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Experimental investigations on heat and mass transfer performances of a liquid desiccant cooling and dehumidification system

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HIGHLIGHTS

- A liquid desiccant cooling and dehumidification system is thorough studied on its performance.
- The dilution rate of desiccant solution is reduced by 39.64%.
- 22.3% energy savings can be realized for cooling and dehumidifying moisture air.

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ABSTRACT

The last decades have witnessed the growth interest in Liquid Desiccant Dehumidification Systems (LDDS). In the conventional LDDS, the high dilution rate of desiccant solution in dehumidifier leads to a high desiccant regeneration frequency, which consequently results in more thermal energy consumed by desiccant regeneration system. Therefore, a more energy efficient Liquid Desiccant Cooling and Dehumidification (LDCD) system is developed in this study, which mainly composes of a cooling coil and dehumidifier. A simple static model is proposed to predict the performances of heat and mass transfer process in this system. The thermal efficiency, moisture effectiveness and desiccant dilution rate are utilized as the performance indicators. The influences of several relevant parameters on the cooling and dehumidification performances of LDCD system are investigated. The model predictions are compared with the experimental data, and the results show that the model predictions are well in line with the experimental data with the maximum errors less than 10%. Moreover, the feasibility of LDCD system in reducing the dilution rate of desiccant solution and the system energy consumption is validated. The results indicate that the dilution rate of desiccant solution and energy consumption of the LDCD system are reduced by 39.64% and 22.3% over the conventional LDDS, respectively.

1. Introduction

The electricity consumed by Heating, Ventilating and Air Conditioning (HVAC) systems has been growing rapidly in recent decades. In Singapore, approximately 70% of total building energy is consumed by HVAC systems, and the majority is used in cooling and dehumidification applications [1]. Therefore, the Liquid Desiccant Dehumidification Systems (LDDS) have been proposed as an alternative method to obtain more energy efficient and a better indoor air quality in buildings [2,3]. Compared with the conventional air dehumidification system, LDDS not only can achieve efficient air humidity control and about 40% of operating cost saving, but also can utilize environmental friendly liquid desiccants and the dilute desiccant solutions can be regenerated by utilizing low-grade renewable energy [4–7].

The first liquid desiccant air-conditioning system was probably developed by Lof [8] in 1955, the system operating with the triethylene glycol solution as the liquid desiccant. Since then, various types of LDDS have been proposed and investigated, these systems can be classified into four categories: the counter-flow packed-type LDDS [9,10], the cross-flow packed-type LDDS [11,12], the internally-cooled/heated LDDS [13,14] and the membrane-based LDDS [15,16]. Among these, the counter-flow packed-type LDDS is widely utilized and studied. Abdulrahman et al. [17] proposed a hybrid liquid desiccant air conditioning system with counter flow configuration for greenhouse applications, the dehumidification performances were investigated by using the artificial neural network method. Langroudi et al. [18] investigated the performance of the counter flow packedtype dehumidification system through response surface methodology. Luo et al. [19]

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Nomenclature

$A_{cc/de}$	heat transfer area in cooling coil or dehumidifier (m^2)
b_1-c_7	constants (dimensionless)
c_1-c_{10}	lumped model parameters (dimensionless)
c_w	special heat capacity of chilled water (J/(kg °C))
c_s	special heat capacity of desiccant solution (J/(kg °C))
$C_{cc,min}$	minimum specific heat rate in cooling coil (W/(°C))
$C_{de,min}$	minimum specific heat rate in dehumidifier (W/(°C))
COP	coefficient of performance (dimensionless)
$E_{chiller}$	energy consumption of chiller system (kW)
e_1-e_4	constants (dimensionless)
$E_{ch,pump}$	energy consumption of chilled water pump (kW)
$E_{de,fan}$	energy consumption of fan in dehumidification system (kW)
$E_{de,pump}$	energy consumption of pump in dehumidification system (kW)
E_{re}	energy consumption of regeneration system (kW)
$E_{re,fan}$	energy consumption of fan in regeneration system (kW)
$E_{re,pump}$	energy consumption of pump in regeneration system (kW)
E_{total}	total energy consumption (kW)
$\dot{m}_a$	air mass flow rate (kg/s)
H	Henry's law constant (dimensionless)
$\dot{m}_{re,s}$	desiccant solution mass flow rate in regenerator (kg/s)
$\dot{m}_s$	desiccant solution mass flow rate in dehumidifier (kg/s)
$\dot{m}_w$	chilled water mass flow rate (kg/s)
$\dot{m}_{w,r}$	backwater mass flow rate (kg/s)
N_{de}	mass transfer rate in dehumidifier (g/s)
$P_{a,sat}$	saturated vapor pressure of air (Pa)
Q_{cc}	heat transfer rate in cooling coil (kW)
Q_{de}	heat transfer rate in dehumidifier (kW)
$T_{a,cc}$	outlet air temperature of cooling coil (°C)
$T_{a,i}$	ambient air temperature (°C)
$T_{a,o}$	outlet air temperature of dehumidifier (°C)
T_{equ}	equilibrium temperature (°C)
$T_{re,i}$	inlet desiccant solution temperature of heater (°C)
$T_{re,o}$	outlet desiccant solution temperature of heater (°C)
$T_{s,i}$	inlet desiccant solution temperature of dehumidifier (°C)
$T_{w,i}$	inlet chilled water temperature (°C)
$T_{w,r}$	backwater temperature (°C)
ω_{cc}	outlet air humidity of cooling coil (g/kg (dry air))
ω_{equ}	equilibrium humidity ratio (g/kg (dry air))
ω_i	inlet air humidity of cooling coil (g/kg (dry air))
ω_o	outlet air humidity of dehumidifier (g/kg (dry air))
φ_a	relative humidity of air (%)
η_h	thermal efficiency (%)
η_m	moisture effectiveness (%)
ξ_i	concentrations of desiccant solution before dehumidification process (%)
ξ_o	concentrations of desiccant solution after dehumidification process (%)

Subscripts

a	air
cc	cooling coil
ch	chiller system
de	dehumidifier
i	inlet
o	outlet
re	regeneration system
s	desiccant solution
w	chilled water

designed a liquid desiccant based fin-tube type internally-cooled dehumidifier with a cross-flow configuration to improve the dehumidification efficiency. The system performance was investigated simulative and experimentally. A theoretical model of the dehumidifier was developed based on the correlated heat transfer coefficient to calculate the best air flow direction length for dehumidifier. Moreover, the system dehumidification performance was also studied with a CFD model [20]. To eliminate the desiccant solution carryover, Kumar et al. [21] designed a spray tower with new types of wire mesh packing. Three different configurations of packings were tested, and the results show that the performance was improved by about 30% over the conventional spray tower.

Besides that, some new structure types of LDDS had also been designed and investigated. Xiong et al. [22] proposed a novel two-stage LDDS to improve the system thermal performance. The thermal coefficient performance of the proposed system rises from 0.24 to 0.73 over the basic LDDS. The influences of the mass flow rates of air and desiccant solution and desiccant regeneration temperature on system performance were discussed. Mcnevin and Harrison [23] designed a multi-stage dehumidification system to achieve efficient operation over a wide range of air and chilled water inlet conditions. The system performance was investigated by both of experiment and simulation. In addition, Yang et al. [24] introduced a liquid desiccant dehumidifier using ultrasonic atomization technology. The desiccant solution was sprayed in the form of tiny droplets with a diameter of 50 μm. The system dehumidification performance was investigated experimentally.

The desiccant solution is diluted in the dehumidifier. Thus, to realize the reuse of desiccant solution, the desiccant regeneration system is necessary. The largest energy requirement in LDDS is desiccant regeneration process [25,26]. And the operating cost of desiccant regeneration system is tightly related to the frequency of desiccant solution regeneration, and thus related to the dilution rate of desiccant solution in dehumidifier. With a higher desiccant dilution rate, more energy will be consumed by the desiccant regeneration system. Hence, this may result in reducing the whole system energy efficiency and not conducing to the widespread use of LDDS. It is noted that the drawback of the high dilution rate of desiccant solution in the dehumidifier exists in the above systems. To address the aforementioned drawbacks and realize efficient air temperature and humidity control, a Liquid Desiccant Cooling and Dehumidification (LDCD) system is developed. The LDCD system is combined by a cooling coil and a dehumidifier. The air is first cooled and dehumidified in cooling coil first and then dehumidified again in dehumidifier. Since the system cooling and dehumidification performance is affected by several key parameters, so a reliable performance investigation is required. In this work, the following questions are to be explored:

1. How to accurately describe the air cooling and dehumidification processes in the proposed LDCD system?
2. Do the relevant inlet parameters of the process air, chilled water and desiccant solution have any remarkable influences on the cooling and dehumidification performances of LDCD system? Are the trends of experimental results consistent with model predictions?
3. Compare with the conventional LDDS, what are the potentials of the desiccant solution dilution rate reducing and energy saving of the proposed LDCD system?

To address above questions, a platform of the LDCD system is built, and a large amount of experiments under different operating conditions are conducted. The thermal efficiency and moisture effectiveness are used to assess system cooling and dehumidification performances. The influences of the process air mass flow rate, chilled water mass flow rate