>
<th>Energy</th>
<th>Exergy</th>
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<tbody>
<tr>
<td>Input heat $Q_{\text{in}}$ (kW)</td>
<td>3730</td>
<td>1946</td>
<td>2719.5</td>
<td>1185.8</td>
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<tr>
<td>Input heat $Q_{\text{gen}}$ (kW)</td>
<td>667.3</td>
<td>204.5</td>
<td>338.9</td>
<td>118</td>
</tr>
<tr>
<td>Turbine generated work $W_{\text{tur}}$ (kW)</td>
<td>774.7</td>
<td>774.7</td>
<td>474.8</td>
<td>474.8</td>
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<tr>
<td>Refrigeration capacity $Q_{\text{ref}}$ (kW)</td>
<td>298.8</td>
<td>39.42</td>
<td>259.5</td>
<td>30.5</td>
</tr>
<tr>
<td>Pump input work $P_1$ (kW)</td>
<td>23.14</td>
<td>23.14</td>
<td>16.21</td>
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<tr>
<td>Pump input work $P_2$ (kW)</td>
<td>9.275</td>
<td>9.275</td>
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<td>0</td>
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<tr>
<td>Pump input work $P_3$ (kW)</td>
<td>0</td>
<td>0</td>
<td>4.32</td>
<td>4.32</td>
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<tr>
<td>Overall efficiency (%)</td>
<td>24.24</td>
<td>37.28</td>
<td>24.0</td>
<td>38.7</td>
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Table 2: Cycle parameter and efficiency value comparison for two absorption systems.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
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<tr>
<td>Ambient temperature (K)</td>
<td>298.15</td>
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<td>Ambient pressure (kPa)</td>
<td>100</td>
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<tr>
<td>TUR2 inlet ammonia vapor concentration</td>
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<tr>
<td>TUR1 inlet temperature (K)</td>
<td>823.15</td>
</tr>
<tr>
<td>TUR1 inlet pressure (kPa)</td>
<td>25000</td>
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<tr>
<td>MCP inlet temperature (K)</td>
<td>308.15</td>
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<tr>
<td>MCP inlet pressure (kPa)</td>
<td>7400</td>
</tr>
<tr>
<td>GEN inlet temperature (K)</td>
<td>403.15</td>
</tr>
<tr>
<td>Minimum allowable dryness in the turbine exhaust</td>
<td>0.9</td>
</tr>
<tr>
<td>Compressor isentropic efficiency</td>
<td>0.85</td>
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<td>Pump isentropic efficiency</td>
<td>0.87</td>
</tr>
<tr>
<td>Turbine isentropic efficiency</td>
<td>0.85</td>
</tr>
<tr>
<td>Refrigeration temperature (K)</td>
<td>280.15</td>
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<td>Pinch point temperature difference (K)</td>
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<tr>
<td>Concentration of the basic solution (%)</td>
<td>0.42</td>
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<tr>
<td>TUR2 inlet degree of superheat (K)</td>
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</tr>
<tr>
<td>Reactor fuel cost ($/\text{MWh}$)</td>
<td>7.4</td>
</tr>
</tbody>
</table>

Table 3: Main assumptions and initial input parameters for the base case system performance.
cost at the TUR2 inlet pressure of 3.5 MPa, where the optimal unit product cost performance of 11.667 $/GJ is obtained.

The variation of thermal efficiency, exergy efficiency, and specific cost of the product exergy streams with the degree of overheat $\Delta T_{11}$ and WHE1 pitch point difference $\Delta T_8$ is illustrated in Figures 5 and 6. The variation curves of system performance for two temperature differences show an opposite tendency, and it is the coupling relationships between two subcycles that cause the appearance. To simplify the tedious analysis content, take the variation of thermal efficiency for WHE1 pitch point difference $\Delta T_8$ for example. For the thermal efficiency, the split ratio is on the
increase, the net power output of the Brayton subcycle is on the decrease, the mass flow rate of the absorption power/cooling subcycle rises from 189.3 to 191.2 kg/s, and the relevant energy output is on the increase; however, the input heat capacity decrease rate of 0.56% is lesser than that of 0.83% for the overall decrease rate of the proposed system; therefore, the thermal efficiency is inversely proportional to $\Delta T_8$. For the specific cost of system products, the decrease rate of fuel cost streams and the system components drops from 0.57% to 0.40%, while the decrease rate of product exergy stream cost rises from 1.48% to 1.74%; the unit cost of system products rises from 11.52 to 13.63 $/GJ$ consequently. The maximum thermal efficiency and exergy efficiency reach 69.0% and 64.0%, respectively.

Figure 4 shows the variation of thermal efficiency, exergy efficiency, and specific cost of the product exergy streams with the system TUR2 inlet pressure $P_{11}$.

Figure 5: Variations of thermal efficiency, exergy efficiency, and specific cost of the product exergy streams with the system WHE1 outlet degree of overheat $\Delta T_{11}$. 

![Graph](image-url)
with TUR2 inlet ammonia concentration $X_{11}$. As the $X_{11}$ rises, the split ratio drops from 0.426 to 0.286, and the net power output of the Brayton subcycle increases from 136.4 to 175.5 MW, which has a dominant influence on the increase in overall energy output, and the increase rate is 4 times that of the supplied heat in the IHE; therefore, thermal efficiency and exergy efficiency increase from 67.1% to 84.1% and from 62.5% to 70.7%, respectively. For the specific cost of system products, the fuel cost streams and the system components rise from 4552.5 to 4882.1 $/h and product exergy stream cost rises from 648.7 to 788.7 $/h, but the minimum increase rate of the latter is 2 times larger than the former one; the unit cost of system products thereby decreases gradually.

Figure 8 presents the effects of GEN outlet temperature $T_{10}$ on thermal efficiency, exergy efficiency, and the specific cost of the product exergy streams. As the GEN outlet temperature $T_{10}$ rises, the net power output of the Brayton
subcycle, supplied heat, exergy, and exergy cost in the IHE remain unchanged, and energy output and exergy of the absorption power/cooling subcycle are on the increase; hence, the thermal efficiency and exergy efficiency are improving, and the maximum thermal efficiency and exergy efficiency reach 69.5% and 63.4%, respectively. For the specific cost of system products, the exergy cost of the system components is on the decrease and the product exergy streams cost is on the increase, and the unit cost of products drops consequently. And the minimum specific cost of system products reaches 11.61 $/GJ.

The variation of thermal efficiency, exergy efficiency, and the specific cost of the product exergy streams with ABS1 outlet ammonia concentration $X_{14}$ (molar fraction %) is shown in Figure 9. As the ABS1 outlet ammonia concentration rises, the net power output of the Brayton subcycle, supplied heat, exergy, and exergy cost in the IHE remain unchanged, and energy output and exergy of the absorption power/cooling subcycle are on the decrease; hence, the thermal efficiency and exergy efficiency are declining. For the specific cost of system products, the exergy cost of the system components and the product exergy streams cost is on the decrease, and the
minimum increase rate of the latter is 1.8 times larger than the former one; hence, the unit cost of products declines consequently.

Since certain parameter variations have opposing effects on system performance, it is necessary to figure out the reasons behind them. Figure 10 presents the effects of TUR1 inlet pressure \( P_5 \) and TUR2 inlet pressure \( P_{11} \) on thermal efficiency, exergy efficiency, and the specific cost of the product exergy streams. The split ratio of the combined system rises with the TUR2 inlet pressure \( P_{11} \) and drops with the TUR1 inlet pressure \( P_5 \), and the related influence of TUR1 inlet pressure \( P_5 \) is weaker than that of TUR2 inlet pressure \( P_{11} \). Figure 10(a) shows that the cycle will exhibit a maximum thermal performance of 69.5% under the specified condition when \( P_5 \) and \( P_{11} \) reach 30 and 4 MPa, respectively. For the exergy efficiency, TUR1 inlet pressure \( P_5 \) has a stronger influence on the exergy efficiency; therefore, the system can achieve higher overall exergy efficiency under lower \( P_5 \) and higher \( P_{11} \), which favors the feasibility of implementation. For the specific cost of system products, as illustrated in Figures 3 and 4, TUR1 inlet pressure \( P_5 \) and TUR2 inlet pressure \( P_{11} \) have extremely inverse impacts on the unit cost of products, and it is apparent that there exists an optimal unit product cost under the TUR1 inlet pressure \( P_5 \) of 29 MPa, and the minimum unit product cost reaches 11.57 $/GJ.

The variation of thermal efficiency, exergy efficiency, and the specific cost of the product exergy streams with TUR1 inlet temperature \( T_5 \) and WHE1 pitch point difference \( \Delta T_8 \) is depicted in Figure 11. For thermal efficiency, exergy efficiency, and the specific cost of system products, as illustrated in Figures 2 and 6, \( T_5 \) and \( \Delta T_8 \) have extremely opposite impacts. The split ratio and mass flow rate of the absorption power/cooling subcycle are two factors influencing the coupling relationships and overall system performance, the split ratio of the combined system rises with the WHE1 pitch point difference \( \Delta T_8 \) and drops with the TUR1 inlet temperature \( T_5 \), and the absorption power/cooling subcycle mass flow rate rises with \( T_5 \) and \( \Delta T_8 \). However, the related influence of \( \Delta T_8 \) is weaker than that of \( T_5 \). Figure 11(a) shows that the cycle exhibits a maximum thermal performance of 71.6% under the specified condition when \( T_5 \) and \( \Delta T_8 \) reach 923.15 K and 16 K; moreover, the

![Figure 10: Variations of system performance with the system TUR1 inlet pressure and TUR2 inlet pressure.](image)
thermal efficiency and exergy efficiency gains under higher TUR1 inlet temperatures are extremely limited. For the specific cost of system products, it is evident that WHE1 pitch point difference $\Delta T_8$ is the dominant affecting factor under lower $T_5$, and the minimum unit product cost reaches 10.02 $/GJ$.

The effects of TUR1 inlet temperature $T_5$ and GEN outlet temperature $T_{10}$ on thermal efficiency, exergy efficiency, the specific cost of the product exergy streams and absorption power/cooling subcycle mass flow rate are illustrated in Figure 12. As shown in Figure 12(d), the mass flow rate of the absorption power/cooling subcycle rises with GEN outlet temperature $T_{10}$ and the TUR1 inlet temperature $T_5$. For the thermal efficiency, however, the related influence of the GEN outlet temperature is weaker than that of the TUR1 inlet temperature. Figure 12(a) shows that the cycle exhibits a maximum thermal performance of 72% under the specified condition when $T_5$ and $T_{10}$ reach 923.15 K and 418.15 K; moreover, GEN outlet temperature $T_{10}$ has a negligible effect on the exergy efficiency, and relevant gains under higher TUR1 inlet temperatures are limited as well. For the specific cost of system products, it is evident that TUR1 inlet temperature $T_5$ is the dominant affecting factor, and the minimum unit product cost reaches 10.07 $/GJ$.

4.2. System Performance Optimization. For exergoeconomic evaluation of the combined system, the exergy destruction rate $\dot{E}_{d,i}$ and the cost rate of exergy destruction rate $\dot{C}_{d,i}$ are chosen as assessment criteria; these parameters are calculated for each component of the proposed system, and the detailed expression is expressed as follows:

$$\dot{C}_{d,i} = c_{f,i} \cdot \dot{E}_{d,i},$$

where $c_{f,i}$ is the unit cost of fuel exergy supplied for the specific component.

The objective function is to maximize $\eta_{en}$ and $\eta_{ex}$ or minimize $c_{P,\text{total}}$:
\[623.15 \leq T_5 \leq 923.15,\]
\[15 \leq P_5 \leq 30,\]
\[2.5 \leq P_{11} \leq 4,\]
\[0 \leq \Delta T_{11} \leq 20,\]
\[15 \leq \Delta T_8 \leq 60,\]
\[0.775 \leq X_{11} \leq 0.925,\]
\[383.15 \leq T_{10} \leq 418.15,\]
\[0.37 \leq X_{14} \leq 0.43.\]  

And the optimal parameter selection of the decision variables and system performance for the thermal efficiency case, the exergy efficiency case, the unit cost of product case, and the base case are illustrated in Table 4. Compared to the base case, three optimal cases exhibit remarkably improvement in thermal efficiency, exergy efficiency, and refrigeration capacity, and the exergy efficiency case is a promising candidate for moderate maximum operating temperature and pressure. Moreover, relevant thermal efficiency, exergy efficiency, and the unit cost of products of the exergy efficiency case are 82.44%, 72.35%, and 5.805 $/GJ, respectively, which are 13.7% and 9.17% higher for the first two items and 83.66% lower for the last item than that of the base case.

Figure 13 presents the contribution proportion of each component in the relevant cost rate $Z_i$, the exergy destruction rate $E_{de,i}$, and the cost rate of exergy destruction $C_{de,i}$ for the base case, $\eta_{en}$ case, $\eta_{ex}$ case, and $c_{P,total}$ case, respectively. TUR1 and MCP account for the first- and second-largest cost rate for each case, RET and GEN make up roughly 34% exergy destruction rate and 20% cost rate of exergy destruction rate in three optimal cases, and reducing heat transfer difference in relevant equipment can further promote the system performance.
For multiobjective optimal parameter selection purposes, considering that contradictory goals exist in the optimization process and the ambiguous feature of the Pareto-GA method, a more straightforward approach to weighting each objective function is used [36]. And the integrated objective function is expressed as follows:

\[
\text{The objective function is to maximize } (w_1 \times \eta_{en} + w_2 \times \eta_{ex} + w_3 \times (1 - c_{p,\text{total}}/c_{\text{reactor}})):
\]

\[
0 \leq w_1, w_2, w_3 \leq 1,
\]

\[
w_1 + w_2 + w_3 = 1.
\]

And the relevant weighting factors \(w_1\), \(w_2\), and \(w_3\) on the synthetic performance of the multiobjective functions are listed in Table 5. Of the three multiobjective optimal cases, case 1 is based on the average weight of \(\eta_{en}\), \(\eta_{ex}\), and \(c_{p,\text{total}}\), while the remaining cases add weights to the specific cost of system products; nevertheless, results show that the decision variable selection of the three multiobjective optimal cases is approaching, and the major deviations lie in the TUR1 inlet pressure \(P_5\) and WHE1 pitch point difference \(\Delta T_8\). Therefore, the ultimate system scheme and optimal parameter values can be confirmed with flexibility in the optional deviation variables.
5. Conclusions

In this paper, a completely recuperative \( \text{SCO}_2 \) Brayton/absorption integrated power/cooling system is proposed in which the low-grade waste heat extracted from the PRC of the Brayton subcycle is completely recovered to generate additional electrical and cold energy. Based on the decision parameters, the performances of the integrated system are evaluated and optimized through the exergoeconomic approach. Furthermore, the main conclusions are summarized as follows:

1. Thermal efficiency and exergy efficiency are positively related to TUR1 inlet temperature, pressure, TUR2 inlet pressure, TUR2 inlet ammonia concentration, and GEN outlet temperature. The unit cost of products is positively correlated with WHE1 outlet degree of overheat, WHE1 outlet degree of overheat, and ABS1 outlet ammonia concentration, and within the key parameter variation scopes investigated, there exists an optimal TUR2 inlet pressure of 3.5 MPa.

2. The TUR2 inlet temperature, the WHE1 outflow overheat degree, and the WHE1 hot end temperature difference have significant influences on the split ratio; however, unreasonable high effectiveness in relevant equipment has adverse effects on net output capacity decreases and the unit cost of products.

3. Compared to the base case, three optimal cases exhibit remarkable improvement in thermal efficiency, exergy efficiency, and refrigeration capacity. And the exergy efficiency case is a promising candidate for \( \eta_{\text{en}}, \eta_{\text{ex}}, \) and \( c_{P,\text{total}} \) of 82.44%, 72.35%, and 9.489 $/GJ$, respectively.

4. Exergy destruction and cost rate distribution of each component for three optimal cases reveal that TUR1 and MCP account for the first- and second-largest cost rate, while the RET and GEN make up roughly 34% of exergy destruction rate, and reducing heat transfer difference in relevant equipment can further promote the system performance.

5. In conclusion, integrating with the absorption power/cooling subcycle instead can significantly improve the system performance and environmental condition adaptability, especially at efficient utilization of low- and medium-temperature renewable energy sources. Therefore, the ultimate system scheme and optimal parameter values can be confirmed with flexibility in the optional deviation variables.

Appendix

The detailed exergoeconomic balance equations and relevant auxiliary equations for each component in the proposed system are expressed in Table 6. The capital investment models are listed in Table 7 for each component of the proposed system [27]. The costs of throttling values VAL1 and VAL2 and splitter SPL are ignored for that the required expenditures are smaller compared to the remaining components, and the detailed information for the thermodynamics properties of each state point is summarized in Table 8.

For the capital cost of PCHE for HTR, LTR, and IHE, the required heat transfer area is calculated in an integral mean temperature difference method. Taking into account the impact of property variation, the relevant heat transfer areas are better evaluated by integral mean temperature difference (IMTD) instead of logarithmic mean temperature difference.
<table>
<thead>
<tr>
<th>Components</th>
<th>Exergoeconomic balance equations</th>
<th>Auxiliary equations</th>
</tr>
</thead>
<tbody>
<tr>
<td>MCP</td>
<td>( \dot{C}<em>2 = \dot{C}<em>1 + C</em>{36} + Z</em>\text{MCP} )</td>
<td>( \frac{C_{34}}{E_{34}} = \frac{C_{36}}{E_{36}} )</td>
</tr>
<tr>
<td>RCP</td>
<td>( C_{3b} = C_{3b} + C_{35} + Z_\text{RCP} )</td>
<td>( \frac{C_{34}}{E_{34}} = \frac{C_{35}}{E_{35}} )</td>
</tr>
<tr>
<td>TUR1</td>
<td>( \dot{C}<em>6 + \dot{C}</em>{34} = \dot{C}<em>5 + Z</em>\text{TUR1} )</td>
<td>( \frac{C_6}{E_6} = \frac{C_5}{E_5} )</td>
</tr>
<tr>
<td>IHE+HTS</td>
<td>( \dot{C}_3 = \dot{C}<em>4 + Z</em>\text{HTS} )</td>
<td></td>
</tr>
<tr>
<td>HTR</td>
<td>( \dot{C}_4 + \dot{C}_7 = \dot{C}_6 + \dot{C}<em>3 + Z</em>\text{HTR} )</td>
<td>( \frac{C_d}{E_d} + \frac{C_f}{E_f} + \frac{C_d}{E_d} + \frac{C_f}{E_f} = 0 )</td>
</tr>
<tr>
<td>LTR</td>
<td>( \dot{C}<em>8 + \dot{C}</em>{3a} = \dot{C}_7 + \dot{C}<em>2 + Z</em>\text{LTR} )</td>
<td>( \frac{C_d}{E_d} = \frac{C_f}{E_f} + \frac{C_d}{E_d} = \frac{C_f}{E_f} )</td>
</tr>
<tr>
<td>WHE1</td>
<td>( C_{11} + C_9 = C_{4a} + C_{33} + Z_\text{WHE1} )</td>
<td>( \frac{C_{4b}}{E_{4b}} = \frac{C_{4a}}{E_{4a}} )</td>
</tr>
<tr>
<td>WHE2</td>
<td>( \dot{C}<em>1 + \dot{C}</em>{33} = \dot{C}<em>{10} + \dot{C}</em>{32} + Z_\text{WHE2} )</td>
<td></td>
</tr>
<tr>
<td>P2</td>
<td>( \dot{C}<em>{32} + \dot{C}</em>{31} = \dot{C}<em>{37} + Z</em>\text{P2} )</td>
<td></td>
</tr>
<tr>
<td>ABS2+VAL3</td>
<td>( \dot{C}<em>{31} + \dot{C}</em>{45} = \dot{C}<em>{17} + \dot{C}</em>{27} + \dot{C}<em>{44} + Z</em>\text{ABS2+VAL3} )</td>
<td>( \frac{C_{4a}}{E_{4a}} = \frac{C_{4b}}{E_{4b}} = \frac{C_{4c}}{E_{4c}} = 0 )</td>
</tr>
<tr>
<td>P1</td>
<td>( \dot{C}<em>{15} + \dot{C}</em>{14} + \dot{C}<em>{38} + Z</em>\text{P1} )</td>
<td>( \frac{C_{4b}}{E_{4b}} = \frac{C_{4a}}{E_{4a}} )</td>
</tr>
<tr>
<td>ABS1+VAL1</td>
<td>( \dot{C}<em>{41} + \dot{C}</em>{40} + \dot{C}<em>{49} + \dot{Z}</em>\text{ABS1+VAL1} )</td>
<td></td>
</tr>
<tr>
<td>CON</td>
<td>( C_{23} + C_{43} = C_{42} + C_{22} + Z_\text{CON} )</td>
<td></td>
</tr>
<tr>
<td>SUC</td>
<td>( C_{24} + C_{27} = C_{26} + C_{23} + Z_\text{SUC} )</td>
<td>( \frac{C_{28}}{E_{28}} = \frac{C_{27}}{E_{27}} )</td>
</tr>
<tr>
<td>EVA+VAL2</td>
<td>( \dot{C}<em>{26} + \dot{C}</em>{47} = \dot{C}<em>{24} + \dot{C}</em>{46} + Z_\text{EVA+VAL2} )</td>
<td>( \frac{C_{26}}{E_{26}} = \frac{C_{24}}{E_{24}} )</td>
</tr>
<tr>
<td>SHR1</td>
<td>( \dot{C}<em>{20} + \dot{C}</em>{13} = \dot{C}<em>{12} + \dot{C}</em>{19} + Z_\text{SHR1} )</td>
<td></td>
</tr>
<tr>
<td>SHR2</td>
<td>( \dot{C}<em>{21} + \dot{C}</em>{29} = \dot{C}<em>{20} + \dot{C}</em>{28} + Z_\text{SHR2} )</td>
<td></td>
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<tr>
<td>TUR2</td>
<td>( \dot{C}<em>{12} + \dot{C}</em>{39} = \dot{C}<em>{11} + Z</em>\text{TUR2} )</td>
<td></td>
</tr>
<tr>
<td>RET+GEN</td>
<td>( \frac{C_{22}}{E_{22}} = \frac{C_{10}}{E_{10}} + \frac{C_{28}}{E_{28}} )</td>
<td>( \frac{C_{22}}{E_{22}} = \frac{C_{10}}{E_{10}} + \frac{C_{28}}{E_{28}} )</td>
</tr>
<tr>
<td>SPL</td>
<td>( \dot{C}<em>{15} = \dot{C}</em>{16} )</td>
<td>( \frac{C_{15}}{E_{15}} = \frac{C_{16}}{E_{16}} )</td>
</tr>
</tbody>
</table>
Table 7: The capital investment models for each system component.

<table>
<thead>
<tr>
<th>System component</th>
<th>The capital investment cost function</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reactor</td>
<td>$Z_c = c \cdot Q_{\text{in}}, \quad c = 283$ $$/\text{kWt}$</td>
</tr>
<tr>
<td>Turbine</td>
<td>$Z_t = 479.34 \cdot m_{\text{in}} \cdot 1/0.93 - \eta_t \cdot \ln \left( T_R \right) \cdot \left( 1 + c_0.036 \cdot T_m^{-0.44} \right)$</td>
</tr>
<tr>
<td>Compressor</td>
<td>$Z_c = 71.1 \cdot m_{\text{in}} \cdot \frac{1}{0.92 - \eta_t} \cdot P_{R_c} \cdot \ln \left( P_{R_c} \right)$</td>
</tr>
<tr>
<td>Ammonia heat exchanger</td>
<td>$Z_h = 2143 \cdot A_{h}^{0.514}$</td>
</tr>
<tr>
<td>Pump</td>
<td>$Z_p = 1120 \cdot W_{p}^{0.8}$</td>
</tr>
</tbody>
</table>

Table 8: Thermodynamic properties, exergy streams, and unit cost of system products for the base case system performance.

<table>
<thead>
<tr>
<th>State no.</th>
<th>$T$ (K)</th>
<th>$P$ (kPa)</th>
<th>$m$ (kg/s)</th>
<th>$x$ (%)</th>
<th>$\dot{E}$ (kW)</th>
<th>$\dot{C}$ ($$/h$)</th>
<th>$c$ ($$/GJ$)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>308.15</td>
<td>7400</td>
<td>1454.6</td>
<td>—</td>
<td>294,124.3</td>
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<td>2</td>
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<td>25300</td>
<td>1454.6</td>
<td>—</td>
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<td>0.420</td>
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<td>299,785.3</td>
<td>28.666</td>
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<td>19</td>
<td>308.23</td>
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<td>0.420</td>
<td>2904,905.6</td>
<td>299,785.2</td>
<td>28.666</td>
</tr>
<tr>
<td>20</td>
<td>328.31</td>
<td>1370.01</td>
<td>331.57</td>
<td>0.420</td>
<td>2906,771.8</td>
<td>301,515.0</td>
<td>28.613</td>
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$W_{\text{brayton}}$ (MW) 139.1 $W_{\text{absorption}}$ (MW) 40.8 $W_{\text{refrigeration}}$ (MW) 142.42

$\eta_{\text{en}}$ 68.74% $\eta_{\text{ex}}$ 63.18% $c_{\text{P,total}}$ 11.67 $$/GJ$
As depicted in Figures 14 and 15, for modeling purposes, IMTD can be obtained by dividing the heat transfer length into infinitely small nodes to capture the effect of properties and integration of the local temperature difference.

\[
dQ = U_i \cdot dA_i \cdot \Delta T,
\]

\[
\int_{0}^{Q_0} \frac{dQ}{\Delta T(Q)} = \int_{0}^{A_i} U_i \cdot dA_i,
\]

\[Q = U \cdot A \cdot T D_{IMTD}\]

**Variables**

- **c**: Cost per exergy unit ($/GJ)
- **e**: Specific exergy (kJ/kg)
- **h**: Specific enthalpy (kJ/kg)
- **i_r**: Annualized interest rate
- **m**: Mass flow rate (kg/s)
- **n**: System effective operational period (year)
- **s**: Specific entropy (kJ/(kg·K))
- **w**: Weighting factor
- **C**: Exergy cost rate ($/h)
- **E**: Exergy rate (kW)

**Greek Letters**

- **γ**: Fixed operation and maintenance expenditure
- **ΔT**: Heat transfer temperature difference (K)
- **ε**: Effectiveness
- **η**: Thermal/exergy efficiency
- **τ**: Annual system operation time
- **ω**: Alterable operation and maintenance expenditure.

**Subscripts and Abbreviations**

- **0**: Surroundings
- **1, 2, ..., 51**: State points
- **absorption**: Absorption power/cooling subcycle
- **brayton**: Brayton subcycle
- **ch**: Chemical
**Data Availability**

The data used to support the finding of this study are included within the article. Further data or information is available via corresponding email from the corresponding/first author upon request.

**Conflicts of Interest**

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

**Authors’ Contributions**

Bingchuan Han wrote the original draft. Yongdong Chen was responsible for the conceptualization and supervision. Gaige Yu reviewed the paper. Xiaohong Wu was responsible for the validation. Quishuang Zhang was responsible for the software. Taotao Zhou was responsible for the model validation and syntax check.

**Acknowledgments**

We gratefully acknowledge the financial support for the grants of the Natural Science Foundation of Anhui Province (No. 2008085QE261), Key Research and Development Project of Anhui Province (No. 202104a05020024), and Doctoral Science and Technology Foundation of Hefei General Machinery Research Institute (No. 2020011748).

**References**


