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Performance study on a low-temperature absorption–compression cascade refrigeration system driven by low-grade heat

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ABSTRACT

This paper presents a performance study on a low-temperature absorption–compression cascade refrigeration system (LACRS), which consists of an absorption subsystem (AS) and a vapor compression auto-cascade subsystem (CS). In the system, low-grade heat of AS is used to subcool the CS, which can obtain cold energy at −170 °C. A simulation study is carried out to investigate the effects of evaporating temperature and low-grade cooling capacity on system performance. The study results show that as low-grade cooling capacity from the AS is provided to the CS, high-grade cooling capacity increases, compressor power consumption decreases, and the COP of the CS therefore increases. Comparing with compression auto-cascade cycle, the largest COP improvement of LACRS is about 38%. The model is verified by experimental data. An additional high-grade cooling capacity is obtained experimentally at −170 °C. The study results presented in this paper not only demonstrate the excellent performance of the LACRS, but also provide important guidance to further system design, and practical application.

1. Introduction

Recently, in order to save energy, there have been significant interests in utilizing various low-grade heat energy e.g., solar energy, waste heat energy, and geothermal energy [1–3]. In refrigeration engineering, low-grade heat energy may be used in sorption, ejection and desiccant cycles to provide useful cold energy. However, a very low evaporating temperature cannot usually be obtained with these cycles [4,5], limiting their practical applications. In addition, the efficiencies of these cycles are also relatively low as their COPs drop quickly at low evaporating temperatures [6–8]. However, a low-grade heat driven refrigeration cycle can be combined with a vapor compression refrigeration cycle to form a hybrid cycle to take the advantages of both cycles. This improves the efficiency of energy utilization [9–11].

An absorption–compression hybrid cycle is a typical example of combined cycles. Pratihar et al. [12] studied an absorption–compression cycle in which its compressor located between its absorber and desorber. The relationship between COP and relative SHX area was studied, the optimum COP was obtained at 39.0% relative SHX area. An absorption–compression cycle with a compressor located between its evaporator and absorber was studied by Wu et al. [13]. The maximum energy saving rates were 17.7–29.2% in the study case. Han et al. [14] investigated a novel absorption–compression cycle. The COP of the proposed system was 41.9% higher than that of a conventional ammonia–water absorption refrigerator, and it generated 46.7% more cooling energy. Furthermore, other innovative modifications in cycles were reported [15–17].

In the previously mentioned cycles, their compressors are directly connected to other system components filled with solution. Therefore, these compressors should be oil free and corrosion resistant, leading to high initial cost. Furthermore, system stability is also in question. Therefore, absorption–compression cascade system consisting of an absorption subsystem (AS) and a compression subsystem (CS) is suggested. The evaporator of the AS is also the condenser of the CS. Between the two subsystems, there is only heat transfer but no mass transfer. A study on an absorption–compression cascade system was carried out by Jain et al. [18]. The electric power consumption was reduced by 61.0% and the COP of was improved by 155.0% to a traditional vapor compression refrigeration system. Cimsit et al. [19] reported an optimization study on an absorption–compression system. The system had potential to reduce electric energy consumption by 50.0%. The COP and exergetic efficiency were improved about 7.0% and 3.1% respectively. Jain et al. [20] analyzed three configurations of absorption–compression cascade refrigeration system. The results

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Nomenclature

A	heat transfer area (m^2)
AS	vapor absorption refrigeration subsystem
COP	coefficient of performance
CS	vapor compression refrigeration subsystem
CV	control volume
d	density (kg/m^3)
H	enthalpy (kJ/kg)
h	heat transfer coefficient ($W/(m^2\,K)$)
K	coefficient of thermal conductivity (W/(m K))
LACRS	low-temperature absorption–compression cascade refrigeration system
$\dot{m}$	mass flow rate (kg/s)
M_m	molar mass (kg/kmol)
P	pressure (MPa)
Q	heat transfer rate (W)
r	fouling resistance (K/W)
R	molar ratio
T	temperature (°C)
UA	overall heat transfer coefficient (W/K)
UR	upgrade ratio
V_C	volume flow rate of the compressor (m^3/s)
W_{CS}	compressor power consumption (W)
x	mass fraction
Z	refrigerant discharged by compressor
ΔQ	difference of heat transfer rate (W)
ΔT	difference of temperature (°C)
δ	thickness (m)
η	efficiency

Subscript

cal	calculating
cold	the cold side of heat transfer
con	condensing
conv	conventional
dis	discharge
eva	evaporating
exp	experimental
gen	generator
hg	high-grade
hot	the hot side of heat transfer
lg	low-grade
i	component number of Mixture
i1	the inlet of the first control volume
o1	the outlet of the first control volume
rec	recuperator
s	isentropic
suc	suction
v	volumetric
w	tube wall
1–32	state point

showed they reduced electricity consumption by 50.0%, 76.8% and 88.3% respectively. Chen et al. [21] proposed a novel absorption–compression cascade system where low-grade (high temperature) cooling capacity (Q_{lg}) provided by its AS subcooled the refrigerant in the CS, but did not condense it. Hence, the high-grade cooling capacity (Q_{hg}) from the CS can be increased by the same amount, i.e., $Q_{hg} = Q_{lg}$. This is to say that low-grade cooling capacity could be totally upgraded to high-grade cooling capacity. A modeling study on such a system was carried out by He et al. [22]. The system COP was improved by 75.0% and the optimized AS evaporating temperature of 15.3 °C was obtained.

However, the evaporating temperatures in all the above hybrid systems are not lower than −100 °C. In fact, in the temperature range lower than −100 °C, vast applications of conventional refrigeration technologies are consuming huge quantity of energy, such as auto-cascade compression cycle. Therefore, refrigeration cycles which save energy are strongly desired in this temperature range.

Following Chen et al.'s work [21], a novel low-temperature absorption–compression cascade refrigeration system (LACRS) was proposed recently [23,24], which based on auto-cascade compression cycle and could use low-grade heat to help obtain cooling capacity at as low as −170 °C. It not only consumes less electric energy than traditional auto-cascade compression system, but also greatly expands the application range of absorption–compression refrigeration system. However, this system has not been carefully studied yet. To enable the practical application of LACRS, a lot of studies are necessary, such as improving system performance, investigating the variation range of system operating parameters, and evaluating energy efficiency. The purpose of this paper is to understand the system performance and operating parameters by carrying out a simulating and experimental study. This study can also provide important theoretical guidance to the further experimental investigation, system design and practical application.

2. Model development for the LACRS

2.1. Parameters

The LACRS consists of an absorption subsystem (AS) and a vapor compression subsystem (CS), as shown in Fig. 1. The CS is an auto-cascade compression refrigeration system with a rectification column. The AS is an auto-cascade absorption system, which provides low-grade cooling capacity (Q_{lg}) to precool the saturated liquid from the bottom of the rectification column in the CS. This heat transfer process takes place in the cascade heat exchanger as shown in Fig. 1. Therefore, the high-grade cooling capacity (Q_{hg}) available from the CS would increases. Q_{lg} and Q_{hg} can be evaluated by Eqs. (1) and (2), respectively.

$$Q_{lg} = \dot{m}_{20}(H_{21} - H_{20}) \tag{1}$$
$$Q_{hg} = \dot{m}_{6}(H_{7} - H_{6}) \tag{2}$$

The increase in Q_{hg} (ΔQ_{hg}) is determined as follows,

$$\Delta Q_{hg} = Q_{hg} - Q_{hg}^{0} \tag{3}$$

where Q_{hg}^{0} is the cooling capacity of the CS without utilizing the precooling from the AS. ΔQ_{hg} is upgraded from Q_{lg}. Since not all Q_{lg} can be upgraded, an upgrade ratio (UR) is determined by

$$UR = \Delta Q_{hg}/Q_{lg} \tag{4}$$

The COP of the CS is given by

$$COP_{CS} = Q_{hg}/W_{CS} \tag{5}$$

where, W_{CS} is compressor power input.

For the above three parameters, ΔQ_{hg} and UR directly indicate the energy improvement of the coupling of the two subsystems. COP_{CS} provides a general view on performance and an evaluation indicator on the energy-saving effect of the LACRS. They are key