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Model validation and case study on internally cooled/heated dehumidifier/regenerator of liquid desiccant systems

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ABSTRACT

Traditional dehumidifiers and regenerators of liquid desiccant systems often use packed columns supporting adiabatic heat and mass transfer between air and liquid desiccant. As new-style equipment, internally-cooled dehumidifiers can improve dehumidification performance due to restraining temperature increase of the desiccant. Similar to internally-cooled dehumidifiers, an idea of internally heating is imitated to put forward internally-heated regenerators. The uniform mathematical model for an internally cooled dehumidifier and internally heated regenerator was presented and validated by comparison of computation results with experimental data in this study. The case study focused on the parameters distribution comparisons of the internally cooled/heated dehumidifier/regenerator with adiabatic ones and demonstrated coupled heat and mass transfer behavior. The results show that the internally-heated regenerator can produce higher regeneration efficiency than the adiabatic one to produce better energy efficiency and eliminate the dehumidification possibility which would happen in adiabatic regenerators. The internally-cooled dehumidifier can also provide better dehumidification performance comparing with the adiabatic one; however its benefit would be not as good as the internally-heated regenerator. In addition, effect of the width of the air channel on internally cooled/heated dehumidifier/regenerator was discussed and the results can help the optimal design of this kind of dehumidifiers and regenerators.

1. Introduction

Liquid desiccant system can utilize low grade heat such as solar energy [1] to bring remarkable potential for saving energy [2] and improving indoor air quality. Many new air conditioning systems using liquid desiccant have been proposed as alternatives to the traditional cooling systems [3]. Dehumidifier and regenerator are the most important components, since they determine the efficiency and thermal performance of liquid desiccant systems. In the dehumidifier, air is contacted with the concentrated desiccant solution and dehumidified; in the regenerator, the diluted and heated desiccant solution is directly contacted with ambient air, which can take out the water from the desiccant solution.

In the past decades, the packed dehumidifiers and regenerators were highly attractive since packed columns can provide long residence time and much contacting area with simple structure and low cost. Coupled heat and mass transfer between liquid desiccants and air happen simultaneously and adiabatically in the packing. Many researchers (Grossman [4], Gandhidasan [5], Stevens [6], Goswami [7], Chung [8], Liu [9], Yin [10,11], Ren [12]) have conducted some experimental and theoretical studies on this kind of adiabatic dehumidifiers and regenerators. All results indicated that in the packed dehumidifiers and regenerators heat and mass transfer wears away with the progress of processes due to the change of local temperature and concentration gradients, which restricts the performance of them. Internally-cooled dehumidifiers like coil-type dehumidifiers are different from the adiabatic dehumidifiers using packing, and in the former ones the release latent generated by dehumidification can be removed by the coolant to keep higher mass transfer gradients. Khan [13] presented the performance analysis of an internally cooled absorber using thin plate heat and mass exchanger cooled by direct evaporation with lithium chloride as desiccant. Enthalpy effectiveness and humidity effectiveness were brought out to define the thermal performance of the absorber. It was found that the number of mass transfer units had great effect on the enthalpy and humidity effectiveness. Khan [14] investigated numerically on an internally cooled dehumidifier using tube-fin exchanger with the air crossing flow using one-dimensional definite difference model. Jain et al. [17] conducted experimental study on wetted wall column dehumidifier with lithium bromide as desiccant. It was found that the improper wetness had important effect on the performance of dehumidifier and regenerator and gave the wetness factor. Theoretical model predicted the experimental data with ±30%. Saman and Alizadeh [18] presented the experimental results using cross-flow type plate heat exchanger (PHE) as a dehumidifier/cooler. Many flow passages were separated by

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Nomenclature

Cp	specific heat	kJ/(kg °C)
Comp.	computation result	
D	diffusion coefficient	m^2/s
d	width of air channel	m
E	error between simulation and measurement data	
Exp.	experiment data	
G_a	flow rate of air	m^3/s
G_s	flow rate of desiccant	kg/s
H	height	m
h_a	enthalpy of air	kJ/kg
h_C	heat transfer coefficient between air and desiccant	$W/(m^2\,°C)$
h_D	mass transfer coefficient based on air humidity ratio difference	$kg/(m^2\,s)$
h_s	enthalpy of desiccant solution	kJ/kg
h_w	heat transfer coefficient between water and desiccant	$W/(m^2\,°C)$
$h_{v,s}$	enthalpy of moisture air on solution surface	kJ/kg
m_s	desiccant flow rate in control volume	kg/s
M	measurement result	
k	coefficient in correlations, dimensionless	
Le	Lewis number, dimensionless	
NTU	Transfer Unit Number, dimensionless	
Q_h	heat supplied	kW
q_v	latent heat of vaporization	kJ/kg
Re	Reynolds number, dimensionless	
S	simulation result	
Sc	Schmidt number, dimensionless	
Sh	Sherwood number, dimensionless	
T	temperature	°C
U_a	air velocity	m/s
U_E	uncertainty in comparison	
U_M	uncertainty of measurement data	
Us	uncertainty of simulation results	
W	width	m
x_i	input parameters of the model	
X	desiccant concentration	%
z	height of control volume	m

Greek letters

β	wetness coefficient, dimensionless	
η	efficiency, dimensionless	
ω	humidity ratio of the air	kg/kg
ρ	density	kg/m^3

Subscripts

a	air
in	inlet
out	outlet
reg	regeneration
s	desiccant solution
Ts, sat	saturation status under temperature of T
v	vapor
v, s	vapor on solution surface
w	cooling water

thin plastic plates, and one side of each thin plastic plate was for the desiccant solution/air passage; the other side is for water/air passage. So the indirect evaporation process in the water/air passage provided cooling to the dehumidification process. The experimental results indicated the effects of inlet parameters of air and solution on the PHE effectiveness. Considering the fact that there were different velocities and properties in air layer and desiccant falling film in the cross-section normal to the flow, some investigators [19–22] considered the coupled heat and mass transfer as two-dimensional cases using Navier–Stokes equations, but for solution of the equations, very complex computations have to be taken besides assumptions about boundary conditions, so simplified models need developing and validating to probe into the heat and mass transfer behavior between air and desiccant in dehumidifiers and regenerators.

Similar to internally-cooled dehumidifiers, an idea of internally heating is imitated to put forward internally-heated regenerators. Till now, little work has been conducted on internally-heated regeneration processes. In addition, simultaneous heat and mass transfer between liquid desiccant and air is very complex, and it need further analysis on the coupled heat and mass transfer processes of internally cooled/heated dehumidifier/regenerator. This paper will put forward the idea of internal heating source based on internally cooling, and investigate simultaneous heat and mass transfer behavior of an internally-cooled dehumidifier and internally-heated regenerator using a mathematical model validated by experimental data.

2. Methodology

2.1. Mathematical model

In adiabatic dehumidifiers, temperature of liquid desiccant and air would rise because of the release of vaporization heat, and mass transfer driving force–vapor partial pressure difference between liquid desiccant and air reduces; as a result, dehumidification performance would decay with the progress of the process in dehumidifiers. Similarly in regeneration processes of desiccant solution, temperature of liquid desiccant goes down in adiabatic regenerators because it is necessary for desiccant solution to provide latent heat of vaporization to transfer water in the desiccant solution to air. Hence, the regeneration performance would become weaker and weaker with the process going in regenerators. In order to keep more driving force in dehumidifiers and regenerators, heated or cooled sources are proposed to import to the regeneration and dehumidification processes. Here plate heat exchanger is used for the internally cooled/heated dehumidifier/regenerator. Fig. 1(a) shows the schematic of internally cooled/heated dehumidifier/regenerator. Air and desiccant enter the plate exchanger from the top, and desiccant solution ($LiCl–H_2O$) is sprayed to the walls by distributing device and flow down along walls to the bottom by its gravity; there air and desiccant solution contact directly to conduct the heat and mass transfer. Fig. 1(b) shows the fluid flow configuration in a unit. In each unit there are two channels in the dehumidifier/regenerator with 1 m height; one (channel II) is for the liquid desiccant and air with the width of 12 mm, and the other (channel I) is for coolant or heating fluid with the width of 2.5 mm. When dehumidification process is necessary, the coolant enters into channel I. While in regeneration processes, the heating fluid enters into the channel I.

Along the height of the dehumidifier and regenerator, finite segments are divided to generate control volumes for describing the heat and mass transfer processes, and the control volume is shown in Fig. 1(c). To demonstrate the simultaneous heat and mass transfer for the processes, following assumptions are made in the derivation of the steady-state governing equations for the dehumidifier and regenerator:

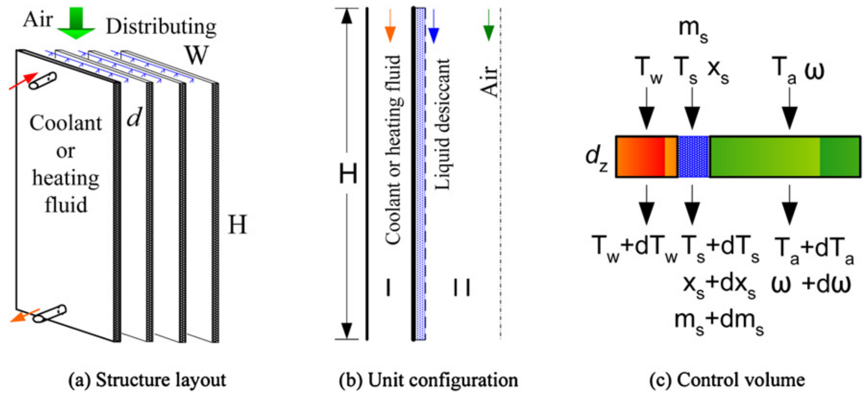


Fig. 1. Internally cooled/heated dehumidifier/regenerator.

(1) Due to very small width of channels and desiccant, air and water well mixing in the cross-section normal to their flow, gradients for each fluid only exist in their respective flow directions;
(2) Considering the width of the channel is very small, the properties of water, desiccant solution and air are considered uniform within the control volume;
(3) The Lewis number keeps constant under a set of definite inlet parameters;
(4) Local wall temperature of the heat exchanger is equal to the local water temperature.

Jain et al. [17] showed that only a portion of total available area was actually wetted and available for heat and mass transfer. So the wetness efficiency should be concerned in description of the simultaneous heat and mass transfer. Mass transfer happens only at wetted area under the vapor partial pressure difference between the liquid desiccant surface and air:

$$\frac{1}{2}G_a \cdot d\omega_a = h_D \cdot (\omega_{T_s,\mathrm{sat}} - \omega_a) \cdot W \cdot \beta\, dz \tag{1}$$

Here,

$$G_a = d \cdot W \cdot U_a \tag{2}$$

Heat transfer conducts both wetted area and non-wetted area under the temperature difference between air and desiccant or walls:

$$\frac{1}{2}G_a \cdot Cp_a \cdot dT_a = h_C(T_s - T_a)W\beta\, dz + h_w(T_w - T_a)W(1-\beta)\, dz \tag{3}$$

The enthalpy of the air is changed resulting from the change of temperature and humidity, and therefore the total heat exchange amount of the air is defined as:

$$dh_a = Cp_a \cdot dT_a + (q_v + Cp_v \cdot T_a)\, d\omega_a \tag{4}$$

Substitute Eqs. (1) and (3) into (4) to yield:

$$dh_a = \big[\beta h_C \cdot (T_s - T_a) + h_w(1-\beta)(T_w - T_a) + \beta h_D \cdot (\omega_{T_s,\mathrm{sat}} - \omega_a) \cdot (q_v + Cp_v \cdot T_a)\big]\frac{2W\, dz}{G_a} \tag{5}$$

The Lewis number and NTU are defined as:

$$Le = \frac{h_C}{h_D \cdot Cp_a} \tag{6}$$

$$NTU = \frac{h_D \cdot W \cdot H\beta}{G_a/2} \tag{7}$$

Substitute Eqs. (6) and (7) into (5) to yield:

$$dh_a = NTU \cdot \left[Le(h_{v,s} - h_a) + (Le-1)(q_v + Cp_v \cdot T_a)(\omega_{Ts,\mathrm{sat}} - \omega_a) + \frac{h_w(1-\beta)(T_w - T_a)}{\beta h_D}\right]\frac{dz}{H} \tag{8}$$

The enthalpy of liquid desiccant is changed because of heat exchange with both the air and the heating or coolant fluid, defined as:

$$G_w \cdot Cp_w \cdot dT_w + d(G_s \cdot h_s) + G_a \cdot dh_a = 0 \tag{9}$$

Here the enthalpy change of the desiccant solution is defined as:

$$dh_s = Cp_s \cdot dT_s \tag{10}$$

Combining Eqs. (9) and (10), yield to:

$$dT_s = -\frac{1}{Cp_s}\left(\frac{G_a}{G_s} \cdot dh_a + \frac{G_w}{G_s}Cp_w \cdot dT_w + \frac{G_a}{G_s}Cp_s \cdot T_s \cdot d\omega_a\right) \tag{11}$$

Considering the energy balance of control volume, the temperature of heating water or coolant can be defined as:

$$\frac{dT_w}{dz} = h_w \cdot W\frac{\beta(T_s - T_w) + (1-\beta)(T_a - T_w)}{G_w \cdot Cp_w} \tag{12}$$

According to the mass balance between the desiccant solution and the air, the concentration change of the desiccant solution control volume can be calculated as:

$$dX_s = -\frac{dG_s}{G_s + dG_s}X_s \tag{13}$$

Here,

$$dG_s = -G_a\, d\omega_a \tag{14}$$

Eqs. (1) and (2) are the governing equations of heat and mass transfer processes of air, which is used to solve the temperature and humidity ratio of air in control volumes. Eq. (10) can infer the temperature change of solution in control volumes. Temperature change of cooling or heating fluid is determined by Eq. (11). After calculating air parameters, flow rate and concentration change of desiccant solution are computed by Eqs. (12) and (13). During above computation, the Lewis number is usually assumed as constant, however the NTU number is greatly depended on the mass transfer coefficient between air and desiccant solution. Therefore, experimental data is necessary to determine the mass transfer coefficient between air and desiccant solution and to validate the model.

2.2. Empirical correlations and model validation

Since many assumptions and parameter estimation have to be used to the model, it is necessary to validate the model by comparison of the computation results with the experimental data. Here the model accuracy is verified by comparison of computation outlet parameters of air and desiccant with the experimental outlet measurement data. Experimental data and more detailed information about the setup can be referred to the paper by Yin et al. [24].

Parameters of air and liquid desiccant have important effect on the heat and mass transfer in internally cooled/heated dehumidifiers/regenerators. Here we concentrated our attention to two main parameters of them, which are desiccant temperature and air velocity. Based on this consideration, some experiments were carried out. The experimental parameters are shown in Table 1. The mass transfer coefficient can be expressed by the *Sh* dimensionless number. Based the experimental data, the nonlinear regression correlation is drawn as follows in dehumidification and regeneration.

In dehumidification:

$$Sh = 4.513 \times 10^{-3} k \cdot Re^{1.56} Sc^{0.33} \tag{15}$$

$$\text{Here} \quad k = 76.456 T_s^{-2.991} \tag{16}$$

In regeneration:

$$Sh = 4.6767 k \cdot Re^{1.55} Sc^{0.33} \tag{17}$$

$$\text{Here} \quad k = 5.52 \times 10^4 T_s^{-3.36} \tag{18}$$

The mass transfer coefficient can be calculated by:

$$h_D = \frac{D \cdot Sh}{d} \tag{19}$$

Here d is the width of air channel, the value of which in experimental is 0.012 m.

To determine the validation of the model, a comparison error, E, is defined with respect to the measurement data, M and the simulation result S. And the uncertainty U_E in the comparison error is expressed as the follows proposed by Coleman and Stern [15].

$$E = M - S \tag{20}$$

$$U_E = \sqrt{U_M^2 + U_S^2} \tag{21}$$

Here U_M is the uncertainty in the measurement and U_S is the uncertainty in the simulation. The measurement uncertainties of different parameters are shown in Table 1. The simulation model uncertainty U_S is determined by the uncertainty propagation equation proposed by Taylor [16].

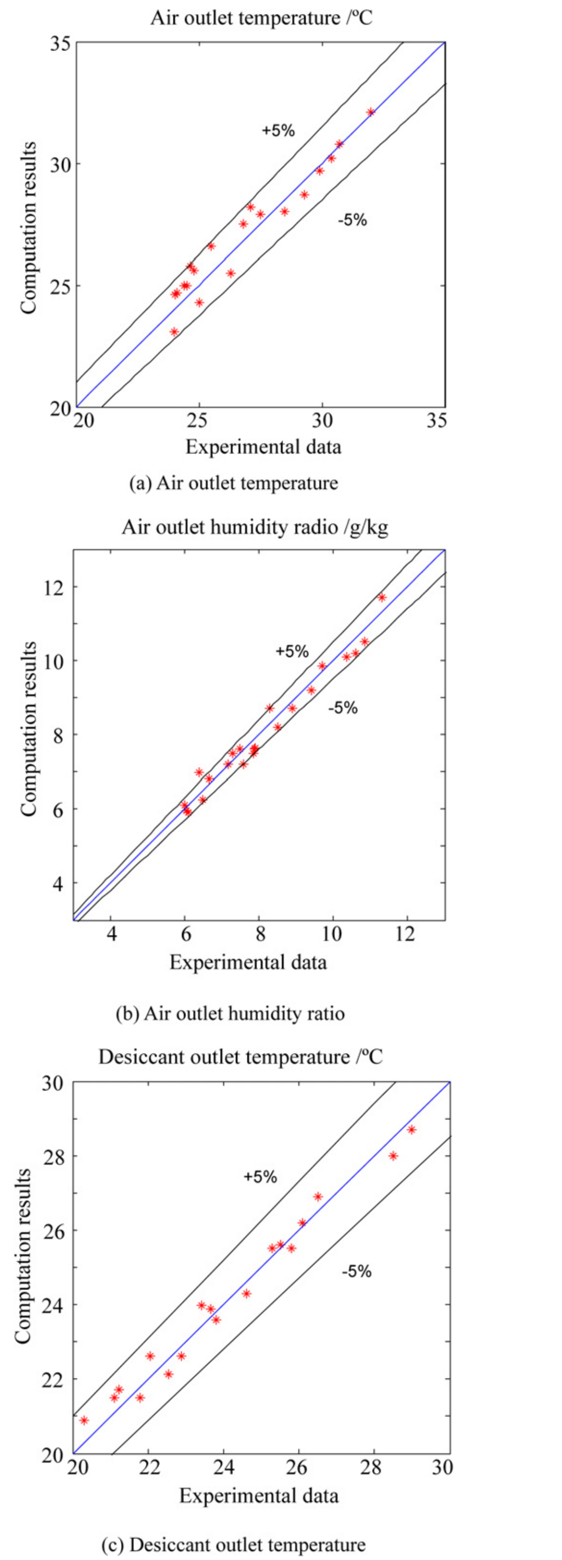
$$U_S = \sqrt{\left(\frac{\partial U_S}{\partial x_1}\delta x_1\right)^2 + \left(\frac{\partial U_S}{\partial x_2}\delta x_2\right)^2 + \cdots + \left(\frac{\partial U_S}{\partial x_n}\delta x_n\right)^2} \tag{22}$$

Here x_i term is the input parameter to the model. The validation process is a comparison of E and U_E. If absolute value of E is less than U_E, then the combination of all the errors of measurement and simulation is smaller than the estimated validation uncertainty and the validation is achieved, the level of validation of the model is U_E, otherwise, the proposed model needs some improvement. Based on Eq. (22) and Table 1, calculation results showed that the uncertainty of the model for predicted outlet temperature of air, desiccant and outlet air humidity were respectively 4.83%, 6.59% and 8.32%.

Fig. 2 displays the comparison results of outlet air temperature, humidity ratio and desiccant temperature under different operation conditions for internally-cooled dehumidification processes.

Table 1
Inlet parameters and uncertainties of air and desiccant solution in experiments.

	T_a (°C)	U_a (m/s)	ω_a (g/kg)	T_s (°C)	X_s (%)	G_s (kg/s)
Dehumidification	26–32	1.5–4	9.5–14.5	20–28	36–40	0.12
Regeneration	21–30	1.5–4	6–12	55–78	32–39	0.1–0.12
Uncertainty	0.3	2%	5%	0.3	0.6	5%



(a) Air outlet temperature

(b) Air outlet humidity ratio

(c) Desiccant outlet temperature

Fig. 2. Comparison of computation results with experimental data for dehumidification.

Table 2
Comparison of the computation results with experimental data for regeneration.

	$G_{a,\text{in}}$ (m^3/s)	$T_{a,\text{in}}$ (°C)	$\omega_{a,\text{in}}$ (g/kg)	$T_{w,\text{in}}$ (°C)	$G_{s,\text{in}}$ (kg/s)	$T_{s,\text{in}}$ (°C)	$X_{s,\text{in}}$ (%)	Le	$\omega_{a,\text{out}}$		$T_{a,\text{out}}$		$T_{s,\text{out}}$	
									Exp.	Comp.	Exp.	Comp.	Exp.	Comp.
1	0.062	26.5	11.4	70.5	0.1	70.0	37.5	0.3	24.5	24.8	37.1	37.3	66.3	65.4
2	0.067	26.5	11.5	70.5	0.1	70.8	38.0	0.35	23.9	23.9	38.9	38.8	64.8	65.2
3	0.076	26.0	11.7	70.5	0.1	71.0	38.0	0.35	24.0	23.7	39.8	39.5	64.9	64.1
4	0.051	21.0	6.3	65.0	0.12	64.5	32.2	0.39	20.8	21.1	29.4	29.4	58.6	59.1
5	0.050	21.7	6.3	74.0	0.12	74.3	33.6	0.45	28.0	28.4	34.1	34.4	67.7	67.8
6	0.050	22.4	6.3	78.0	0.12	77.1	34.8	0.5	27.3	28.2	35.3	35.6	73.1	71.4
7	0.050	22.9	6.7	76.0	0.12	74.0	36.2	0.5	25.5	25.3	36.8	36.8	70.3	69.3
8	0.050	23.1	6.6	76.0	0.12	74.5	37.3	0.5	24.3	23.7	37.1	37.0	69.9	69.9
9	0.050	25.1	7.2	78.0	0.12	75.1	38.3	0.5	23.5	22.4	37.7	37.2	70.3	70.3
10	0.050	26.6	7.2	78.0	0.12	73.0	39.2	0.5	21.4	21.6	38.4	38.7	69.9	70

The comparison results show the difference of 5% between predicted value and experimental data, which are very close or less than the uncertainties of the model for predicted results. The accuracy of the model is under influence of many factors, such as measure error of air and desiccant parameters, approximation of parameters in the model, of which the Lewis number and efficient contact area between air and desiccant are two important parameters. It is found that Lewis number was changed between 0.7 and 1.5 under the experimental operation conditions, and also we found the same concern described in the paper by Jain et al. [17], only part of the total available area was wetted by desiccant solution and available for heat and mass transfer. Here the wet coefficient used to describe the efficient contact area was around 0.7 during the validation, which is greatly depended on flow rate of liquid desiccant and air.

Table 2 shows the comparison of the computation results of outlet parameters with experimental data for internally-heated regenerations. The outlet humidity ratio and temperature of air and desiccant temperature are mainly concerned. Comparing the computation results and experimental data, it is found that the differences between them are acceptable and the Lewis number changed between 0.3 and 0.5 under experimental conditions. The validations indicate that the model can give reasonable prediction for the heat and mass transfer between air and liquid desiccant and in the following section a case study conducted based on the validated model.

3. Case study and discussion

Here the main objective is to conduct a case study to compare the internally cooled/heated dehumidifier/regenerator with the adiabatic ones, and obtain the distribution characteristic of the parameters of the liquid desiccant and air along the height of the dehumidifier/regenerator, and to analyze the heat and mass transfer processes. The physical model of the dehumidifier/regenerator is shown in Fig. 1, and the dehumidifier/regenerator is adiabatic if the coolant/heating fluid does not work, otherwise the dehumidifier/regenerator is internally cooled/heated if the coolant/heated fluid works. The operation conditions in per channel unit are shown in Table 3. The coolant/heating fluid is water. The desiccant is aqueous lithium chloride ($LiCl–H_2O$). The properties of the desiccant are referred to the paper by Conde [23]. It was supposed that wetness efficiency was 0.8 during the simulation and the Lewis number was 0.5 in the regeneration and 1.2 in the dehumidification according to the test results.

Table 3
Inlet parameters of air and desiccant for the case study.

	$G_{a,\text{in}}$ (m^3/s)	$T_{a,\text{in}}$ (°C)	$\omega_{a,\text{in}}$ (g/kg)	$G_{s,\text{in}}$ (kg/s)	$T_{s,\text{in}}$ (°C)	$X_{s,\text{in}}$ (%)	G_w (kg/s)	$T_{w,\text{in}}$ (°C)
Dehumidification	0.0131	28	11.5	0.008	26	39	0.04	24
Regeneration	0.0144	29	11.5	0.008	60	37.5	0.04	65

Fig. 3 demonstrates the comparison of the solution, air and water parameters in both internally heated and adiabatic regenerators. Fig. 3(a) indicates the distributions of temperature of the solution and water along the height of the regenerators. The temperature of solution in the internally-heated regenerator is obviously higher than that in the adiabatic one due to the attendance of the hot water, and therefore the heating water temperature decreases from 65 to 61 °C gradually. From the figure, it is found that in the internally-heated regenerator the temperature of the desiccant solution decreases from 60 to 45.2 °C firstly and then increases. The reason is that at the beginning the more mass transfer would take place between solution and air, and furthermore the air temperature is very low at the entrance; it means that desiccant solution would expend much heat for that. With the progress of the process, the desiccant temperature would increase due to the domination of the internally hot water.

Fig. 3(b) displays the distribution of the desiccant concentration in both regenerators. The concentration in the internally-heated regenerator increases from 37.5% to 38.3% along the height, but in the beginning increases and then decreases in the adiabatic regenerator, which means the dehumidification has happened in it and air is dehumidified, shown in Fig. 3(d). The reason is that the desiccant temperature decreases sharply shown in Fig. 3(a) so that the vapor partial pressure of desiccant solution surface is less than that of the air, which means that no enough heat is supplied for the whole regeneration process, however the internally hot water can restrain it to give the good regeneration performance. Thus, it is very important to choose reasonable regenerator physical model and operation parameters.

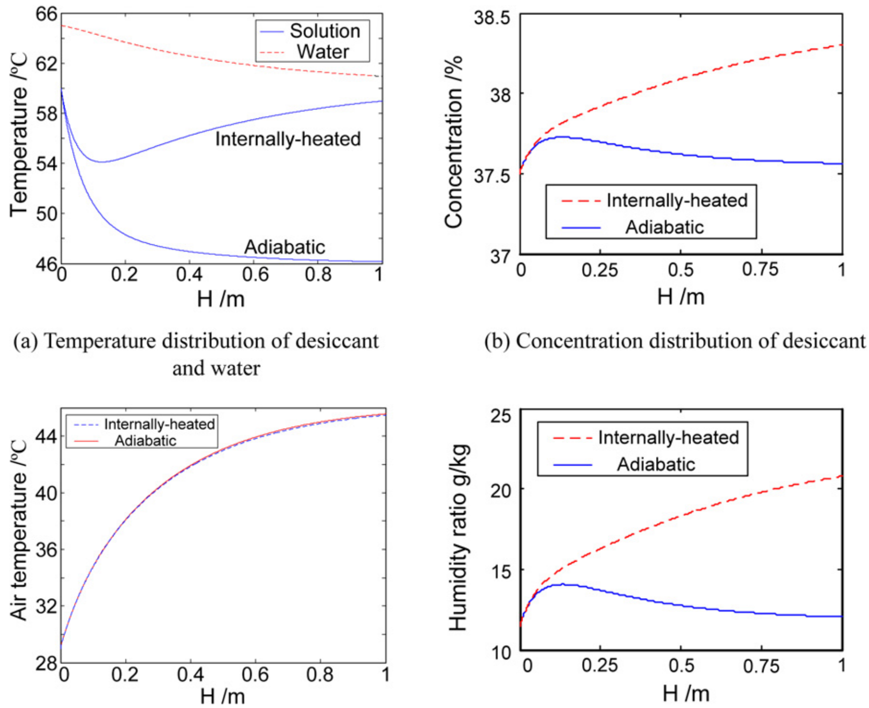
Fig. 3(c) shows the temperature distribution of air in both regenerators. From the figure it seems there is almost same air distribution in both regeneration processes. In our experiment data we also found the result, which is almost same outlet air temperature was obtained in both regenerators, and the detailed experimental results could be referred to the paper by Yin et al. [24]. To discuss energy efficiency of the two regeneration models, thermal efficiency of regeneration is often defined as follows:

$$\eta_{\text{reg}} = \frac{\Delta\omega_a \cdot q_v}{Q_h} \tag{23}$$

In the above equation, Q_h is heat power supplied for the regeneration. In desiccant regeneration process, suppose that the desiccant from the regenerator is heated directly and circulated in the regenerator, and therefore for the regeneration unit the energy consumption is used to increase the enthalpy of the air, i.e.

$$\eta_{\text{reg}} = \frac{\Delta\omega_a \cdot q_v}{G_a \rho_a (Cp_a \Delta t_a + \Delta\omega_a \cdot q_v)} \tag{24}$$

According to Fig. 3(c), we can suppose the change of air temperature is same in both regenerators. But the change of air humidity



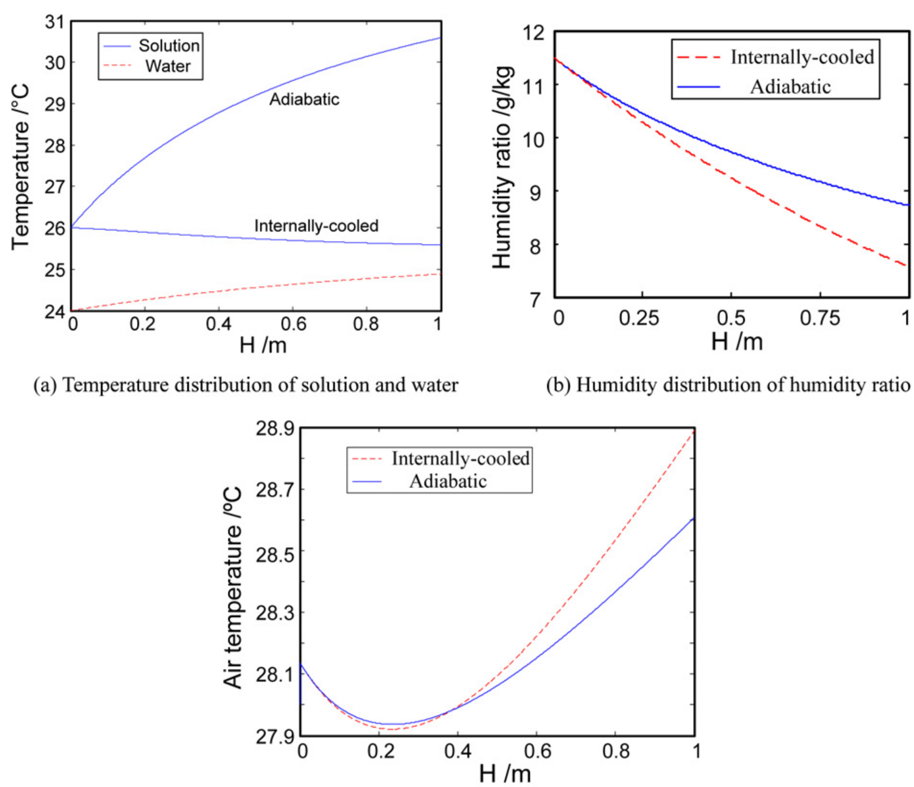
(a) Temperature distribution of desiccant and water

(b) Concentration distribution of desiccant

(c) Air temperature distribution of air

(d) Humidity ratio distribution of air

Fig. 3. Comparison of parameters distribution of internally-heated regenerator with adiabatic one.



(a) Temperature distribution of solution and water

(b) Humidity distribution of humidity ratio

(c) Temperature distribution of air

Fig. 4. Comparison of parameters distribution of internally-cooled dehumidifier with adiabatic one.

ratio in the internally-heated regenerator is much more than that in the adiabatic one shown in Fig. 3(d). Combining Eq. (20), it is concluded that the regeneration efficiency of the internally-heated regenerator is higher than that of the adiabatic one.

Fig. 4 shows the comparison of parameters of the solution and air in the adiabatic and internally cooled dehumidifiers. Fig. 4(a) displays that temperature distribution of the desiccant and water in both dehumidifiers. In the adiabatic dehumidifier, the desiccant

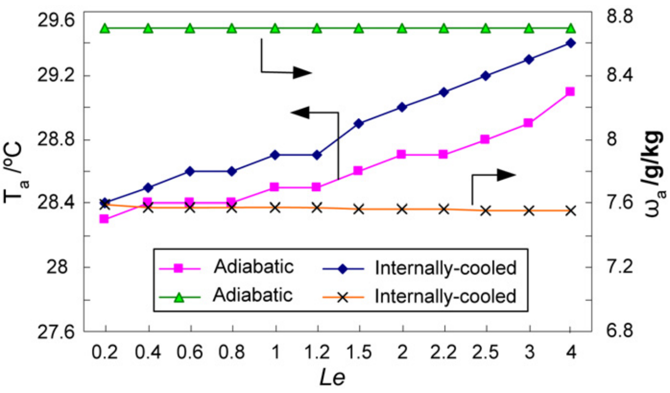


Fig. 5. Effect of the Lewis number on the outlet parameter of dehumidification.

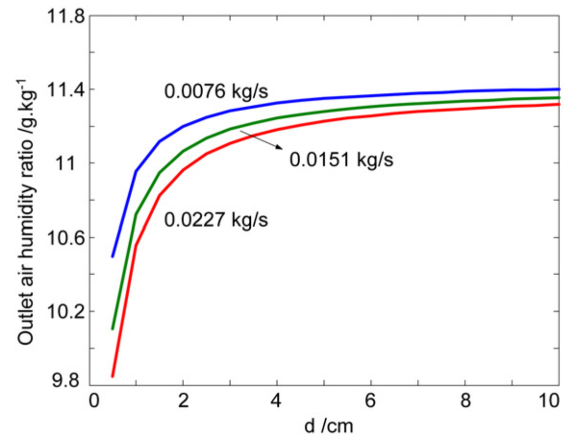


Fig. 6. Effect of the channel II width on the dehumidification with different air flow rates.

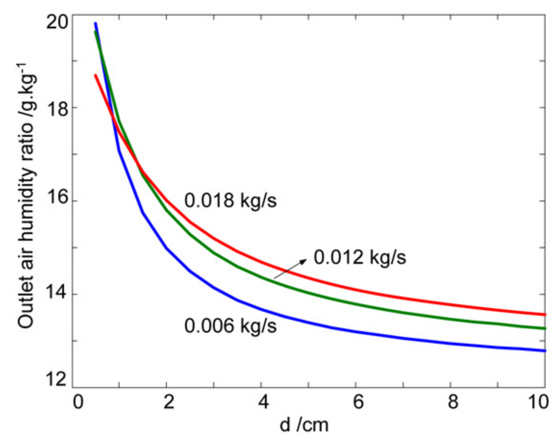


Fig. 7. Effect of the channel II width on the regeneration with different air flow rates.

temperature increases from 26.0 to 30.6 °C gradually because of the release of vapor latent heat during dehumidification process, which would make mass transfer between air and desiccant decay, however internally-cooled dehumidifier can remove the latent heat by internally coolant to keep lower temperature of desiccant, which manage to give less outlet humidity ratio of the air, just as shown in Fig. 4(b) humidity ratio of the dehumidified air is less 1 g/kg than that in the adiabatic one, anyway the difference is not as obvious as that in the regenerators.

Fig. 4(c) depicts the comparison of temperature distributions of air in both dehumidifiers. In this case, the figure shows that the air temperature decreases at the beginning and then increases in both dehumidifiers, which is because the temperature of the air is lower than the desiccant at the beginning, and then with the progress of the dehumidification, the latent heat release dominates the process so that the air temperature increases. From the figure, it can be seen that the outlet temperature of the air in the internally-cooled dehumidifier is higher than that in the adiabatic one, although the difference between them is very small, only 0.3 °C, the reason for which is that in the internally-cooled dehumidifier more mass transfer causes more latent heat release from the dehumidification, and as a result the air would gain more heat to be with higher outlet temperature. The outlet temperature difference between both dehumidifiers greatly depends on the Lewis number and operation conditions. During our experiments, the same fact happened under some operation conditions, which was that the outlet air temperature in the internally cooled dehumidifier was slightly higher than that in the adiabatic one. In above computation for the dehumidification, the Lewis number was assumed as 1.5. In order to check the behavior of effect of Lewis number on the process, Fig. 5 demonstrates outlet temperature and humidity ratio of the air at different Lewis numbers in both dehumidifiers. The results indicate that the Lewis number has little influence on outlet humidity ratio of the air, and however has a little influence on outlet temperature of air since the outlet temperature of air changes less than 1 °C with the obvious change of Lewis number. The results also show that the outlet temperature of the air in the internally-cooled dehumidifier is slightly higher than that of the adiabatic one.

Usually the internally cooled/heated dehumidifier/regeneration expected is with suitable dimensions and good thermal performance. The width of the heat and mass transfer channel is an important parameter for optimal design of this kind of dehumidifier. Fig. 6 indicates the effects of the width of channel between air and desiccant with three different air flow rates (0.0076, 0.0151 and 0.0227 kg/s) under the typical operation conditions shown in Table 3. In this case, for better dehumidification performance it is recommended that the width of the channel should be less than 2 cm. The results show that the dehumidification performance goes down with increase of the channel width, which is the result from that smaller width of the channel II can provide higher air velocity and mass transfer coefficient. However, too small width of the channel would bring another problem, which is too high pressure resistance in the channel and more energy consumption of pumps. Fig. 7 shows the effects of the air channel width on the regeneration with three different air flow rates (0.006, 0.012 and 0.018 kg/s) under the operation conditions shown in Table 3. The results show that higher air flow rate can produce better regeneration performance, for better regeneration performance it is recommended that the width of the air channel should be less than 1.5 cm, however pressure resistance problem should also be considered here as in the optimal design of the dehumidifier. Hence, the designs of the internally cooled/heated dehumidifier/regenerator need to consider roundly the dimension size, thermal performance and air pressure resistance.

4. Conclusions

The idea of internal heat/sink source was put forward in order to keep high temperature of desiccant solution in the regenerator and lower temperature of the desiccant solution in the dehumidifier for better mass transfer performance compared with traditional packing, and a unitized model for the internally-cooled dehumidifier and internally-heated regenerator was presented. By utilizing experimental data, the correlations of mass transfer coefficient between air and desiccant were developed for the dehumidification and regeneration, which indicated the effect of temperature and flow rate of desiccant on mass transfer coefficients. In addition, model validation was carried out by comparing the computation results with experimental data. It was found that the errors were within 5% and showed acceptable accuracy. Whereas further study is still necessary to develop more comprehensive mass transfer coefficient correlations under wider range experimental conditions and deal more accurately with the model assumptions, such as actual Lewis number, definite description of wetness. Anyhow, the present study highlights the importance of actual Lewis number and wetness, which should be paid more attention in future study.

A case of the internally cooled/heated dehumidifier/regenerator using the plate exchanger with the parallel flow was studied numerically to discuss the heat and mass transfer behavior between air and desiccant. The results show that the internally-heated regenerator can avoid the dehumidification possibility which would happen in adiabatic one, and in addition it could offer higher regeneration efficiency than the adiabatic one to produce better energy efficiency. The internally-cooled dehumidifier can also provide better dehumidification performance comparing with the adiabatic one; however its benefit would be not as good as the internally-heated regenerator. Effect of the air channel width on the dehumidifier and regenerator was discussed and reasonable width is recommended to optimize the design of this kind of dehumidifier and regenerator.

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