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Performance evaluation and optimal configuration analysis of a CO_2/NH_3 cascade refrigeration system with falling film evaporator–condenser

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ABSTRACT

This study presents a CO_2/NH_3 cascade refrigeration system, in which a falling film evaporator–condenser is used as the cascade heat exchanger. The thermodynamic analysis results of the proposed system show an improvement in the coefficient of performance (COP) due to the smaller temperature difference provided by this type of cascade heat exchanger. Furthermore, an effectiveness-NTU method based model is developed by considering the constraint of the total thermal conductance. The developed model is subsequently used to examine the influences of the main parameters on the system configurations under the maximum COP condition. Results obtained reveal that, when the overall system COP is maximized, thermal conductance allocation ratios are dominated mainly by the temperature differences of the three heat exchangers and the effectiveness factors of the condenser and the evaporator. This study could contribute to the further development and the optimal design of CO_2/NH_3 cascade refrigeration systems.

1. Introduction

Cascade refrigeration systems have been widely used in commercial and industrial applications where quite low temperatures are required. The cascade refrigeration system incorporates two or more refrigeration circuits with relevant refrigerants. Earlier some HCFC or HFC refrigerants, such as R22, R23 and R404A, are commonly used in cascade refrigeration systems. Due to the harmful effect of those refrigerants on the environment, the use of natural refrigerants as alternative refrigerants have attracted renewed interest in recent years. Especially, the main group of natural refrigerants, NH_3 and CO_2, are now being chosen for use in cascade refrigeration systems [1]. The application of CO_2/NH_3 in cascade refrigeration systems has been made possible through an enormous effort from industry, research institutes and universities [2–5]. Therefore, the development of cascade refrigeration systems using natural refrigerants CO_2/NH_3 could be regarded as a feasible strategy for commercial or industrial refrigeration industry.

As well known, in a two stage cascade refrigeration system the two refrigeration circuits are thermally coupled through an intermediate cascade heat exchanger, i.e. evaporator–condenser. The intermediate operating temperature (condensing temperature of the low-temperature circuit or evaporating temperature of the high-temperature circuit) and the temperature difference in the cascade heat exchanger play important roles in determining the configuration and optimal performance of the cascade system [6–9]. Currently, several types of cascade heat exchangers such as plate, shell-and-plate or shell-and-tube heat exchangers can be employed for cascade systems to couple the two refrigeration circuits. Due to the characteristics of heat transfer process in those heat exchangers, the temperature difference is usually chosen to be in a range of 5–10 °C. According to the second law of thermodynamics, however, the use of heat exchangers with a small temperature difference as the cascade heat exchangers is beneficial to improving the system performance. For this purpose, falling film type heat exchangers could be another attractive option for the cascade refrigeration system because this kind of heat exchangers may have a smaller temperature difference in heat transfer due to its flow and heat transfer characteristics. In fact, falling film type heat exchangers have been widely applied in air separation, chemical industries, etc. [10]. Therefore, the falling film type heat exchangers including vertical plate-fin type and shell-and-tube type heat exchangers are proposed to be as the cascade heat exchanger in the CO_2/NH_3 cascade refrigeration system.

In present study, the performance evaluation of a CO_2/NH_3 cascade refrigeration system with a falling film evaporator–condenser has been carried out to find the optimum operating regime of intermediate temperature when the evaporator–condenser has a smaller temperature difference. Furthermore, an ε-NTU method (effectiveness-number of transfer units) based model for the cascade refrigeration system has been developed and the detailed analyses on the optimal configuration of the system have been also conducted. The objective of this study is to provide a guide for construction, design and theoretical evaluation of a CO_2/NH_3 cascade refrigeration system with falling film evaporator–condenser and

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Nomenclature

C	heat capacity rate (kW K^{-1})
CO_2	carbon dioxide
COP	coefficient of performance
HTC	high-temperature circuit
h	specific enthalpy (kJ kg^{-1})
LTC	low-temperature circuit
$\dot{m}$	mass flow rate (kg s^{-1})
NH_3	ammonia
NTU	number of transfer unit
$\dot{Q}$	heat transfer rate (kW)
R	thermal conductance allocation ratio
T	temperature (°C or K)
ΔT	temperature difference (°C or K)
UA	thermal conductance (kW K^{-1})
v	specific volume (m^3 kg^{-1})
$\dot{W}$	compressor's power(kW)

Greeks symbols

ε	effectiveness factor of heat exchanger
η_{sh}	isentropic efficiency of HTC compressor
η_{sl}	isentropic efficiency of LTC compressor
η_{vh}	volumetric efficiency of HTC compressor
η_{vl}	volumetric efficiency of LTC compressor
ξ_f	heat capacity rate ratio of the cooling medium and the coolant
ξ_r	mass flow rate ratio of the refrigerants in HTC and LTC
ξ_{rv}	theoretical volumetric flow ratios of the two compressors
π	compression ratio of compressor

Subscripts

c	condenser
e	evaporator
fc	cooling medium in the condenser
fe	coolant in the evaporator
h	high-temperature circuit
i	inlet
l	low-temperature circuit
m	evaporator–condenser
max	maximum
mc	intermediate condensing
me	intermediate evaporating
opt	optimum
s	overall system

support the development of CO_2/NH_3 cascade refrigeration systems.

2. System description and modeling

Fig. 1(a) schematically shows a CO_2/NH_3 cascade refrigeration system with falling film evaporator–condenser. The system still consists of the high-temperature circuit (HTC) with ammonia and the low-temperature circuit (LTC) with carbon dioxide. In this system, however, a falling film evaporator–condenser is considered as the intermediate cascade heat exchanger. In this case, the system is equipped with two gas–liquid separators in HTC and LTC, respectively. This system layout may ensure realizing the falling film evaporation and condensation in a falling film type cascade heat exchanger. In this system, the saturated liquid NH_3 from the gas–liquid separator of HTC flows downward to the evaporation channels of evaporator–condenser through an inside liquid distributor. At the same time, the generated saturated vapor CO_2 from the gas–liquid separator of LTC enters the condensing channels of evaporator–condenser. In the evaporator–condenser, NH_3 evaporates to a saturated vapor on one side while CO_2 condenses to a saturated liquid on another side. On the other hand, the resulting saturated vapor NH_3 from the evaporator–condenser mixes the saturated vapor NH_3 from the gas–liquid separator and then flows to the compressor of HTC. In LTC, the resulting saturated liquid CO_2 from the evaporator–condenser returns the gas–liquid separator and then partially vaporizes due to cooling the superheating vapor CO_2 from the compressor. The residual saturated liquid CO_2 then flows to the expansion valve of LTC. Note that the other main components used in the system and their working processes are the same as those of a conventional CO_2/NH_3 cascade refrigeration system. For example, the condenser in HTC rejects a heat flow to its cooling medium, and the evaporator in LTC, absorbs heat from the coolant or the refrigerated space. Fig. 1(b) shows the detailed processes for both HTC and LTC cycles in a schematic P–h diagram.

In order to evaluate the thermodynamic performances and obtain the main component configuration relationship of the above described system, the ε-NTU method for heat exchangers and common thermodynamic cycle analysis method are applied to develop the mathematical model. For modeling, some assumptions are also considered:

(1) All components are assumed to be a steady-state and steady-flow process.
(2) The isentropic efficiencies of compressors are taken into account.
(3) The throttling process in expansion valves is isenthalpic.
(4) Refrigerant pressure drop and heat losses in the system are neglected.

Based on the mass and energy conservation, the governing equations for components may be written as follows:

For the condenser, the heat transfer rate is

$$\dot{Q}_c = \dot{m}_h(h_2 - h_3) \tag{1}$$

$$\dot{Q}_c = \varepsilon_c C_{fc}(t_c - t_{fc,\,i}) \tag{2}$$

$$\varepsilon_c = 1 - \exp(-\mathrm{NTU}_c) \tag{3}$$

$$\mathrm{NTU}_c = UA_c/C_{fc} \tag{4}$$

where $\dot{m}_h$, h_2 and h_3 are the mass flow rate of the refrigerant in HTC, the specific enthalpies of the refrigerants at the inlet and outlet of the condenser, respectively; T_c is the condensing temperature of HTC, $T_{fc,\,i}$ is the inlet temperature of the cooling medium in the condenser, ε_c is the effectiveness factor of the condenser, C_{fc} is the heat capacity rate of the cooling medium, UA_c is the thermal conductance of the condenser, NTU_c is the number of transfer units for the condenser. From Eqs. (2)–(4), we obtain expressions as

$$\dot{Q}_c = \frac{\varepsilon_c}{\ln\left(\frac{1}{1-\varepsilon_c}\right)} UA_c(T_c - T_{fc,i}) \tag{5}$$

$$C_{fc} = \frac{UA_c}{\ln\left(\frac{1}{1-\varepsilon_c}\right)} \tag{6}$$