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Research Note

Performance study on the dehumidifier of a packed bed liquid desiccant system

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Mass transfer coefficient;
Optimum ASMR.

Abstract. A liquid desiccant evaporation cooling air conditioning system is introduced in this paper, in which the dehumidifier and regenerator play the most important role. For liquid-gas contact, packed towers with a low pressure drop provide good heat and mass transfer characteristics for compact designs. The experimental data have been obtained from a built prototype of a liquid desiccant system in a packed bed unit with a surface area per unit volume ratio of 125 m^2/m^3; the liquid desiccant being lithium chloride. The result showed that the mean mass transfer coefficient of the packing dehumidifier was 0.02 kg/m^2s. Also, the absorber characteristic parameter, the packing size or Number of Transfer Units (NTU) and Air-to-desiccant Solution Mass flow Rate ratio (ASMR) are crucial parameters. These parameters, which affect humidity effectiveness and enthalpy efficiency, are introduced and defined in this paper, and high efficiency may be achieved if proper values of these variables are selected. In this study, enthalpy efficiency from experimental results has been compared with analytical computation results based on optimum ASMR. From the analytical solution, an optimum mass flow rate ratio can be deduced for optimal design of both dehumidifiers and regenerators. In this study, the behavior of the absorber has also been considered.

1. Introduction

Application of desiccant air conditioner systems is proposed as an alternative solution for reducing energy consumption and greenhouse gas emissions in hot and humid locations. These systems improve Indoor Air Quality (IAQ) in residential, public and commercial buildings by delivering more fresh air. Liquid desiccant systems handle the latent load cooling in air conditioning systems, without increasing energy use. These systems also displace CFC and HCFC based cooling equipment that contributes to depletion of the Earth's ozone layer, and helps to reduce global warming [1-7].

The concept of liquid desiccant air conditioning systems goes back to the use of triethylene glycol as a desiccant to dehumidify air, and was described by Lof in 1955 [1]. Researchers show that desiccant cooling systems can reduce overall energy consumption by shifting the energy used away from electricity and towards renewable, cheaper fuels and waste energy. It is a good application for solar energy [1-7]. Commercially available liquid desiccants include lithium bromide, lithium chloride, triethylene glycol and calcium chloride. Lithium chloride and calcium chloride are the universal desiccants mostly used, lithium chloride having excellent regeneration performance but high cost, while calcium chloride has lower performance and low cost.

Al-Farayedhi et al. listed several important con-

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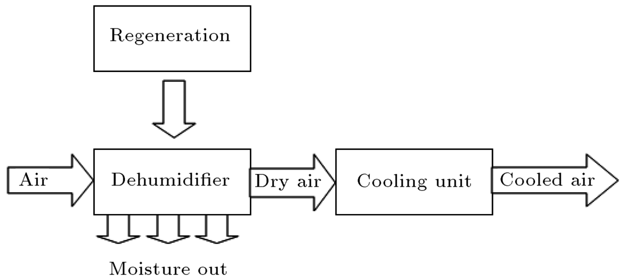

Figure 1. Main component of a desiccant cooling system.

siderations in choosing the liquid desiccant solution for dehumidification [2]. The main components of a desiccant cooling system are the absorber or dehumidifier, regenerator and cooling unit (Figure 1). A common configuration for de-humidifiers and regenerators includes a finned-tube surface, coil-type absorber, spray tower, and packed tower. There are generally three flow patterns for the dehumidifier, namely; parallel flow, counter flow and cross flow [3].

Recently, membrane-based liquid desiccant air dehumidification has also been utilized. Compared to common liquid desiccant air dehumidification, it has several benefits: a) Membranes are used to separate the solution and process air. Solution droplet cross-over, a serious problem encountered in traditional liquid desiccant air dehumidification, is prevented; b) The packing density of hollow fibers is high, so heat and mass transfer capabilities are high. Turbulent fluid flow and conjugate heat mass transfer in the cross-flow hollow fiber membrane contactor for liquid desiccant air dehumidification is solved based on a free surface model. The fluid flow and mass transfer in these systems have been investigated by many researchers [8-11].

One example of liquid desiccant cooling systems is represented in Figure 2. Moist air enters the bottom of the dehumidification tower and travels up through a packing material. The cool strong liquid desiccant enters the top of the tower and travels down the packing materials, counter current to the air. Since the cool strong desiccant vapour pressure is less than the moist air vapour pressure, water vapour is transferred from the air to the liquid desiccant, and dehumidified air leaves the absorber tower [12].

Packed bed liquid desiccant for three pattern flow in a dehumidifier has been studied by many researchers [13-18]. These researchers developed models based on earlier works of Olander [3,4] and Treybal [15]. Stevens et al. [16] introduced an effectiveness (ε-NTU) relationship. Sadasivam and Balakrishnan [19] defined NTU on the basis of minimum capacity fluid, which is consistent with the NTU definition used in heat exchanger design. Many researchers studied the effect of inlet parameters on outlet parameters and the performance of the regenerator and dehumidifier.

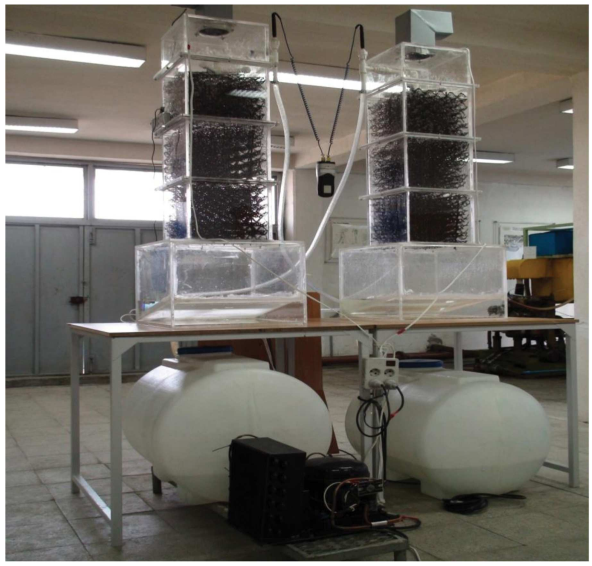

Figure 2. Main components of the packed bed liquid desiccant cooling system model.

Patnaik et al. [17] studied the performance of a packed bed absorber and regenerator. In their study, the air to solution mass (ASMR) ratio in the absorber varied between 1.3 and 3.3, and, at which ratio, efficient energy storage was not achievable. Sadasivam and Balakrishnan [19] used a packed tower absorber with LiBr-H2O solution, and found that the optimal value for ASMR is between 1.4 and 2.

Mago and Goswami [20] tested a hybrid solar liquid desiccant cooling system and found that in this system, the air conditioning performance improves by decreasing the outlet humidity and temperature of the air.

Kessling et al. [18] state that for a high energy storage capacity, a high air to solution mass ratio is required to achieve great differences between salt inlet and outlet concentrations.

Fumo and Goswami [21] studied the performance of packed tower absorbers and regenerators by mathematical and experimental methods, and analysed the influence of the major thermodynamics variables, individually; for example, desiccant flow rate and air flow rate. Gommed and Grossman [6] gave a mass transfer coefficient of 0.02 kg/m^{2}s, and an effective surface area of packing material of 44.5 m^2. Yin et al. [22] used liquid solution mass flow rate to air flow rate ratios of 1.3912 and 0.7269 for a dehumidifier and a regenerator, respectively, but the method used for these ratios was not discussed in this study.

In this paper, an experimental investigation into the performance of a packed-tower dehumidifier, including its mass transfer coefficient, is reported. We show the influence of dimensionless parameter air to liquid desiccant flow rate ratio (ASMR) on its performance, which was studied under different operating

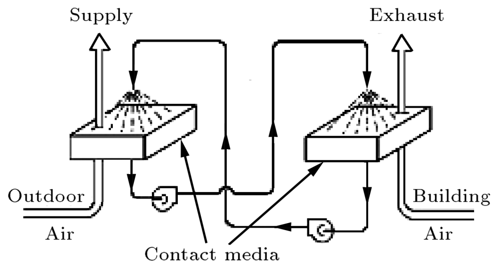

Figure 3. Tower enthalpy recovery loop [23].

conditions. The results will be valuable to the system, optimizing operation under the specified configuration.

2. Methodology

According to ASHRAE handbook guidelines [23], a loop for an enthalpy recovery system was utilized (Figure 3). The Lewis number and NTU are defined by these equations:

$$L_e = \frac{h_c}{h_d C_{pa}}, \tag{1}$$

$$\text{NTU} = \frac{h_d \text{VLA}}{m_a}. \tag{2}$$

For this system, the efficiency can be defined by Eq. (3), and the humidity effectiveness is defined by Eq. (4).

$$\varepsilon_h = \frac{h_{ai} - h_{ao}}{h_{ai} - h_{ar}}, \tag{3}$$

$$\varepsilon_\omega = \frac{\omega_{ai} - \omega_{ao}}{\omega_{ai} - \omega_{eo}}. \tag{4}$$

Chen et al. [13] showed through analytical solution that if the Lewis number is 1, i.e. enthalpy and humidity efficiency can be determined by the following three equations:

$$\varepsilon_\omega = \frac{\frac{1-e^{-\text{NTU}(1-m^*)}}{1-m^* e^{-\text{NTU}(1-m^*)}}}{1-e^{-\text{NTU}}}, \tag{5}$$

$$\varepsilon_h = \frac{1-e^{-\text{NTU}(1-m^*)}}{1-m^* e^{-\text{NTU}(1-m^*)}+1-m^*}, \tag{6}$$

$$m^* = \frac{\dot{m_a} C_{pa}}{\dot{m_1} C_{pe}}. \tag{7}$$

3. Experimental set-up and procedure

The experimental apparatus is designed and set-up in the moderate and humid climate of Iran, in the southern region of the Caspian Sea. In this location, the average high temperature in the summer is about 35°C and relative humidity is 80-85%.

Figure 2 is a photograph of the liquid desiccant system designed by the authors of this paper and manufactured in this location. For this condition, from a finite difference model, the packing height of the dehumidifier was selected to be 85 cm [12], the size varying for other conditions. The packing material used was polypropylene with a specific surface area of 125 m^2/m^3. As shown in Figure 2, the absorber and regenerator towers have 30 cm × 30 cm cross sectional area. A circulation pump was used to circulate the lithium chloride, which was used as the desiccant. Unused desiccant was stored in a tank, and its temperature adjusted by heat exchange with the environment. Axial extract fans were installed on top of the towers, to extract the outdoor air from the bottom of the towers, and counter flow to the lithium chloride in the absorber and regenerator. The desiccant was distributed by spray, which was located at the top of the tower. The instruments that were used to measure different variables were:

(a) A portable digital hot wire anemometer was used to measure the air velocity and air flow rate.

(b) A portable digital humidity meter, range of 10% to 95% R.H and a resolution of 1%, was used to measure the Relative Humidity (RH).

(c) The temperature of air and the liquid desiccant temperature were measured by K-type thermocouples. This instrument was used for measuring the air dry bulb temperature, dew point and desiccant temperature. Using the wet bulb and dry bulb temperatures of the air, the humidity of the air can be obtained.

(d) Rota meters with a range of 100 to 1000 liters per hour (l/h) were used to measure the desiccant flow rate.

(e) A refract meter was used to measure the desiccant concentration.

The main features of the different measurement devices are shown in Table 1.

4. Results and discussion

The performance of a liquid desiccant cooling system depends on the thermodynamics variables, such as air and desiccant flow rate, air temperature and humidity,

Table 1. Specification of different measurement devices.

Devices	Type	Accuracy
Thermometers	K-type thermocouple 0412-yk90HT	0.1°C
Air flow meter	Hot wire anemometer YK-2004AH	0.001 m^3/min
Solution flow meter	Rota meter	10 L/h
Humidity meter	YK-90HT	3%

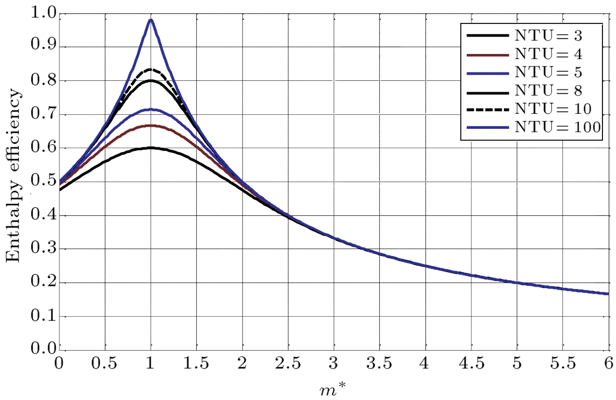

Figure 4. Effect of m^* and NTU on the enthalpy efficiency.

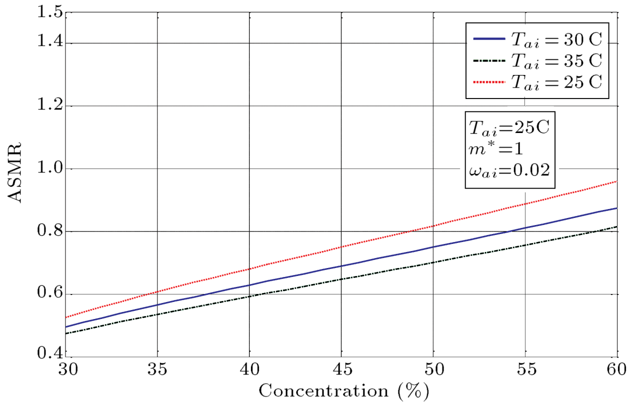

Figure 5. Effect of ASMR on concentration.

desiccant concentration and temperature. Air to solution mass flow rate ratio "ASMR" is an important parameter for dehumidification and energy storage. In liquid desiccant air conditioning systems, a high desiccant flow rate increases the dehumidification rate. However, a high desiccant flow rate in this system is not the optimal choice, because, at high desiccant flow rate, regeneration energy cost, desiccant cost and transport cost will increase.

Eq. (6) is plotted in Figure 4. In this figure, enthalpy efficiency is plotted against m^* for different NTU values. This figure shows that there is an optimum number for the efficiency of enthalpy. In this figure, if desiccant flow rate is infinite, i.e. $m^* = 0$, the efficiency is equal to 0.5, and, for large NTU values, if m^* is equal to 1, i.e. $m^* = 1$, the efficiency is equal to 1 ($\varepsilon_h = 1$). This means that there is an air-to-desiccant flow rate ratio (ASMR) to obtain maximum enthalpy efficiency. Optimum m^* varies between 0 and 1, and any m^* value larger than 1 is not considered efficient.

Since m^* is a function of the specific heat of air, the solution and ASMR, using this formula (Eq. (7)), we can obtain the optimum air-mass to solution-mass flow rate ratio (ASMR) for any given condition.

$$\text{ASMR} = \frac{\dot{m_a}}{\dot{m_l}} = \frac{m^* C_{pe}}{C_{pa}}. \tag{8}$$

The specific heat of the solution is a function of solution concentration and temperature. The specific heat of air in equilibrium with the solution (C_{pe}) is a function of air temperature, solution temperature and the concentration of solution.

The effects of solution concentration and temperature on optimized ASMR are shown in Figure 5. When the solution concentration is fixed, ASMR becomes smaller, while the solution temperature rises. This can be explained by the fact that increasing the temperature results in an improvement of saturation specific heat. When the solution temperature is fixed, the ASMR becomes higher, while the solution concentration rises. Increasing the solution concentration reduces the pressure on the surface, and saturation specific heat decreases accordingly, while solution specific heat decreases comparatively less. Thus, the designer can select an optimum ASMR using this optimization for any given desiccant temperature, concentration and air temperature.

The experiments were run under different conditions in the moderate and humid ambient of the location described previously. One of the design variables is the air mass flow rate to solution mass flow rate ratio (ASMR). Based on the results of mathematical study and optimization, this ratio has been selected under different conditions. The effects of this parameter and other variables on the outlet parameter are studied by experiment.

The effect of air mass flow rate to solution mass flow rate ratio (ASMR) on humidity effectiveness is shown in Figure 6. The humidity effectiveness decreases with increasing ASMR. This can be explained by the fact that the dehumidification rate is proportional to the desiccant flow rate. However, for an increase in air flow rate, the effectiveness decreases, because, for high air flow rate, the air will be in contact with the liquid for a shorter time, resulting in a lower change in air humidity ratio.

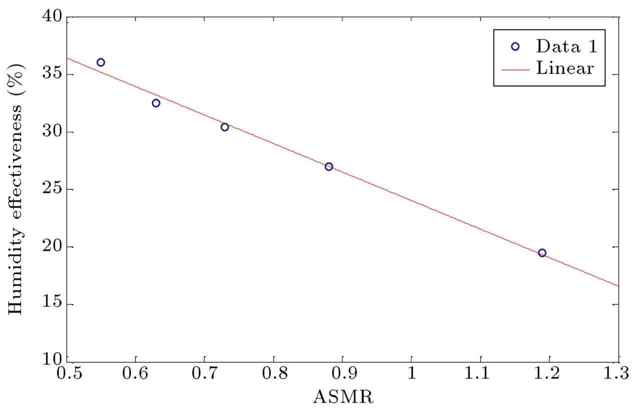

Figure 6. Effect of ASMR on humidity effectiveness.

We estimated the mass transfer coefficient, F_m, from Eq. (9) as:

$$F_m = \frac{m_{\text{cond}}}{\Delta\omega.A}. \tag{9}$$

$\Delta\omega$ is the logarithmic mean humidity difference:

$$\Delta\omega = \frac{(\omega_{a\text{in}} - \omega_{e\text{out}}) - (\omega_{a\text{out}} - \omega_{e\text{in}})}{\text{Ln}[\frac{(\omega_{a\text{in}} - \omega_{e\text{out}})}{(\omega_{a\text{out}} - \omega_{e\text{in}})}]}. \tag{10}$$

Results show that the average value of the mass transfer coefficient of the packing tower of a dehumidifier obtained from geometric dimensions reported in this experimental study is about 0.02 kg/m^{2}s. In determining the dehumidification rate, it was found that there was maximal humidity effectiveness with the optimum air to solution mass flow rate ratio.

Since, in this study, the behavior of the absorber is considered, the enthalpy efficiency in the dehumidifier tower on the air side is defined as:

$$\varepsilon_h = \frac{h_{ai} - h_{ao}}{h_{ai} - h_{s,i}}. \tag{11}$$

The effect of ASMR on the enthalpy efficiency in a dehumidifier is shown in Figure 7. In these figures, the relative humidity of air is 60-65% and temperature is 30-32°C. The influence of m^* and NTU on enthalpy efficiency is studied using Eq. (6). Maximum efficiency is nearly 0.5 for $m^* = 1$ and NTU $= 2$, however, for this experimental study (NTU $= 1.75$), efficiency is less than 40% and greater than 35%.

In this experimental study, for maximum efficiency, m^* must be equal to 1 and, according to Eq. (8), optimum ASMR must be less than 0.5. But, due to some restrictions, the ASMR is increased to 0.54, and, for this reason, enthalpy efficiency is lower than optimum efficiency.

The experiment is done for inlet air temperature, which is one of the effective parameters in dehumidification. Figure 8 shows that with an increase in inlet air temperature, dehumidification will decrease.

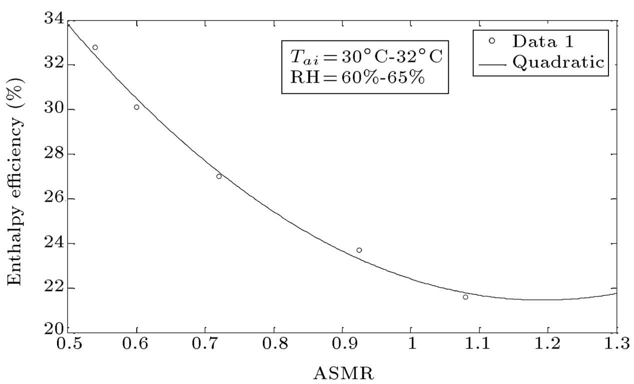

Figure 7. Influence of ASMR on enthalpy efficiency.

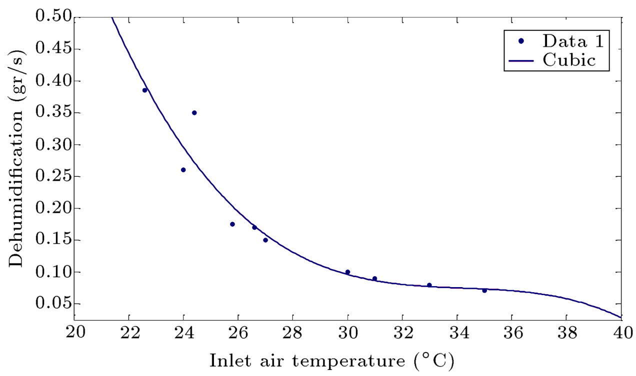

Figure 8. Influence of inlet air temperature on dehumidification.

5. Conclusion

The hybrid air conditioning system described in this paper utilizes low-grade heat energy and this system is environmentally friendly. The dehumidifier and regenerator are key components of the liquid desiccant air conditioning system. This paper presents an experimental model of the dehumidification of air with a packing tower structure using H_2O-Licl liquid desiccant.

From the results of experiments, the average mass transfer coefficient of the packing dehumidifier is 0.02 kg/(m^2 s). The performance predicted within the literature shows very good agreement with the experimental data available from this experimental study.

In this study, we found that there is an optimum air-to-desiccant ratio, which is very useful in the design of dehumidifiers and regenerators, and for maximum dehumidifier efficiency at this suitable ASMR. From analytical solution and based on enthalpy efficiency, an optimum flow rate ratio can be deduced for optimal design of both dehumidifiers and regenerators. In experiments of dehumidification, it is found that there is maximal humidity effectiveness at a suitable humidity of the ASMR.

Nomenclature

A	Area of packing m^2
h	Enthalpy (=Kj/kg)
h_c	Heat transfer coefficient (= w/ m^2-K)
h_d	Mass transfer coefficient(= kg/ m^2-s $\Delta\omega$)
L_e	Lewis number
$m^{\cdot}$	Mass flow rate(=kg/s)
NTU	Packing size
RH	Relative humidity
T	Temperature°C

L	Length of packing (m)
V	Transfer surface area per unit volume of packing (m^2/m^3)

Greek

ω_a	Air humidity ratio (kg_{vapour}/kg_{air})
ω_{ae}	Air humidity ratio at equilibrium state
ε_ω	Humidity efficiency
ε_h	Enthalpy efficiency

Subscripts

a	Air
Cond	Condensation
in	Inlet
e	Equilibrium state
out	Outlet
L	Liquid
0	Reference state
r	Return
s	Desiccant solution

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