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A new absorption–compression refrigeration system using a mid-temperature heat source for freezing application

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Abstract

The use of an absorption refrigeration system is a promising way to utilize waste heat from industrial processes. Ammonia–water absorption refrigeration system is commonly used for freezing applications with temperatures lower than 0 °C. When the refrigeration temperature is lower than -30 °C, the performance dramatically decreases. We proposed a new absorption–compression refrigeration system to produce cooling energy at -30 °C to -55 °C. The proposed system comprised three subsystems, namely, a power generation subsystem using an ammonia–water mixture as the working fluid, an ammonia–water absorption refrigeration subsystem, and a CO\textsubscript{2} compression refrigeration subsystem. The system utilized the heat source in a cascade manner. The power subsystem converted the high-temperature portion of heat into power to drive the CO\textsubscript{2} compression refrigeration subsystem, thereby resulting in the generation of low-temperature cooling energy. The low-temperature portion of heat is converted into cooling energy to offer the heat sink of the CO\textsubscript{2} compression refrigeration subsystem. A simulation study was conducted, and results showed that the coefficient of performance of the proposed system was 0.277, which was approximately 50% higher than that of a conventional two-stage absorption refrigeration system. This work may provide a new way to produce low-temperature cooling energy using mid-temperature heat source.

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Keywords: Absorption-compression refrigeration; Freezing application; Mid-temperature heat source; Thermodynamic analysis

1. Introduction

The demand for refrigeration at low evaporating temperature is increasing, particularly for rapid freezing, storage of medical materials and high-heat-flux electronics [1, 2]. However, reaching a refrigerating temperature below -30 °C is difficult using a traditional single-stage absorption refrigeration systems [3]. Hence, two-stage or cascade refrigeration systems are developed and often used for low-temperature applications. The high- and low- temperature circuits in a cascade system are filled with different appropriate refrigerants to obtain better performance, compared with a two-stage refrigeration
system. However, both two-stage refrigeration and cascade refrigeration systems show the disadvantage of high electricity consumption.

To reduce electricity consumption while obtaining a low refrigeration temperature, Fernández–Seara et al. [4] studied a cascade refrigeration system with a CO$_2$ compression system and an NH$_3$/H$_2$O absorption system at an evaporation temperature of -45 °C. This system has a coefficient of performance (COP) of 0.253. Garimella and Brown [5] developed a novel cascade absorption–compression system that coupled a single-effect LiBr/H$_2$O absorption cycle and a subcritical CO$_2$ vapor–compression cycle to generate low-temperature refrigerant (-40 °C) for high-heat-flux electronics used in a naval ship.

However, multi-input systems are often unreliable [6]. Therefore, Rogdakis and Antonopoulos [7] studied a NH$_3$/H$_2$O absorption refrigeration system that is merely driven by waste heat. For an ambient temperature of 30 °C, the theoretical COP is in the range of 0.03 to 0.40 when the lowest temperature is in the range of -64 °C to -30 °C. He et al. [8] proposed a novel absorption refrigeration system using R134a and R23 mixed refrigerants and dimethylformamide solvent. The new system used a two-stage absorber in series to reduce the evaporation pressure, and the lowest refrigeration temperature reached -62.3 °C with a COP of 0.023 under a generation temperature of 184.4 °C.

In this study, we propose a new absorption–compression refrigeration system for low-temperatures refrigeration based on the cascade utilization of mid-temperature heat source. The energy efficiency boosting mechanism of the proposed system is elucidated.

2. System description

The detailed configuration of the new absorption–compression refrigeration system is shown in Fig. 1(a). It comprised a Rankine power subsystem using a mixture working fluid, an absorption subsystem in the high-temperature refrigeration stage, and a single-stage compression subsystem in the low-temperature stage. NH$_3$/H$_2$O solution was used as the working fluid in the power and absorption subsystems, whereas CO$_2$ was used in the compression subsystem.

![Fig. 1. (a) Schematic of the new absorption–compression refrigeration system; (b) Schematic of the two-stage NH$_3$/H$_2$O refrigeration system](image-url)
refrigeration subsystem (ARS) and compression refrigeration subsystem (CRS) operated as high-temperature and low-temperature circuits respectively, and were combined to form a cascade refrigeration system. The two circuits were connected to each other through a cascade heat exchanger (CHEX), which acted as an evaporator for the ARS and a condenser for the CRS. The low-temperature portion of the external heat source and the exhaust vapor of the TUR provided heating loads for the absorption subsystem in GHEX and the reboiler (REB), respectively. The TUR provided the power required by the compressor (COMP) in the CRS. External power was not required, but a mid-temperature heat source for the new system was required. Therefore, this system could be used as a stand-alone unit for refrigeration at low temperatures.

A typical two-stage NH$_3$/H$_2$O refrigeration system was selected as the reference system in this study, as shown in Fig. 1(b) [7]. The two-stage system is obtained by adding a low-pressure branch containing one low-pressure evaporator, one low-pressure absorber, two heat exchangers, two throttling devices, and one pump to a conventional single-stage NH$_3$/H$_2$O absorption refrigeration system.

3. System evaluation and simulation

3.1. Evaluation criteria

Typical evaluation criteria for refrigeration systems, such as COP and exergy efficiency $\eta_{ex}$, were used to evaluate the performance of the proposed system.

For this new system, the flue gas was directly discharged to the surroundings after utilization. It is how much cooling capacity could be produced by unit mass of flue gas, rather than COP, that is more concerned and reasonable. Therefore, the cooling capacity per unit mass of flue gas $\omega$ was adopted for the system performance evaluation and was expressed as follows:

$$\omega = \frac{Q_c}{m_f}$$ (1)

where $Q_c$ represents the refrigeration output of the compression refrigeration subsystem and $m_f$ represents the mass flow rate of flue gas.

3.2. General assumptions and parameters specification

In this study, simulations of both the proposed and reference systems were conducted using the commercial software Aspen Plus, which contained various models that could be used for power generation and refrigeration processes. All models were based on mass and energy balance, with a default relative convergence error tolerance of 0.01%. The Predictive Soave-Redlich-Kwong equation of state was used to calculate the thermodynamic properties of the ammonia–water mixture [9]. The Peng–Robinson equation of state was used to calculate the thermodynamic properties of CO$_2$. The STEAM–TA equation of state was used to calculate the thermodynamic properties of H$_2$O.

According to several related studies [10, 11], the parameter specifications are listed in Table 1. The main assumptions for the system simulations were as follows:

<table>
<thead>
<tr>
<th>Items</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump efficiency (%)</td>
<td>75</td>
</tr>
<tr>
<td>Turbine isentropic efficiency (%)</td>
<td>85</td>
</tr>
<tr>
<td>Minimum exhaust dryness of turbine (%)</td>
<td>90</td>
</tr>
<tr>
<td>HRVG hot side temperature difference (°C)</td>
<td>30</td>
</tr>
<tr>
<td>MITA of HRVG (°C)</td>
<td>15</td>
</tr>
<tr>
<td>MITA of general heat exchangers (°C)</td>
<td>5</td>
</tr>
<tr>
<td>Pressure drop from REC to CON (bar)</td>
<td>0.1</td>
</tr>
<tr>
<td>Pressure drop from EVA to ABS (bar)</td>
<td>0.2</td>
</tr>
<tr>
<td>Ammonia mass concentration of the refrigerant (−)</td>
<td>0.998</td>
</tr>
</tbody>
</table>
(a) The cycle was in steady-state.
(b) The general pressure loss is neglected, except for the rectifiers, absorbers, and the throttle valves. The flow crossing the throttle valve process was isenthalpic.
(c) Heat loss was ignored.
(d) The solution at the outlets of REB, ABS (absorber), and CON (condenser) was based on a saturated state.
(e) The isentropic efficiency of CO₂ compressor can be expressed in terms of compression ratio \( R_p \) [2], as follows:

\[
\eta_{s,\text{COMP}} = 0.00476R_p^2 - 0.09238R_p + 0.89810
\]

4. Results and discussion

4.1. System performance

The flue gas temperature was usually approximately 350 °C; thus, a TUR inlet temperature of 320 °C was selected with a proper minimum internal temperature approach, and the inlet pressure was 100 bar. The cooling water temperature was 30 °C. The calculations were based on a unit mass flow rate (1 kg/s) of the flue gas fed into the system.

Table 2 illustrates the thermodynamic performance of the proposed and reference systems. In the proposed system, heat recovered from the 350 °C flue gas was 226.56 kW, and the refrigeration output was 62.70 kW at an evaporation temperature of -55 °C. The COP, reached 0.277. The cooling capacity per unit mass of flue gas \( \omega \) was 62.70 kJ/kg. The simulation results for the reference system (derived under the same assumptions) are shown in Table 3. With unit mass flow rate of the flue gas fed into the system, the COP, was 0.185, and \( \omega \) was 43.01 kJ/kg. The COP, and \( \omega \) of the proposed system increased by 49.73% and 45.78% over those of the reference system, respectively.

Besides an energy analysis, an exergy analysis was performed to reveal the irreversibility in each process and to show the possibilities and methods for system performance improvement. The results of the exergy analysis are presented in Table 3, and these data indicated where exergy destruction and loss occurred. With the same exergy input (109.32 kW), the exergy outputs of the proposed and reference systems were 21.03 and 14.25 kW, respectively. The exergy efficiency of the proposed system \( \eta_{ex} \) was 21.03%, which was 6.78 percentage points higher than that of the reference system.

Exergy destruction and loss in the systems could be divided into four parts. The exergy efficiency enhancement of the proposed system was primarily due to the decrease of exergy destruction and loss in the first part. It included the exergy destruction in the components where working fluids absorb heat from the heat source, such as in HRVG and GHEX of the proposed system and REB of the reference system.

The exergy destruction in this part of the proposed system was 11.38 kW. The exergy destruction in this part of the reference system existed in REB and reached 33.83 kW, which was much higher than that in the proposed system.

4.2. Energy saving mechanism analysis

To analyze the heat exchange processes between the working fluids and heat source more intensively, the \( t-Q \) diagrams of the heat exchange processes in the proposed and
The heat duty $Q$ was normalized by the thermal energy of flue gas $Q_f$ to show the fraction of the heat source energy utilized in the system.

In the proposed system [Fig. 2(a)], high-temperature portion of the flue gas heat was used in HRVG, thereby heating the working fluid in the power subsystem to generate superheated vapor for power generation. Low-temperature portion of the flue gas heat was absorbed by the NH$_3$/H$_2$O solution in the absorption refrigeration subsystem to preheat the basic solution, resulting in heating load reduction of the reboiler.

The temperature match between the hot gas and cold fluids was well organized by the integration, resulting in low exergy destruction and loss during heat source energy utilization. In the reference system [Fig. 2(b)], mid-temperature flue gas was used to directly heat the basic solution, and temperature decreased from 350 °C to 126 °C, whereas the generation temperature of the two-stage NH$_3$/H$_2$O refrigeration system was 150 °C. The exergy destruction in REB was significant because of the serious mismatch in temperature between the hot and cold fluids (with the maximum temperature difference of 200 °C).

Furthermore, the REB in the proposed system used TUR exhaust vapor as heat source, and the heating load of REB was decreased by the heat recovery in GHEX and VHEX (vapor heat exchanger). By choosing a proper TUR back pressure, the exhaust vapor temperature could match the temperature of the solution in REB. In this way, the energy utilization was improved in the proposed system by reducing the exergy destruction and loss. When the quality of the heat source was at the same level, the energy consumption could be reduced.

Table 3. Comparison of the exergy distributions in the proposed and reference systems

<table>
<thead>
<tr>
<th>Items</th>
<th>Proposed system</th>
<th>Reference system</th>
<th>kW</th>
<th>kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exergy input</td>
<td>109.32</td>
<td>109.32</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Exergy destructions and loss</td>
<td>86.33</td>
<td>93.74</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 Heat exchange processes with heat source</td>
<td>11.38</td>
<td>33.83</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HRVG (REB)*</td>
<td>10.17</td>
<td>33.83</td>
<td></td>
<td></td>
</tr>
<tr>
<td>GHEX</td>
<td>1.21</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 Heat exchange processes with heat sink</td>
<td>31.65</td>
<td>33.49</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Flue gas loss</td>
<td>15.91</td>
<td>14.20</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(Mid-pressure) Absorber</td>
<td>12.49</td>
<td>11.34</td>
<td></td>
<td></td>
</tr>
<tr>
<td>CON2 (CON)</td>
<td>3.25</td>
<td>7.96</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3 Internal heat exchange processes</td>
<td>23.78</td>
<td>24.32</td>
<td></td>
<td></td>
</tr>
<tr>
<td>SHEX (HSHE)</td>
<td>3.87</td>
<td>8.77</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(MSHE)</td>
<td>-</td>
<td>1.14</td>
<td></td>
<td></td>
</tr>
<tr>
<td>REB</td>
<td>2.23</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>REC and PC</td>
<td>6.99</td>
<td>6.87</td>
<td></td>
<td></td>
</tr>
<tr>
<td>VHEX</td>
<td>3.37</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(Mid-pressure) Subcooler</td>
<td>0.41</td>
<td>0.24</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(LSUB)</td>
<td>-</td>
<td>2.82</td>
<td></td>
<td></td>
</tr>
<tr>
<td>CHEX</td>
<td>6.91</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(MEVA/LABS)</td>
<td>-</td>
<td>4.48</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4 Other components</td>
<td>19.51</td>
<td>2.09</td>
<td></td>
<td></td>
</tr>
<tr>
<td>TUR</td>
<td>4.05</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>COMP</td>
<td>10.51</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pumps and valves</td>
<td>4.96</td>
<td>2.09</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Exergy output</td>
<td>22.99</td>
<td>15.58</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Exergy efficiency, $\eta_{ex}$ (%)</td>
<td>21.03</td>
<td>14.25</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* The items in () are only for the reference system.

Fig. 2 $t$-$Q$ diagrams of the heat exchange process with heat source in the (a) proposed and (b) reference systems
5. Conclusions

A new absorption–compression refrigeration system for freezing application was proposed. In this new system, the cascade use of mid-temperature heat source was implemented. The heat source could be engine flue gas, process waste heat, or solar energy. The proposed system could be used as a stand-alone unit to meet the low-temperature refrigeration load (approximately -55 °C) without additional electricity or power input.

For the proposed system, the cooling capacity per unit mass of flue gas was 62.70 kJ/kg, and COP reached 0.277; these values were higher by 45.78% and 49.73%, respectively, than those of the conventional two-stage NH₃/H₂O absorption refrigeration system. The exergy efficiency in the new system was 20.06%, which was 6.78 percentage points higher than that of the reference system.

The energy saving mechanism for the proposed system was recovered through exergy analysis. The temperature match in the heat exchange processes between the working fluids and heat source was improved dramatically. Consequently, the energy utilization was improved.

This study may provide a new efficient approach to produce low-temperature cooling energy using mid-temperature heat source.

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References


Biography

Yi Chen is a graduate student in University of Chinese Academy of Sciences. His major is Engineering Thermophysics. His research is mainly about cogeneration cycles and systems.