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An absorption–compression refrigeration system driven by a mid-temperature heat source for low-temperature applications

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ABSTRACT

An ammonia–water absorption refrigeration system is a promising way to make use of waste heat to generate cooling energy for freezing applications. When the refrigeration temperature is below −30 °C, the conventional absorption system cannot be adopted because its performance decreases dramatically. In this work, a totally heat-driven absorption–compression refrigeration system is proposed to produce cooling energy at temperatures of −40 °C to −55 °C. The proposed system comprises a heat-driven power generation subsystem using an ammonia–water mixture as the working fluid and an absorption–compression refrigeration subsystem. Simulation results showed that the coefficient of performance and the cooling capacity per unit mass of flue gas reach 0.357 and 84.18 kJ kg^{-1}, respectively. The results of a process energy analysis showed that the cycle coupling configuration of the proposed system enhances its energy cascade utilization. Furthermore, the energy saving mechanism of the proposed system was elucidated by means of an exergy analysis and a pinch point analysis. Finally, a more comprehensive comparison with a heat-driven double-stage compression refrigeration system was conducted to show the advantage of the proposed system. This work may provide a new way to produce low-temperature cooling energy by using a mid-temperature heat source.

1. Introduction

The demand for refrigeration with an evaporating temperature less than −30 °C is increasing quickly, especially for applications such as rapid freezing, storage of medical materials, high-heat-flux electronics and so on [1–4]. However, the use of single-stage absorption or compression refrigeration systems to reach such low temperatures is uneconomical [5–8]. Therefore, double-stage compression refrigeration systems are developed to achieve better performance. The performance can be further improved by adopting different appropriate refrigerants in the high- and low-temperature circuits, for example, NH_3 and CO_2 acting as refrigerants in the high- and low-temperature stages of the cascade compression refrigeration system, respectively [6–12]. The maximum COP (coefficient of performance) of a cascade compression refrigerator reaches 1.01–1.06 when the evaporation temperature is −55 °C and the condensation temperature is 35 °C [1]. However, high electricity consumption is still the main disadvantage of such double-stage compression refrigeration systems.

To reduce electricity consumption while obtaining a low refrigeration temperature, researchers developed an absorption–compression refrigeration system through the combination of an absorption subcycle and a compression subcycle driven by heat and power, respectively. Fernández-Seara et al. [9] studied a cascade refrigeration system with a CO_2 compression system for the low temperature stage and an NH_3/H_2O absorption system for the high temperature stage to generate cooling energy at −30 °C to −50 °C. The two subcycles share one heat exchanger, which operates simultaneously as the condenser of the compression system and the evaporator of the absorption system. The COP of the system can reach 0.253. Garimella and Brown [2] developed a cascade absorption–compression system, which coupled a single-effect $LiBr/H_2O$ absorption cycle and a subcritical CO_2 vapor-compression cycle to generate low-temperature refrigerant (−40 °C) for high heat flux electronics applied in a naval ship.

The absorption–compression systems described above are very complicated because they include a large amount of equipment, and only energy exchange occurs between the absorption and the compression subcycles. In addition, the working fluids of the two subcycles are often different. When the two subcycles adopt the

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same type of working fluid, the system can become an open-style absorption–compression refrigeration system with both energy and working fluid exchanged between the absorption and compression subcycles, and the system is also simpler. The open-style absorption–compression refrigeration system can be seen as a system in which a compressor is added to a traditional absorption refrigeration system, and the compressor can lie at different positions. Kang et al. [10] analyzed four different types of open-style absorption–compression refrigeration cycles (the cycle was called the GAX hybrid cycle in the literature). Their results showed that with a compressor placed between the evaporator and the absorber, when the evaporation pressure is the same as that in the standard cycle, the COP can be improved by 24% compared with the standard cycle. When the absorption pressure is the same as that in the standard cycle, the evaporation temperature can be very low. With a compressor placed between the condenser and the desorber, when the condensation pressure is the same as that in the standard cycle, the maximum desorption temperature can be reduced by approximately 30 °C. When the desorption pressure is the same as that in the standard cycle, hot water at a temperature as high as 106 °C could be obtained by absorbing the condensation heat. Ramesh Kumar and Udayakumar [11,12] focused on a GAX absorption–compression cycle with a compressor placed between the evaporator and the absorber and found that the maximum COP occurs at an optimum degassing range and an optimum compression ratio.

To further reduce or even completely eliminate the electricity consumption for low-temperature applications, some other types of heat-driven refrigeration systems were also proposed and analyzed. Rogdakis and Antonopoulos [3] studied a NH_3/H_2O absorption refrigeration system operating at three pressure levels, which could be used to produce refrigeration temperatures as low as −70 °C. At an ambient temperature of 30 °C, the theoretical coefficient of performance ranges from 0.40 to 0.03 when the lowest temperature is in the range from −30 °C to −64 °C. He et al. [13] proposed a novel absorption refrigeration system using R134a and R23 mixed refrigerants and DMF as the solvent. The new system uses a two-stage absorber in series to reduce the evaporation pressure, and they reported that the lowest refrigeration temperature reached −62.3 °C with a COP of 0.023 at a generation temperature of 184.4 °C.

In this paper, to more efficiently utilize a mid-temperature heat source for low-temperature applications, a heat-driven absorption–compression refrigeration system is proposed and evaluated. Furthermore, the cycle coupling configuration of the new refrigeration system was studied, and its energy saving mechanism was elucidated. In addition, the system was compared with an equivalent refrigeration system to show the advantage of the proposed system.

2. System description

2.1. Description of the proposed system

Fig. 1 shows the configuration and flow of the proposed HACRS (heat-driven absorption–compression refrigeration system). It consists of a power generation subsystem using a mixture for the working fluid and an absorption–compression refrigeration subsystem. The heat from the mid-temperature heat source is converted to power and low-temperature heat through the power generation subsystem, and then the power and heat are used to drive the compressor and reboiler in the absorption–compression subsystem, respectively. NH_3/H_2O solution is chosen as the working fluid in both subsystems. The *p-T-x* diagram is given in Fig. 2 to illustrate the changes of the temperature, pressure and concentration of each stream in Fig. 1. The state points in the diagram correspond to the stream numbers in the schematic configuration. The dash dot line labeled with $x = 1.00$ depicts the saturated state line of pure ammonia, and the other dash dot lines mean the saturated solution lines, which represent the ammonia mass concentration of 0.460, 0.352 and 0.285 from left to right, respectively.

The external heat source (taking engine flue gas as an example) successively goes into a HRVG (heat recovery vapor generator) and a GHEX (gas heat exchanger). In the power generation subsystem, the high-temperature portion of the external heat source is utilized to generate superheated ammonia–water vapor, which expands across a turbine (TUR) to produce power. Corresponding to Fig. 2, the state change 1-2-3-4-(1) means the power generation subsystem. The ammonia mass concentration of the working fluid is 0.460, and state 1 denotes saturated solution. State change from 3 to 4 denotes the expansion process in TUR. The absorption–compression refrigeration subsystem can be divided into solution subcycle and refrigerant streams. An ACOM (ammonia compressor) is added between the subcooler (SUBC) and ABS (absorber). State 7, 8 and 10 are the same, because stream 8 and stream 10 are the branches of stream 7. State change 6-7-8-9 and 6-7-10-11-12 denote the basic solutions from the outlet of ABS into REC, as one section of the solution subcycle. The basic solution is at saturated state at state 6. State change 21-22-23-(6) is the other section of the solution subcycle, i.e., the weak solution streams. State change 13-14-15-16-17-18-19-20-(6) represents the refrigerant streams. The compressor boosts the pressure from the evaporation pressure to the absorption pressure, corresponding to the state change from 18 to 19. The power consumption of the compressor (W_{ACOM}) is met by the work output of the turbine in the power generation subsystem. The high-temperature exhaust vapor of the turbine and the superheated outlet vapor of the compressor provide heating load in the reboiler (REB). The low-temperature exhaust vapor of the turbine and the low-temperature portion of the external heat source preheat the basic solution sequentially (in CON1 and GHEX). There is no demand for external power in this novel absorption–compression refrigeration system; instead, only a mid-temperature heat source is needed.

2.2. Description of the reference system

A HDCRS (heat-driven double-stage compression refrigeration system) was chosen as the reference system in this study. Its schematic configuration and *p-T* diagram are illustrated in Figs. 3 and 4, respectively. The reference system can be separated into a low-parameter water-steam Rankine power subsystem [14] and a CO_2/NH_3 cascade refrigeration subsystem [1]. The low-parameter water-steam Rankine power subsystem recovers the waste heat from the mid-temperature heat source to produce superheated water-steam in the heat recovery steam generator (HRSG). The high-pressure water steam is expanded in the steam turbine (TUR) and then condensed in the condenser (CON1). The state change 1-2-3-(4) denotes the water-steam Rankine power subsystem. The CO_2/NH_3 cascade refrigeration subsystem consists of two single-stage compression refrigeration systems connected by a heat exchanger (cascade heat exchanger, CHEX). The low-temperature stage with CO_2 as refrigerant is used to offer the necessary refrigeration capacity for low-temperature applications, and the high-temperature stage with NH_3 as the refrigerant is used to condense the CO_2 in the low-temperature stage. In CHEX, the evaporation temperature (state 9–12) is 5 K lower than the condensation temperature (state 7). The state change 5-6-7-(8) and 9-10-11-(12) represent the CO_2 compression refrigeration subsystem and the NH_3 compression refrigeration subsystem,