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Thermodynamic analysis of a novel energy-efficient refrigeration system subcooled by liquid desiccant dehumidification and evaporation

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ABSTRACT

A new energy-efficient refrigeration system subcooled by liquid desiccant dehumidification and evaporation was proposed in this paper. In the system, liquid desiccant system could produce very dry air for an indirect evaporative cooler, which would subcool the vapor compression refrigeration system to get higher *COP* than conventional refrigeration system. The desiccant cooling system can use the condensation heat for the desiccant regeneration. Thermodynamic analysis is made to discuss the effects of operation parameters (condensing temperature, liquid desiccant concentration, ambient air temperature and relative humidity) on the system performance. Results show that the proposed hybrid vapor compression refrigeration system achieves significantly higher *COP* than conventional vapor compression refrigeration system, and even higher than the reverse Carnot cycle at the same operation conditions. The maximum *COPs* of the hybrid systems using hot air and ambient air are 18.8% and 16.3% higher than that of the conventional vapor compression refrigeration system under varied conditions, respectively.

1. Introduction

In recent years, refrigeration systems consumed a large amount of energy in maintaining thermal comfort for occupants and suitable climatic conditions for cooling cases, which made up 50% of building energy consumption [1]. The utilization of evaporative condensing indicated energy efficient potential in reduction of power requirements [2,3].

In a traditional refrigeration system, a great deal of condensation heat, which could be used for other purposes, is directly dissipated to the environment. The dissipated heat not only wastes energy, but also causes severe heat island effect in the surrounding areas. Many methods have been attempted to tackle these problems. Some researchers utilized the condensation heat from air conditioners to preheat domestic hot water [4–7], leading to claims that water heating in summer, primarily for bathing. It can be made available virtually free whenever space cooling is required, and is considered one of the most cost effective energy conservation measures. Several other researchers tried to add a heat recovery system on the refrigeration system. For instance, a condensing heat recovery with thermal storage of phase change material (paraffin wax as PCM) was designed and analyzed by Zhang et al. [8]. In addition, Kaushik et al. [9] introduced a canopus heat exchanger for heat recovery for a refrigeration system. A drawback with heat recovery system is the high condensing temperature that increases the energy consumption of the refrigeration system. To solve this problem, Arias and Lundqvist [10] proposed an alternative to heat recovery, namely floating condensing pressure, which improved the coefficient of performance and decreased the energy consumption of the refrigeration system at lower outdoor temperature.

From thermodynamic standpoint, further cooling of liquid refrigerant leaving condenser can significantly increase the cooling capacity and improve the system *COP*. The regenerative refrigeration cycle adopted a liquid-suction heat exchanger as a sub-cooler to improve the system coefficient of performance (*COP*) [11–13]. The liquid-suction heat exchanger achieves condensate subcooling by transferring the refrigerant heat from the condenser outlet to the compressor inlet. This subcooling method may result in the superheated degree existing at the compressor inlet and reduce the system efficiency. Therefore the liquid-suction heat exchanger is not suitable for all air conditioning systems. In order to eliminate the drawbacks of the liquid-suction heat exchanger, Khan et al. [14] and Qureshi et al. [15,16] used integrated and dedicated mechanical-subcooling methods to enhance the system *COP* and also to remove the superheated degree, respectively. Thermoeconomic considerations were given to heat exchanger inventory allocation in vapor compression cycles with mechanical subcooling by Qureshi and Zubair [17], and it was concluded that the cost optimization of the integrated mechanical subcooling system was qualitatively the same as the dedicated subcooling system. Since the

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Nomenclature

COP	coefficient of performance
C_p	specific heat capacity (kJ kg^{-1} K^{-1})
d	differential
G	mass flow rate (kg s^{-1})
h	enthalpy (kJ kg^{-1})
H	height (m)
h_C	heat transfer coefficient (W m^{-2} K^{-1})
h_D	mass transfer coefficient (kg m^{-2} s^{-1})
L	length (m)
Le	Lewis number
m	adiabatic index of refrigerant
NTU	number of mass transfer unit
P	pressure (Pa)
Q	refrigerating capacity (kW)
q_0	specific refrigerating capacity (kJ kg^{-1})
RH	relative humidity (%)
T	temperature (°C)
ΔT_{sc}	subcooling degree (°C)
ΔT_{sh}	superheat degree (°C)
w	specific power consumption (kJ kg^{-1})
W	width (m)
X	Concentration (%)

Greek symbols

ε	effectiveness
ω	humidity ratio (kg kg^{-1})

Subscripts

a	air
AHE	air-to-air heat exchanger
amb	ambient
com	compressor
con	condenser
eva	evaporator
IEC	indirect evaporative cooler
in	inlet
out	outlet
max	maximum
min	minimum
ref	refrigerant
reg	regenerator
s	solution
sc	subcooling
sh	superheat
SHE	solution-to-solution heat exchanger

mechanical-subcooling required an additional compressor to provide the driving force of subcooling, the electricity consumption of the compressor would carry the undesirable effect on the system *COP*. Besides, the expensive initial cost is also one of the drawbacks in the mechanical-subcooling method. To avoid the demerits of mechanical-subcooling method, many studies utilized phase change material and heat pipes in cold storage unit as a subcooler [18–20]. Chieh et al. [21] developed a thermal battery composed of the phase change material and the heat pipes to store energy in air conditioning application. Huang et al. [22] experimentally investigated the performance of the cold storage air conditioning system utilizing a thermal battery as a subcooler. Because the charge heat exchanger of the thermal battery had the larger thermal resistances, it induced the lower ice stored ability. For eliminating the disadvantages discussed in previous studies, Hsiao et al. [23,24] used an ice storage tank as a subcooler to enhance the system capabilities and reduces the superheated degree without inputting the supplemental electricity. By the innovative design of the charge heat exchanger, the overall thermal resistances can be decreased.

The liquid desiccant system using low-grade heat resource was proposed by Lof [25], and its application in air-conditioning systems has been widely investigated [26–28]. In this paper, a novel energy-efficient vapor compression refrigeration system combined with a liquid desiccant cooling system is proposed. The vapor compression system is subcooled by the liquid desiccant cooling cycle driven by condensing heat to achieve more subcooling degree of the refrigerant than the conventional system. This paper will discuss the potential of the performance improvement and the effect of different climatic and operating conditions on the performance of the system, and a comparative study was made on the effect of different utilization ways of condensation heat.

2. System description

Fig. 1 illustrates the novel refrigeration cycle subcooled by the liquid desiccant dehumidification and evaporation, which is composed of refrigeration cycle, closed air cycle and liquid desiccant cycle. The refrigeration cycle includes an evaporator, a compressor, a solution-to-refrigerant heat exchanger and a condenser. Refrigerant R-22 is chosen to be the working fluid inside the refrigeration cycle, and the refrigerating capacity of the baseline system is 30 kW. The closed air cycle consists of an indirect evaporative cooler, an air-to-air heat exchanger, a dehumidifier and an air cooler, while the liquid desiccant cycle is made up of a dehumidifier, a solution cooler, a solution-to-solution heat exchanger, a regenerator and a solution-to-refrigerant heat exchanger. The dehumidifier is internally cooled by cooling water and its physical size is assumed large enough to make the air outlet humidity ratio be in equilibrium with the solution inlet humidity ratio above the surface, which can be achieved as long as the cooling water is sufficient. The regenerator is adiabatic, and its physical scale at several different operating conditions is shown in Table 1. The physical scale of other components of the system is believed large enough to meet our needs.

Fig. 2(a) shows the ideal pressure enthalpy diagram of refrigeration cycle. As indicated in Fig. 2(a), the refrigerant at state 4 is subcooled to state 5 by the indirect evaporative cooler, using the dry closed air absorbing moisture from water. A conceptual schematic diagram for the processes of the liquid desiccant cycle and closed air cycle is shown in Fig. 2(b). There is sensible heat recovery of the closed air from state 7 to state 8. The moist air (state 8) is dehumidified in the dehumidifier to go to state 9. The dehumidified air is precooled by an air cooler and then cooled to state 11 by an air-to-air heat exchanger. In indirect evaporative cooler, the cold dry air is humidified by water evaporation and reaches state 7.

In the liquid desiccant cycle, state 12 and state 14 are in the iso-concentration line of liquid desiccant solution. The diluted solution is preheated by hot concentrated solution leaving the regenerator in solution-to-solution heat exchanger. Prior to entering regenerator, the liquid desiccant is heated to the specified set point temperature (state 14) in the solution-to-refrigerant heat exchanger. Then the hot diluted solution enters into the regenerator and finally reaches state 15. State 15 and state 17 are also in iso-concentration