https://doi.org/10.1016/j.ijrefrig.2016.06.005

An experimental study on the dehumidification performance of a counter flow liquid desiccant dehumidifier

Lingshi Wang[1], Fu Xiao[1,*], Xuejun Zhang[2], Ritunesh Kumar[3]

[1]Department of Building Services Engineering, The Hong Kong Polytechnic University, Kowloon, Hong Kong

[2]Institute of Refrigeration and Cryogenics, Zhejiang University, Hangzhou 310027, China

[3]Mechanical Engineering Department, Indian Institute of Technology, Indore, MP, India-453446

*****Corresponding author. Tel.: +852 2766 4194; Fax: +852 2765 7198.

E-mail address: linda.xiao@polyu.edu.hk

ABSTRACT

This paper presents an experimental study on the dehumidification performance of a counter flow liquid desiccant dehumidifier using structured packing with a high specific surface area (650m^2/m^3). New empirical equations correlating the moisture effectiveness and the enthalpy effectiveness with critical inlet parameters are developed, which can be used to conveniently predict the performance of a similar dehumidifier. The empirical correlations are validated using the experimental data of this study, and compared with the experimental data reported by another researcher. The deviations are within ±10% for the former and within ±15% for the latter. The performance of the present type of packing is also compared with other two types of structured packing available in literature. The influences of the inlet conditions of the air and the desiccant as well as the packing height on the dehumidification performance are also investigated and compared with the results reported in previous studies.

Keywords: Liquid desiccant; Counter flow; Dehumidification; Experiment; Effectiveness; Empirical correlation

Nomenclature

AAD	average absolute difference (%)	ε_h	enthalpy effectiveness
G	mass flow flux of air (kg/m^{2}s)	ε_m	moisture effectiveness
h	specific enthalpy (kJ/kg)	ω	humidity ratio (g/kg)
H	height of packing (m)		
L	mass flow flux of desiccant (kg/m^{2}s)	*Subscripts*	
m	mass flow rate (kg/s)	a	air
m_{de}	moisture removal rate (g/s)	cal	calculated value
P	partial vapor pressure of air (Pa)	e	equilibrium state
Q	enthalpy variation from inlet to outlet of dehumidifier (kW)	exp	experimental value
T	temperature (^{o}C)	in	inlet
		out	outlet
Greek symbols		s	solution
ξ	solution mass concentration (-)		

1. Introduction

Liquid desiccant based air conditioning systems have been attracting more and more attentions in recent years owing to their merits in effective and energy-efficient air dehumidification for indoor environment control (Dai et al., 2001; Ge et al., 2011; Wang et al., 2009; Xiao et al., 2011). The dehumidifier is the key component of the liquid desiccant based air conditioning systems. To predict system performance, optimize the design and operation parameters, and develop control and operation strategies for the hybrid systems, reliable mathematical models of dehumidifiers are indispensable (Ge et al., 2011; Xiao et al., 2011; Zhang et al., 2012). A number of theoretical models and empirical models of different types of dehumidifier were developed (Fumo and Goswami, 2002; Lazzarin et al., 1999; Wang et al., 2013). Experimental research on the dehumidification processes is valuable for validating and improving those models. Meanwhile, some crucial parameters of the models, such as the dehumidification effectiveness (Ge et al., 2011) and the mass transfer coefficient (Wang et al., 2013; Zhang et al., 2012), are usually determined from experimental data due to the complicated coupled heat and mass transfer occurring in a dehumidification process. Experimental study on the liquid desiccant dehumidifier is also beneficial to clearly understanding and enhancing the coupled heat and mass transfer.

Dehumidifiers using packed towers with random packing or structured packing are popular owing to larger contact areas. Oberg and Goswami (1998) experimentally studied the performance of a counter flow dehumidifier using random packing. The influences of various inlet conditions on the moisture removal rate and the moisture effectiveness were assessed. Similar research on a counter flow dehumidifier and a regenerator was conducted by Fumo and Goswami (2002) using lithium chloride aqueous solution as the liquid desiccant. The pressure drop on the air side in the random packing is a big concern. Longo and Gasparella (2009) showed that the structured packing can significantly reduce 65%-75% of the air pressure drop. Besides, the structured packing is easy to be installed when compared with the random packing. Hence, the structured packing has been widely used in various dehumidifiers in recent years. Chung et al. (1996) developed the dimensionless empirical correlations of heat and mass transfer coefficients for both random and structured packings, the deviation was found to be less than ±10% between predicted values by the correlations and experimental data. Yin et al. (2007) developed the empirical correlations of the mass transfer coefficient of the regeneration process and also found there is a maximum efficiency of dehumidification at a certain air inlet humidity ratio. A hybrid model which can be conveniently used for control and optimization of a packed-type dehumidifier was proposed by Wang et al. (2013). The parameters in the correlation equations of the heat and mass transfer coefficients were determined from experiment data. The empirical correlations for mass transfer coefficients and the dehumidification effectiveness of packed-type dehumidifiers from the literature were summarized and analysed by Jain and Bansal (2007).

The popular materials used in the structured packing include plastic, ceramic, metal, and wood fibre (Moon et al., 2009). Zhang et al. (2010) investigated the mass transfer characteristics of a cross flow dehumidifier and regenerator with a cross-corrugated ceramic packing and developed the dimensionless empirical correlations of the overall mass transfer coefficient for the dehumidifier and regenerator. Al. Farayedhi et al. (2002) calculated the heat and mass transfer coefficients in a gauze-type metal packing dehumidifier using three types of liquid desiccants. Zurigat et al. (2004) studied the performance of the dehumidifier with two different structured packings, which are made of aluminium and wood, respectively. The dehumidification performance of the aluminium packing was found worse than that of the wood packing when the desiccant flow is not high. The main reason is the inadequate wetness of the aluminium packing. Actually, the wettability of the packing is important for the performance and design of the dehumidifier. The cellulose fibre paper, which is a wood type material, was proved to be a good adsorbent of desiccant and has the best wettability (Potnis and Lenz, 1996). It provides nearly complete wettability. Thus, this type of packing has becoming popular for liquid desiccant research and applications (Gao et al., 2012; Liu et al., 2006a, 2006b). The representative type for this kind of packing is CELdek packing. Potnis and Lenz (1996) developed the dimensionless mass transfer correlations for the CELdek dehumidifier and regenerator with the packing heights of 30cm and 55cm, respectively. Elsarrag et al. (2004, 2007) experimentally investigated the influences of various design parameters on the performance of the dehumidifier employing CELdek packing. Two mass transfer coefficient correlations were developed for low and high desiccant flow rates. It was found that the performance and mass transfer coefficient of the dehumidifier were not influenced by the liquid flow rate when the liquid to air flow ratio was larger than 2. The performance of the dehumidifier was assessed in the comfort zone and design guidelines of the dehumidifier were developed by Elsarrag et al. (2005). Gao et al. (2012) studied influence of the desiccant and air inlet parameters as well as the packing size on the performance of a cross flow CELdek dehumidifier, and found that a better performance can be achieved without increasing the pressure drop by increasing the width, thickness and height simultaneously. Moon et al. (2009) developed a new dehumidification effectiveness correlation of a cross flow dehumidifier, which has a good agreement with the experimental data. It was found that the desiccant flow rate had the dominant effect on the dehumidification effectiveness at low flow ratios of desiccant to air. Liu et al. (2006a, 2006b) conducted a parameter analysis on the influences of inlet parameters on the moisture removal rate, the enthalpy and moisture effectiveness of a cross flow dehumidifier, and developed the empirical correlations of the enthalpy and moisture effectiveness.

Liquid desiccant dehumidifier usually occupies large installation space. One of the main reasons is the packings used have low specific surface area and poor wettability. The specific surface area is defined as the contact area per unit volume of packing. The packing with higher specific surface area can significantly reduce the size of the dehumidifier which is an important consideration in practical applications. The CELdek dehumidifiers studied in previous research work have a specific surface area of $396m^2/m^3$ (Gao et al., 2012). This paper presents an experimental study on the performance of a counter flow dehumidifier, which uses the CELdek structured packing with higher specific surface area, i.e. $650m^2/m^3$. The performance of this type of packing is compared with other two types of structured packing available in literature. The experimental data are compared with the predicted values using correlations reported by Chung (1994) and Moon et al. (2009). New empirical correlations of the moisture and enthalpy effectiveness are developed and validated in a wider experiment range. The new empirical correlations are also evaluated with experiment results reported by Fumo and Goswami (2002). Finally, the influences of various inlet parameters and the packing height on the dehumidification performance are investigated.

2. Experimental test rig

An experimental test rig has been built to study the liquid desiccant dehumidification performance. The schematic of the experimental test rig is shown in Fig.1. The photograph of the experimental test rig is shown in Fig.2. The test rig mainly consists of four parts: the dehumidifier, the air pre-process facilities, the desiccant pre-process facilities and the measurement system. The dehumidifier is the key component in which the air is dehumidified by the strong desiccant solution. Structured packing (CELdek 5090) with a specific surface area of 650 m^2/m^3 is used in the dehumidifier. The packing is made of porous cellulose fiber paper which has a good wettability. The flow pattern between the air and the desiccant solution is counter flow. The cross section of the packing is 0.3m×0.3m. Three types of packing with the heights of 0.3m, 0.4m and 0.5m are tested separately. The dehumidifier is thermally insulated with the environment by 8mm shell of acrylic plastic glazing.

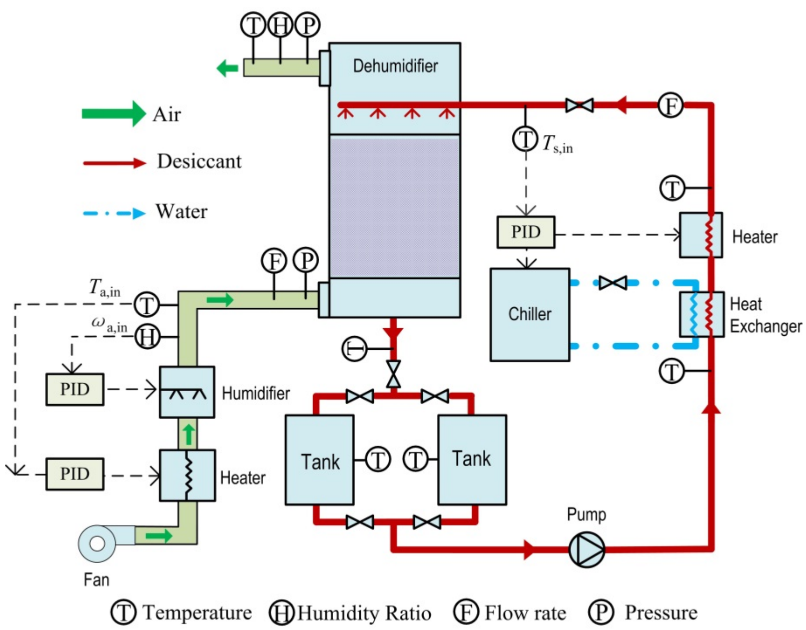

Fig.1. Schematic of experimental test rig.

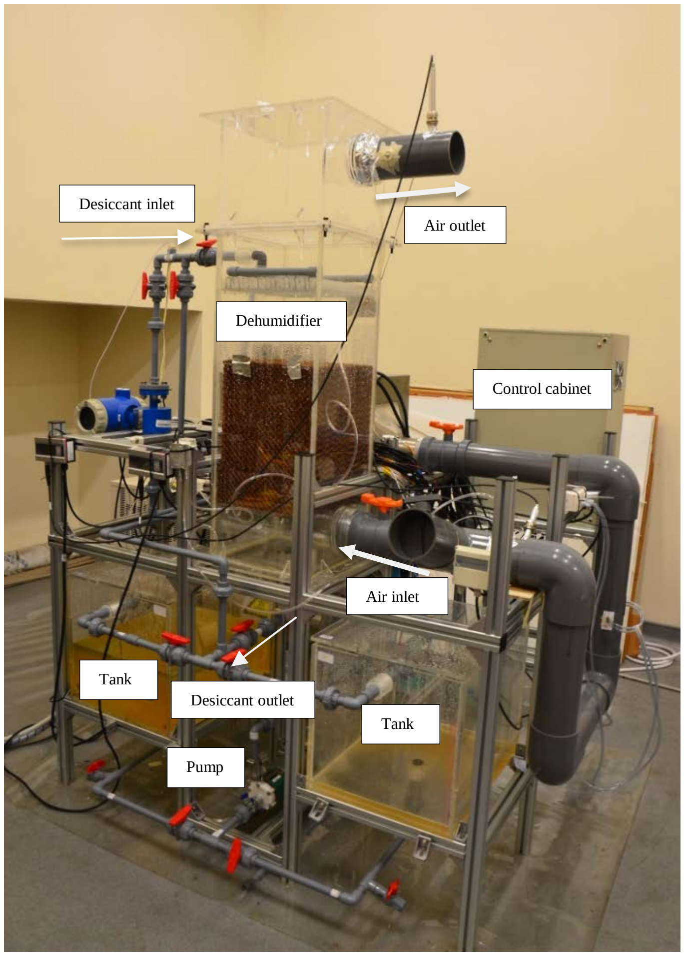

Fig.2. Photograph of experimental test rig.

The air pre-process facilities control the conditions of the air entering the dehumidifier. The temperature and humidity ratio of the air can be controlled at the required conditions by adjusting an electric heater and an electrode humidifier with PID controllers. The desiccant pre-process facilities are used to regulate the flow rate and temperature of the desiccant solution. The lithium chloride (LiCl) aqueous solution is used as the liquid desiccant. Two 40cm(W)×50cm(L)×40cm(H) solution tanks are used. One tank is used to store the strong solution and the other tank stores the diluted solution. The desiccant flow direction between the two tanks can be adjusted manually by four valves. During each experiment, the inlet conditions of the air and the desiccant solution can be maintained stable. The desiccant solution, pumped by a fluorine-lining magnetic pump from the strong desiccant storage tank, can be cooled or heated to a pre-set temperature by a plate heat exchanger and an electric heater. The cooling water in the heat exchanger is produced by a chiller. The solution pipes are CPVC pipes, and all the metal joints and measuring instruments are made of 316L stainless steel for preventing erosion. The desiccant at the required condition is sprayed at the top of the dehumidifier. Even distribution of desiccant solution is critical to the wettability of the packing. A new type of porous plate for uniform distribution of the desiccant solution over the packing has been designed in this study, as shown in Fig. 3. A total of 200 pores for desiccant distribution in the porous plate ensure the desiccant can be uniformly distributed to the packing. Above the porous plate, there are two spray pipes, each with ten equally spaced holes at the bottom. Coupled heat and mass transfer occurs between the desiccant and the air in the dehumidifier, and the diluted desiccant solution leaves at the bottom of the dehumidifier and flows back to the weak solution tank.

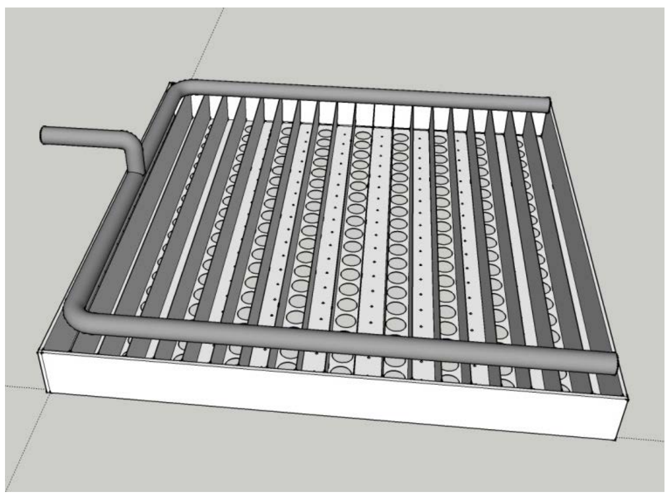

Fig.3. Porous plate distributor for desiccant solution.

Comprehensive instruments for measurement and data collection are installed in the test rig. The temperature of air and desiccant solution are measured by PT100 RTDs (Resistance temperature detector). The air humidity ratio is measured by humidity transducers. The flow rates of air and desiccant solution are measured by a differential pressure flowmeter and an electromagnetic flowmeter, respectively. A specific gravity hydrometer is used to measure the density of the desiccant solution. The concentration of the desiccant solution is calculated by using the temperature and concentration dependent density formula proposed by Conde (2004). The specifications of main measuring instruments used in this study are listed in Table 1. The temperature of air and desiccant as well as the air humidity are maintained constant by PID controllers. All the measuring data are collected by the data acquisition unit Agilent 34972A and stored in the computer.

Table 1
Specifications of measuring instruments.

Device	Type	Accuracy	Range
Thermometer	PT100 RTD	±0.1 ºC	-50-200 ºC
Humidity transducer	HF535-W, HC2-S3	±0.8%RH; ±0.1ºC	0-100%RH; -40-100 ºC
Solution flowmeter	LDE-15 electromagnetic flowmeter	±0.5FS	0.06-6.36m^3/h
Air flowmeter	CP218-BO differential pressure flowmeter	±2%	0-30m/s
Densitometer	Specific gravity hydrometer	±1kg/m^3	1000-1400kg/m^3

3. Experimental conditions and performance indices

In this study, 112 groups of experiments were conducted to test the performance of the counter flow packed-type dehumidifier. A wide experimental range is covered, which are listed in Table 2. To avoid the carryover of desiccant from the dehumidifier, moderate flow rates of the desiccant and the air are adopted. Each experiment with stable inlet conditions can last for 15 minutes, which is long enough for the system to reach and sustain at the steady state for reliable data collection. In order to examine the adiabatic condition of the experiments, the energy balance analysis is conducted.

Table 2
Operating ranges and uncertainty of the experiments.

Parameter	Symbol	Unit	Range	Uncertainty
Air flow rate	m_a	kg/s	0.034-0.082	±0.002
Air inlet temperature	$T_{a,in}$	ºC	25.0-40.5	±0.1
Air inlet humidity ratio	$\omega_{a,in}$	g/kg	10.6-25.1	±0.2
Solution flow rate	m_s	kg/s	0.023-0.120	±0.002
Solution inlet temperature	$T_{s,in}$	ºC	16.4-35.3	±0.1
Solution inlet concentration	ξ	-	0.317-0.401	±0.001
Packing height	H	m	0.3-0.5	±0.001

The enthalpy variations of the air and the solution flowing through the dehumidifier can be calculated by Eq. (1) and Eq. (2), respectively.

$$Q_{\mathrm{a}} = m_{\mathrm{a}}(h_{\mathrm{a,in}} - h_{\mathrm{a,out}}) \tag{1}$$

$$Q_{\mathrm{s}} = m_{\mathrm{s,in}}(h_{\mathrm{s,out}} - h_{\mathrm{s,in}}) + m_{\mathrm{de}}\, h_{\mathrm{s,out}} \tag{2}$$

where, the moisture removal rate of the air (m_{de}) is:

$$m_{\mathrm{de}} = m_{\mathrm{a}}(\omega_{\mathrm{a,in}} - \omega_{\mathrm{a,out}}) \tag{3}$$

The enthalpy of LiCl aqueous solution can be calculated by the fitting formulas reported in literature (Chaudhari and Patil, 2002). However, the fitting coefficients reported by Chaudhari and Patil (2002) do not fit well with their experimental data. Therefore, this study obtains new fitting coefficients from their experimental data, which are shown in Eq. (4).

$$h_s = \mathrm{A} + \mathrm{B}\, T_s + \mathrm{C}\, T_s^{\,2} \text{ , kJ/kg, where, } T_s \text{ in } {}^{\mathrm{o}}\mathrm{C}; \tag{4}$$

A= -5.53883-184.33226ξ+577.90227ξ^2 -73.93852ξ^3+1893.86667 ξ^4

B= 4.22148 -7.07866ξ +13.33801ξ^2 -12.35943ξ^3 -1.28625ξ^4

C= (-6.22815E-5) +0.00589ξ -0.03209ξ^2 +0.05692ξ^3 -0.03644ξ^4

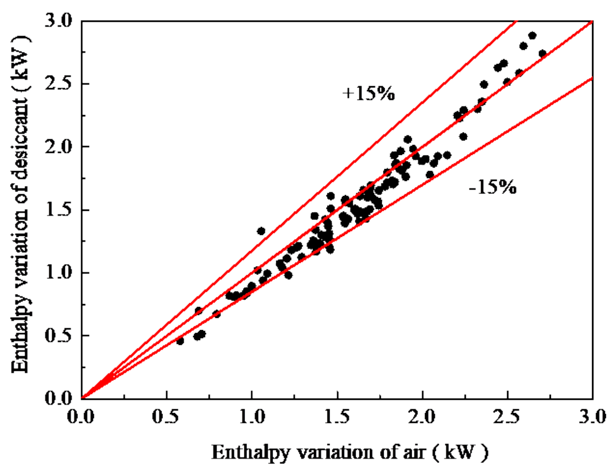

Fig.4. Energy balance analysis between air and desiccant solution.

Fig.4 shows the energy balance of 112 groups of experiment results between the air and the desiccant solution. The results show that almost all of the deviations are within or near ±15%, which indicates that adiabatic condition is well satisfied. Part of the experiment results is given in Table 3. The packing height for cases No.1-10 is 40cm; however, for cases No. 11-12, it is 30cm, and for cases No.13-14, it is 50cm as shown in the first column of Table 3.

The performance of dehumidifiers is usually evaluated by the moisture removal rate (Fumo and Goswami, 2002), the moisture effectiveness (Moon et al., 2009) and the enthalpy effectiveness (Gao et al., 2012). The moisture effectiveness is the ratio of the actual difference between the inlet and outlet air humidity to its maximum possible difference. The enthalpy effectiveness is the same type of ratio for the air enthalpy. These two indices are defined by Eqs. (5)-(6). Knowing these two indices and the inlet air and solution conditions, the leaving air and solution conditions can be determined, which are essential to determine the performance of a dehumidifier and the hybrid system.

Table 3
Dehumidification results of selected experimental conditions.

No.	m_a (kg/s)	$T_{a,in}$ (ºC)	$\omega_{a,in}$ (g/kg)	m_s (kg/s)	$T_{s,in}$ (ºC)	ξ (-)	$\omega_{a,out}$ (g/kg)	$T_{a,out}$ (ºC)	$T_{s,out}$ (ºC)
1	0.0818	30	15.4	0.1179	20.8	0.362	8.8	24.4	26.3
2	0.0802	30	15.8	0.1203	27.8	0.357	11.4	28.4	30.7
3	0.0801	30	17.9	0.1193	20.6	0.340	9.9	23.8	26.1
4	0.0805	25.2	16.2	0.1189	20.8	0.333	9.3	23.3	25.2
5	0.0641	30	16.6	0.0586	21	0.353	9.8	24.9	28.5
6	0.0572	30	13.8	0.0963	24.3	0.350	9.1	25.5	27.2
7	0.0629	32.2	18.8	0.0799	22.7	0.347	10.9	25.7	29.1
8	0.0792	30	12.3	0.0819	21.3	0.358	8.4	24.5	26.3
9	0.0460	30	17	0.1136	21.1	0.358	8.1	23.3	24.9
10	0.0482	29.9	12.2	0.0483	21.3	0.354	8.2	23.8	26.1
11(30cm)	0.0792	29.9	17.2	0.0490	21.3	0.355	11.1	27.2	30.4
12(30cm)	0.0793	30	14.7	0.0818	21.3	0.357	9.4	25.6	27.3
13(50cm)	0.0782	29.9	17	0.0517	21.2	0.355	10.1	26.3	30.6
14(50cm)	0.0791	26.8	16.9	0.0828	21.3	0.356	9.3	24.8	28.2

$$\varepsilon_{\mathrm{m}} = \frac{\omega_{\mathrm{a,in}} - \omega_{\mathrm{a,out}}}{\omega_{\mathrm{a,in}} - \omega_{\mathrm{e,in}}} \tag{5}$$

$$\varepsilon_{\mathrm{h}} = \frac{h_{\mathrm{a,in}} - h_{\mathrm{a,out}}}{h_{\mathrm{a,in}} - h_{\mathrm{e,in}}} \tag{6}$$

where, ω_e and h_e are the equilibrium humidity ratio and enthalpy of air in equilibrium with desiccant surface, which can be expressed by Eqs. (7)-(8). They are related to thermophysical properties of LiCl aqueous solutions. The related fitting formulas of the thermophysical properties have been taken from the literature (Conde, 2004).

$$\omega_{\mathrm{e}} = f(p) = f(T_{\mathrm{s}}\,, \xi) \tag{7}$$

$$h_{\mathrm{e}} = f(\omega_{\mathrm{e}}\,, T_{\mathrm{s}}) \tag{8}$$

Based on the known moisture effectiveness, enthalpy effectiveness and inlet conditions of the air, the moisture removal rate as well as the outlet humidity and enthalpy of the air can be calculated by Eqs. (9)-(11):

$$m_{\mathrm{de}} = \varepsilon_{\mathrm{m}}\, m_{\mathrm{a}}(\omega_{\mathrm{a,in}} - \omega_{\mathrm{e,in}}) \tag{9}$$

$$\omega_{\mathrm{a,out}} = \omega_{\mathrm{a,in}} - \varepsilon_{\mathrm{m}}(\omega_{\mathrm{a,in}} - \omega_{\mathrm{e,in}}) \tag{10}$$

$$h_{\mathrm{a,out}} = h_{\mathrm{a,in}} - \varepsilon_{\mathrm{h}}(h_{\mathrm{a,in}} - h_{\mathrm{e,in}}) \tag{11}$$

When the above air parameters are obtained, the outlet mass flow rate and enthalpy of the desiccant can be calculated by Eqs. (12)-(13).

$$m_{\mathrm{s,out}} = m_{\mathrm{s,in}} + m_{\mathrm{de}} \tag{12}$$

$$h_{\mathrm{s,out}} = \frac{1}{m_{\mathrm{s,in}} + m_{\mathrm{de}}}\left[m_{\mathrm{s,in}}\, h_{\mathrm{s,in}} + m_{\mathrm{a}}(h_{\mathrm{a,in}} - h_{\mathrm{a,out}})\right] \tag{13}$$

Therefore, the overall heat and mass transfer performances of the dehumidifier (including air and desiccant solution) can be determined by using the known moisture effectiveness and enthalpy effectiveness. The moisture effectiveness and the enthalpy effectiveness are two important and very useful performance indices. They are usually determined by empirical correlations, which will be developed in the following part.

The uncertainty analysis for the measured parameters is conducted by using the uncertainty propagation method (Yin et al., 2016), as shown in Eq. (14). The measured parameters can be divided into the directly measured parameters (e.g., T_{a}, H) and the indirectly measured parameters (e.g., m_{de}, ξ). The calculated uncertainties of inlet parameters and packing dimension are shown in Table 2. The uncertainties of the moisture removal rate, the moisture effectiveness and the enthalpy effectiveness in this study are 0.04g/s, 0.03 and 0.03, respectively.

$$\delta u = \sqrt{(\frac{\partial u}{\partial x_1}\delta x_1)^2 + (\frac{\partial u}{\partial x_2}\delta x_2)^2 + (\frac{\partial u}{\partial x_3}\delta x_3)^2 + \cdots + (\frac{\partial u}{\partial x_n}\delta x_n)^2} \tag{14}$$

where δu represents the absolute uncertainty of the indirectly measured parameter u; δx_1, δx_2, δx_3,…, δx_n represents the overall uncertainty of the directly measured parameters x_1, x_2, x_3,…, x_n.

4. Empirical correlations of moisture effectiveness and enthalpy effectiveness

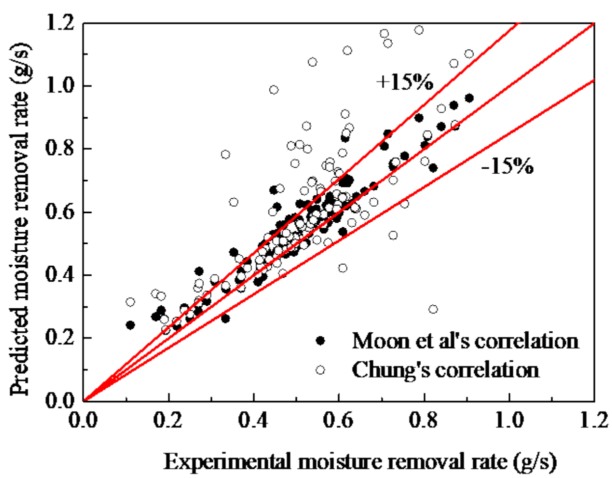

Fig.5. Comparison between predicted results by Moon et al.'s, Chung's correlations and present experiment results.

As mentioned above, the dehumidification capacity of the dehumidifier (i.e. the moisture removal rate) can be predicted by Eq. (9) with the moisture effectiveness correlation. Moon et al. (2009) and Chung (1994) developed the moisture effectiveness correlation for a cross flow and a counter flow dehumidifier with structured packing, respectively. The comparison between the predicted moisture removal rates from their correlations and the present experiment results are shown in Fig. 5. It can be found that Moon et al.'s correlation fits the present experimental data within -15% to +50%. The results from Chung's correlation exhibits even larger deviations from the present experiment results. Although the flow pattern of the dehumidifier in Chung's work is the same as that in present study, Chung's correlation was developed for much larger desiccant flow rate. Both the Moon et al.'s and Chung's correlations can hardly accurately predict the dehumidification performance of the current type of dehumidifier.

Therefore, new empirical correlations of the moisture effectiveness and the enthalpy effectiveness as the functions of inlet parameters of the air and the desiccant as well as the packing height are developed using stepwise regression. The regression equations are given in Eqs. (15)-(16). The validity range of the empirical correlations, which is also the operating range of the experiment tests conducted in this study, are shown in Table 2.

$$\varepsilon_{\mathrm{m}} = 3.5823\, m_{\mathrm{s}}{}^{0.256}\, T_{\mathrm{s,in}}{}^{-0.634}\, \omega_{\mathrm{a,in}}{}^{0.350}\, m_{\mathrm{a}}{}^{-0.322}\, T_{\mathrm{a,in}}{}^{-0.327} \tag{15}$$

$$\varepsilon_{\mathrm{h}} = 0.5644\, m_{\mathrm{s}}{}^{0.324}\, T_{\mathrm{s,in}}{}^{-0.540} m_{\mathrm{a}}{}^{-0.375}\, \xi^{-0.504}\, T_{\mathrm{a,in}}{}^{0.274} \tag{16}$$

Eq. (15) and Eq. (16) indicate that the moisture effectiveness and the enthalpy effectiveness are mainly influenced by five parameters. Four of them are the same, i.e., the air flow rate, desiccant flow rate, desiccant inlet temperature and air inlet temperature. Besides, the air inlet humidity ratio only influences the moisture effectiveness, while the desiccant concentration only influences the enthalpy effectiveness.

The average absolute difference (AAD) between the predicted values and experiment results defined by Eq. (17) is adopted to evaluate the empirical correlations.

$$AAD = \frac{1}{N}\sum_{i=1}^{N}\left|\frac{\Phi_{\mathrm{exp}}-\Phi_{\mathrm{cal}}}{\Phi_{\mathrm{exp}}}\right| \times 100\% \tag{17}$$

where, Φ represents one of the performance indices.

The validation results of the predicted values and the experimental data for the moisture effectiveness, the enthalpy effectiveness and the moisture removal rate are shown in Fig.6. The discrepancies between the predicted values and the experimental data are within ±10% for almost all of these three performance indices. The AADs for the moisture effectiveness, the enthalpy effectiveness and the moisture removal rate are 5.16%, 5.00% and 5.16%, respectively. The comparison results indicate that the new empirical correlations can accurately predict the performance of the dehumidifier.

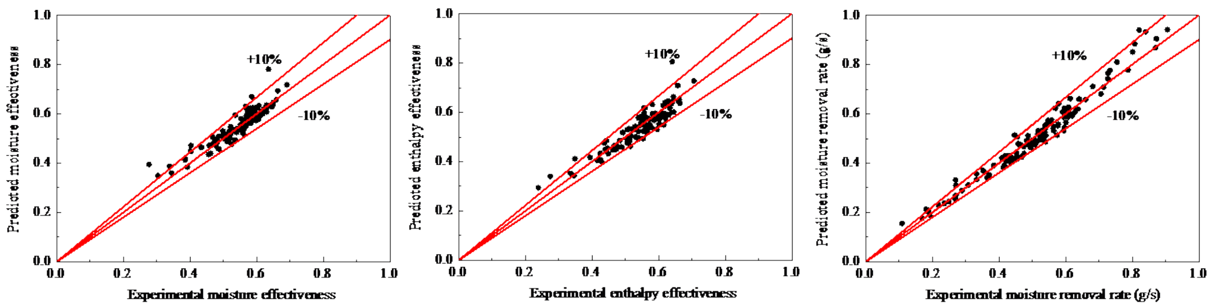

Fig.6. Comparison of predicted results and experiment results for the moisture effectiveness, the enthalpy effectiveness and the moisture removal rate.

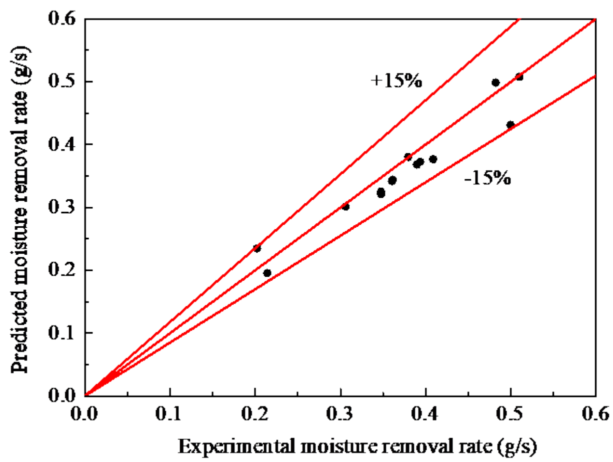

Fig.7. Comparison of predicted moisture removal rates and experiment results from Fumo and Goswami (2002).

The new empirical correlations are also validated using the experimental data from the counter flow dehumidifier tested by Fumo and Goswami (2002). The LiCl solution was used as the liquid desiccant and polypropylene ring was used as the packing material. The predicted results from the new correlations and experiment results of the moisture removal rate are compared, which is shown in Fig. 7. The discrepancies between the predicted values and the experimental data are within ±15%, and the AAD for the moisture removal rate is 6.21%. The predicted results by the new empirical correlations show good agreement with the experiment results of the dehumidifier, although the packing materials are different. Although the desiccant flow rate is much higher in Fumo and Goswami's study, the new empirical correlations also demonstrate satisfactory accuracy.

The performance of the dehumidifier is also compared with other two dehumidifiers using different types of structured packing. Moisture removal rates per unit packing volume for different dehumidifiers are compared at the same inlet conditions of air and desiccant. The specifications of the different packings are shown in Table 4. The comparison results are shown in Fig. 8. It can be seen that No.2 and No.3 dehumidifiers which adopted the CELdek structured packing have much higher moisture removal rates than No.1 dehumidifier. The low dehumidification ability for No.1 dehumidifier may be attributed to poor wettability of the polypropylene gauze-type packing. Compared with No.2 dehumidifier adopted the CELdek packing with the specific surface area of 396m^2/m^3, the present dehumidifier has a higher dehumidification performance at a unit packing volume. Therefore, the size of the dehumidifier can be significantly reduced by using the present packing to meet the requirement of small installation space.

Table 4
Specifications of packing used in different dehumidifiers.

Dehumidifier	Packing type	Specific surface area (m^2/m^3)	Dimension L×W×H (m)
No.1(Yin et al., 2007)	Polypropylene gauze-type	315	1×0.6×0.45
No.2(Gao et al.,2012)	CELdek 7090	396	0.5×0.3×0.5
No.3(Present study)	CELdek 5090	650	0.3×0.3×0.4

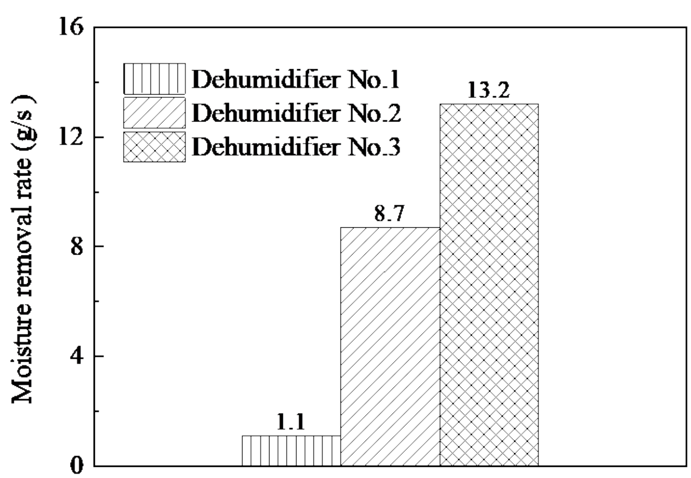

Fig.8. Moisture removal rates per unit packing volume for different dehumidifiers.

5. Parameter analysis on the dehumidification performance

The influences of six inlet parameters of the air and the desiccant as well as the packing height on the dehumidification performance are investigated experimentally. The six inlet parameters including air flow rate, air inlet temperature, air inlet humidity ratio, desiccant flow rate, desiccant inlet temperature and desiccant inlet concentration. The inlet conditions of the air and the desiccant solution are shown in Table 5. The influence of each factor on the moisture effectiveness, the enthalpy effectiveness and the moisture removal rate is analyzed. In addition, the prediction curves for the moisture removal rate obtained from the new correlations are also compared with those from the experimental tests. Finally, the trends of the influence of individual inlet variable and packing height on the dehumidification performance are also compared with the results in literature.

Table 5
Experimental inlet conditions of air and desiccant solution.

	Air			Desiccant solution		
Case	m_a (kg/s)	$T_{a,in}$ (ºC)	$\omega_{a,in}$ (g/kg)	m_s (kg/s)	$T_{s,in}$ (ºC)	ξ(-)
5.1(a)	—	30.0	16.8-17.0	0.113-0.114	21.1-21.3	0.357-0.359
5.1(b)	0.074-0.078	—	16.8-17.0	0.094-0.095	24.8	0.359-0.361
5.1(c)	0.070	33.4-33.6	—	0.094-0.097	22.7-23.0	0.363
5.1(d)	0.063-0.064	30.0	16.3-17.0	—	21.1-21.3	0.355-0.357
5.1(e)	0.079	32.4	20.3-20.5	0.083-0.084	—	0.364-0.366
5.1(f)	0.079-0.080	32.2	21.2-21.4	0.108-0.113	21.1	—
5.2(a)	0.078-0.079	29.9-30.0	16.9-17.2	0.083	21.2-21.3	0.356-0.358
5.2(b)	0.078-0.079	29.9-30.0	16.7-16.9	0.050-0.051	21.1-21.3	0.355-0.357

5.1. Influence of inlet parameters on the dehumidification performance

The influences of the six inlet parameters on the moisture and enthalpy effectiveness as well as the moisture removal rate are shown in Fig. 9 and Fig. 10, respectively. Fig. 10 also shows the prediction curves for the moisture removal rate using the new correlations.

From Fig. 9, it can be seen that the desiccant flow rate and the air inlet humidity ratio have obvious positive influence on the moisture effectiveness, while the air flow rate, the inlet air temperature and the desiccant inlet temperature have obvious negative influence on the moisture effectiveness. For the enthalpy effectiveness, the positive influential parameters include the desiccant flow rate and the air inlet temperature, and the negative influential parameters include the air flow rate, the desiccant inlet temperature and

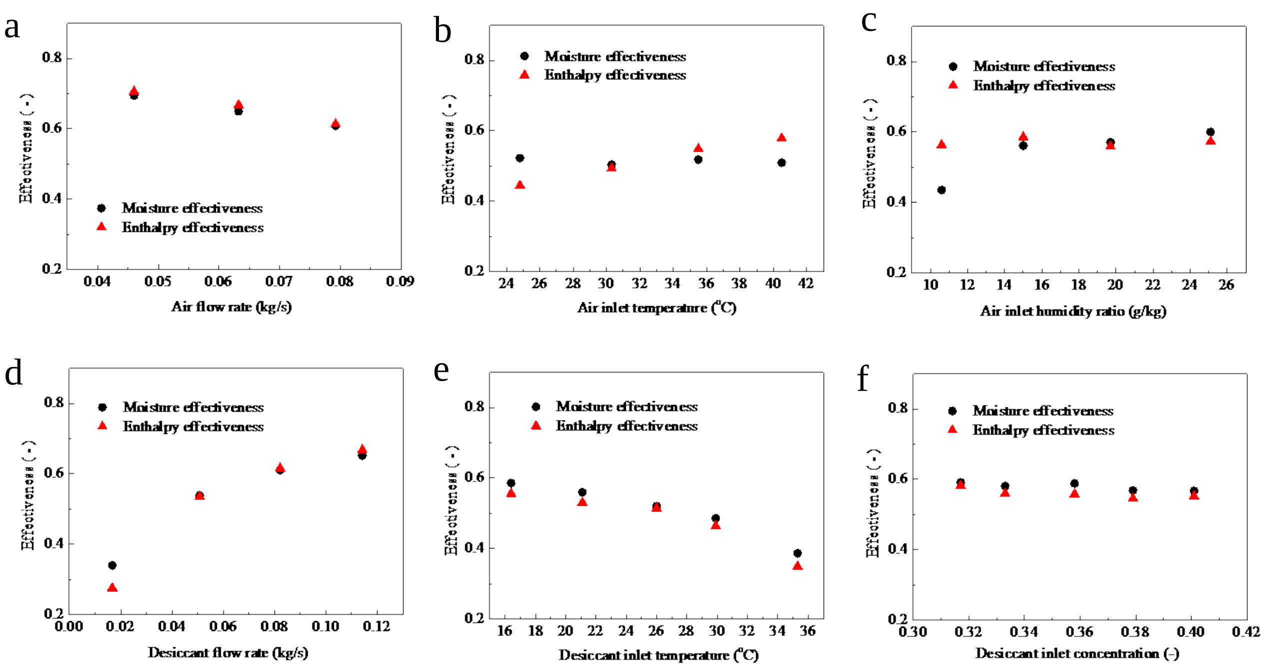

Fig.9. Influences of inlet parameters on the moisture and enthalpy effectiveness.

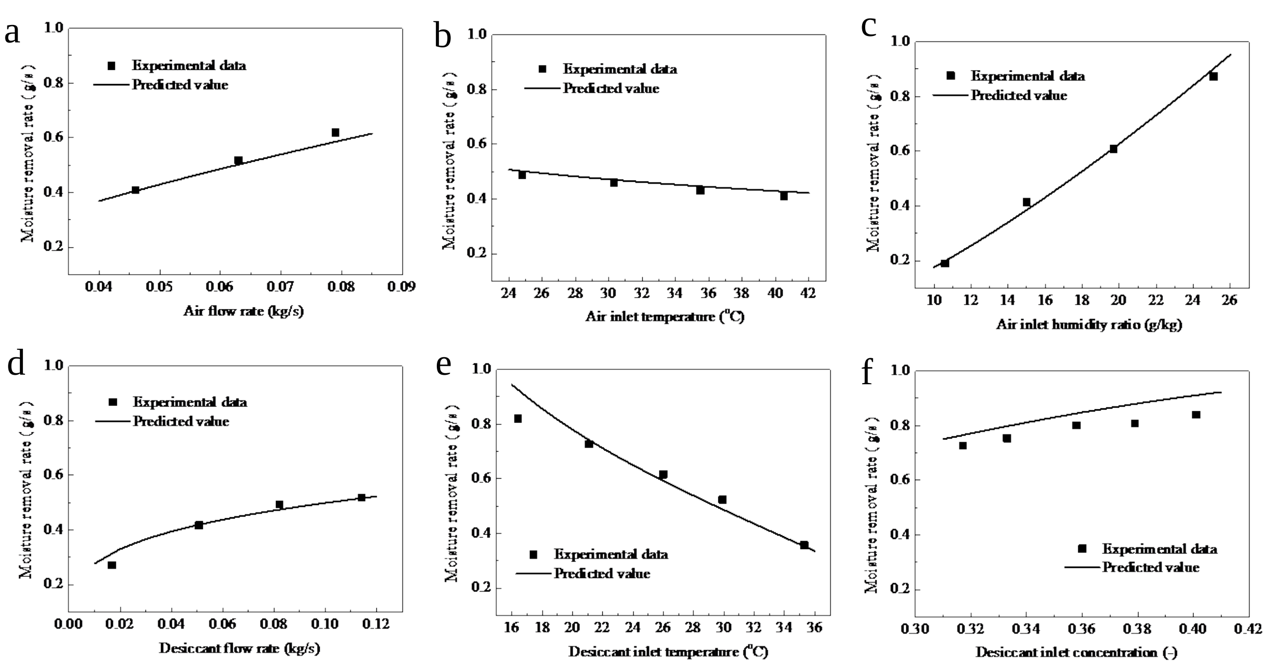

Fig.10. Influences of inlet parameters on the moisture removal rate.

concentration. All of the six parameters have obvious influence on the moisture removal rate, as shown in Fig. 10. The two negative influential parameters are the inlet temperature of the air and the desiccant, while the left four parameters have the positive influence on the moisture removal rate. Increasing the flow rate of air or desiccant leads to an increasing mass transfer coefficient between the air and the desiccant (Zhang et al., 2010). In addition, the rise in temperature and drop in concentration of the desiccant is reduced at a higher desiccant flow rate, which increases the mass transfer potential. As a result, the moisture removal rate increases with a rising flow rate of air or desiccant. However, the effectiveness decreases with the air flow rate because of the shorter contact time. Increasing the temperature of desiccant or air can result in a higher surface vapor pressure of the desiccant, which reduces the mass transfer potential between the air and the desiccant and then reduces the effectiveness and moisture removal rate. The reason for the increase of the moisture removal rate with the desiccant inlet concentration is due to the decrease in the surface vapor pressure of the desiccant. However, a higher surface tension caused by a higher concentration will reduce the wettability of the desiccant (Moon et al., 2009). This effect counteracts the increase of the mass transfer potential, which leads to the moisture effectiveness has little change. The trends observed from Fig. 9 are consistent with the new empirical correlations Eqs. (15)-(16), in which the positive exponent represents the increasing trend while the negative exponent represents the decreasing trend. It also can be found that the prediction curves in Fig. 10 fit well with the experimental data.

The trends of the three performance indices influenced by the inlet parameters are also compared with the results in literature, as shown in Table 6. The mass flow flux (L), which is defined as the mass flow rate per unit cross-sectional area, is used in the Table 6 instead of the mass flow rate (m) for the convenience of comparing with literature. In most cases, the trends reported in this study are similar with those in other studies. In addition, some different trends are also observed. Firstly, the influence of the desiccant flow rate shown in Fig. 9 and Fig. 10 is different from that reported by Oberg and Goswami (1998), and Fumo and Goswami (2002). In their studies the desiccant flow rate almost had no influence on the moisture removal rate and the moisture effectiveness. It is mainly because they used much larger desiccant flow rate, so the dehumidifier has already achieved the maximum dehumidification ability. Secondly, the influence of the air inlet humidity ratio on the moisture effectiveness. It is the increasing trend in the present study while there is no significant influence in other studies. The reason is more desiccant can be stored in this type of packing, so the equilibrium humidity of desiccant in the packing can be maintained at a low level during the dehumidification process, which provides a bigger dehumidification potential for the air with a high inlet humidity ratio. It indicates that this dehumidifier is more suitable to handle the air with high humidity. Thirdly, the influence of the desiccant inlet temperature on the moisture effectiveness. It is the decreasing trend in the present study while the influence is unobvious in other studies. The reason should be attributed to the wider range of the desiccant inlet temperature investigated in this study (i.e., from 16.4°C to 35.3°C) than that in other studies. Another reason may be that the dehumidification performance deteriorates with a high desiccant temperature, and this deteriorating effect is worsen when more desiccant is stored in the packing.

Table 6
Trends of the influence of various experimental conditions on dehumidification performance investigated in this study and reported in previous publications.

Reference	Flow pattern	Desiccant	Performance indice	G (kg/m²s)	$T_{a,in}$ (°C)	$\omega_{a,in}$ (g/kg)	L (kg/m²s)	$T_{s,in}$ (°C)	ξ (-)	H (m)
Present study	Counter flow	LiCl	Range	0.38-0.91	25.0-40.5	10.6-25.1	0.26-1.33	16.4-35.3	0.317-0.401	0.3-0.5
			m_{de}	↑	↓	↑	↑	↓	↑	↔
			ε_m	↓	↓	↑	↑	↓	↔	↔
			ε_h	↓	↑	↔	↑	↓	↓	↔
Oberg and Goswami (1998)	Counter flow	TEG	Range	0.5-2.0	25.0-35.0	11.0-22.0	4.5-6.5	25.0-35.0	0.94-0.96	0.4-0.8
			m_{de}	↑	↔	↑	↔	↓	↑	↑
			ε_m	↓	↔	↔	↔	↔	↔	↑
Fumo and Goswami (2002)	Counter flow	LiCl	Range	0.89-1.51	29.9-40.1	14.0-22.0	5.02-7.42	25.0-35.2	0.33-0.35	
			m_{de}	↑	↔	↑	↔	↓	↑	
Zurigat et al. (2004)	Counter flow	TEG	Range	1.50-2.61	25.4-44.0	16.0-22.0	0.13-1.0	28.0-45.0	0.93-0.98	
			m_{de}	↑	↓		↑	↓	↑	
			ε_m	↓	↓		↑	↑	↑	
Liu et al. (2006a,2006b)	Cross flow	LiBr	Range	1.59-2.43	24.7-33.9	10.0-21.0	2.15-4.55	20.1-29.5	0.426-0.548	
			m_{de}	↑	↔	↑	↑	↓	↑	
			ε_m	↓	↔	↔	↑	↔	↔	
			ε_h	↓	↑	↓	↑	↔	↔	
Moon et al. (2009)	Cross flow	$CaCl_2$	Range	0.91-1.99	26.8-39.0	16-24	1.26-2.57	26.2-38.2	0.33-0.43	
			m_{de}	↑	↓	↑	↑	↓	↑	
			ε_m	↓	↓	↔	↑	↔	↔	
Gao et al. (2012)	Cross flow	LiCl	Range	0.53-0.93	27-38	9.3-21.3	0.67-1.73	22-50	0.32-0.40	
			ε_m	↓	↔	↔	↑	↔	↔	
			ε_h	↓	↑	↓	↑	↔	↔	

↑, Increasing trend; ↓, Decreasing trend; ↔, No significant effect.

5.2. Influence of packing height on the dehumidification performance

The influences of the packing height on the moisture effectiveness, the enthalpy effectiveness and the moisture removal rate are shown in Fig. 11. Two cases of experiments with three different packing heights (0.3m, 0.4m, 0.5m) were conducted. One case (Case a) is at a high desiccant flow rate, the other case (Case b) is at a low desiccant flow rate. According to the new empirical correlations Eqs. (15)-(16), the packing height has no significant influence on the three indices in a wide operating range (when the desiccant flow flux L is less than 1.3 kg/m^{2}s). The results in Fig. 11 show that the three indices slightly increase with the packing height when the packing height is low. However, the increases of these three indices are almost stagnant at a higher packing, especially when the desiccant flow rate is lower.

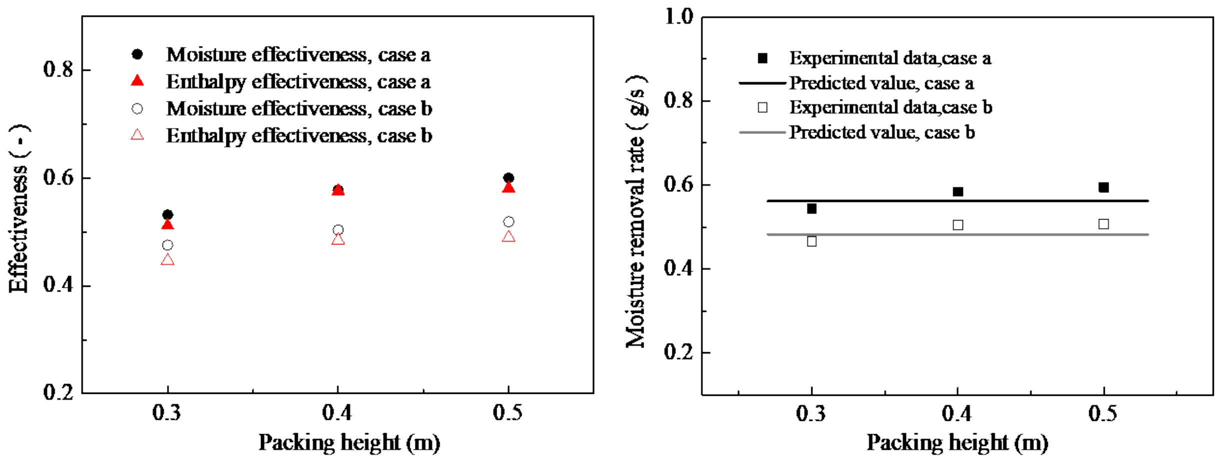

Fig.11. Influence of packing height on the dehumidification performance.

Increasing packing height can enlarge the mass transfer area between air and desiccant, and hence enhances the performance of the dehumidifier. However, the average mass transfer potential reduces when the packing height increases because the desiccant solution is heated by the latent heat released from moisture removal. This effect offsets the increase of the mass transfer area in higher packings. Besides, the solution velocity in the packing could be reduced by the flow resistance of the packing surface when the packing is higher, so more desiccant solution will accumulate in the bottom of the packing and it reduces the actual mass transfer area between air and solution. As a result, the packing height does not show obvious influence on the dehumidification performance. In view of this, there exists a critical value of the packing height for a given desiccant flow rate. When the desiccant flow rate is lower, the decrease of the mass transfer potential is easier to occur, which means the critical height is also lower. Besides, the increase of the packing height may increase the occupied space and the investment. The above analysis indicates that the packing is not the higher the better. Combined with the analysis of Eqs. (15)-(16) and Fig.

11, the packing height should not more than 0.4m when the desiccant flow flux L is less than 1.3 kg/m^{2}s for this type of dehumidifier.

Different from this study, Oberg and Goswami (1998) reported that the dehumidification performance increased with the increase of the packing height, as shown in Table 6. It should be attributed to the large desiccant flow flux adopted in their study. As is discussed above, the dehumidification performance is not sensitive to the packing height at a low desiccant flow, while sensitive to the packing height at a large desiccant flow.

Conclusion

An experimental test rig has been constructed to study the dehumidification performance of a counter flow packed-type dehumidifier. A structured packing with high specific surface area (CELdek 5090) is adopted. The performance of the dehumidifier is evaluated by the moisture effectiveness, the enthalpy effectiveness and the moisture removal rate. New empirical correlations are developed for the moisture effectiveness and the enthalpy effectiveness of the dehumidifier under a wide validation range. The calculated results by the empirical correlations are compared with the experimental data of this dehumidifier and the dehumidifier from Fumo and Goswami (2002), respectively. The comparison results show the deviations between predicted results and experiment results are within ±10% for the former, and within ±15% for the latter. Based on the new empirical correlations, the performance of the dehumidifier can be predicted accurately. The performance of the present type of packing is also compared with other two types of structured packing available in literature. One type is the polypropylene gauze-type, another is the CELdek packing with a lower specific surface area. The results indicate that the present type of packing has the best dehumidification performance when the inlet conditions and packing volume are the same. Using the present type of packing is of benefit to reduce the size of the dehumidifier to meet the requirement of small installation space.

The influence of seven parameters on the performance of the dehumidifier are also analyzed experimentally and using the correlations. The parameters include air flow rate, air inlet temperature, air inlet humidity ratio, desiccant flow rate, desiccant inlet temperature, desiccant concentration and packing height. Both the moisture effectiveness and the enthalpy effectiveness are mainly influenced by the air flow rate, desiccant flow rate, desiccant inlet temperature and air inlet temperature. Besides, the air inlet humidity ratio is found to only influence the moisture effectiveness, while the desiccant concentration only influences the enthalpy effectiveness. The influence trends are compared with those found in previous studies, and the results are generally consistent. This study also shows that this type of dehumidifier works more effectively in handling the air with high humidity ratio, and hence this compact dehumidifier is more suitable for the humid regions. The moisture effectiveness is sensitive to the desiccant inlet temperature in a wide range, which indicates that the desiccant inlet temperature is a suitable control variable for adjusting the supply air humidity. This study also finds that there exists a critical value of the packing height for a given desiccant flow rate, which means the increase of the packing height is not accompanied with an enhanced dehumidification performance when the packing height exceeds this critical value. The experiment results show that the critical height increases with the desiccant flow flux. For the packing studied in this work, the height should not be higher than 0.4m when the desiccant flow flux L is less than 1.3 kg/m^{2}s. The outcomes of this study provide the guidance for the design of compact liquid desiccant dehumidifiers. The new empirical correlations of the dehumidifier are also suitable for the simulation and the development of optimal operation strategies of liquid desiccant based air conditioning equipment.

Acknowledgement

This research was supported by Internal Research Funding (A-PM06) at the Hong Kong Polytechnic University and the National Natural Science Foundation of China (No.51306157). The support is gratefully acknowledged.

References

Al-Farayedhi, A.A., Gandhidasan, P., Al-Mutairi, M.A., 2002. Evaluation of heat and mass transfer coefficients in a gauze-type structured packing air dehumidifier operating with liquid desiccant. International Journal of Refrigeration 25, 330-339.

Chaudhari, S.K., Patil, K.R., 2002. Thermodynamic properties of aqueous solutions of lithium chloride. Phys. Chem. Liq. 40 (3), 317-325.

Chung, T.W., 1994. Predictions of moisture removal efficiencies for packed-bed dehumidification systems. Gas Separation & Purification 8 (4), 265-268.

Chung, T.W., Ghosh, T.K., Hines, A.L., 1996. Comparison between random and structured packings for dehumidification of air by lithium chloride solutions in a packed column and their heat and mass transfer correlations. Ind. Eng. Chem. Res. 35, 192-198.

Conde, M.R., 2004. Properties of aqueous solutions of lithium and calcium chlorides: formulations for use in air conditioning equipment design. International Journal of Thermal Sciences 43(4), 367-382.

Dai, Y.J., Wang, R.Z., Zhang, H.F., Yu, J.D., 2001. Use of desiccant cooling to improve the performance of vapor compression air conditioning. Applied Thermal Engineering 21, 1185-1202.

Elsarrag, E., 2007. Moisture removal rate for air dehumidification by triethylene glycol in a structured packed column. Energy Conversion and Management 48, 327-332.

Elsarrag, E., Elmagzoub, E.M.A., Jain, S., 2004. Mass-transfer correlations for dehumidification of air by triethylene glycol in a structured packed column. Ind. Eng. Chem. Res. 43, 7676-7681.

Elsarrag, E., Elmagzoub, E.M.A., Jain, S., 2005. Design guidelines and performance study on a structured packed liquid desiccant-air conditioning system. HVAC&R Res. 11, 319-337.

Fumo, N., Goswami, D.Y., 2002. Study of an aqueous lithium chloride desiccant system: air dehumidification and desiccant regeneration. Solar Energy 72(4), 351-361.

Gao, W.Z., Liu, J.H., Cheng, Y.P., Zhang, X.L., 2012. Experimental investigation on the heat and mass transfer between air and liquid desiccant in a cross-flow dehumidifier. Renewable Energy 37, 117-123.

Ge, G.M., Xiao, F., Xu, X.H., 2011. Model-based optimal control of a dedicated outdoor air-chilled ceiling system using liquid desiccant and membrane-based total heat recovery. Applied Energy 88, 4180-4190.

Jain, S., Bansal, P.K., 2007. Performance analysis of liquid desiccant dehumidification systems. International Journal of Refrigeration 30, 861-872.

Lazzarin, R.M., Gasparella, A., Longo, G.A., 1999. Chemical dehumidification by liquid desiccants: theory and experiment. International Journal of Refrigeration 22, 334-347.

Liu, X.H., Qu, K.Y., Jiang, Y., 2006a. Empirical correlations to predict the performance of the dehumidifier using liquid desiccant in heat and mass transfer. Renewable Energy 31, 1627-1639.

Liu, X.H., Zhang, Y., Qu, K.Y., Jiang, Y., 2006b. Experimental study on mass transfer performances of cross flow dehumidifier using liquid desiccant. Energy Conversion and Management 47, 2682-2692.

Longo, G.A., Gasparella, A., 2009. Experimental analysis on desiccant regeneration in a packed column with structured and random packing. Solar Energy 83, 511-521.

Moon, C.G., Bansal, P.K., Jain, S., 2009. New mass transfer performance data of a cross-flow liquid desiccant dehumidification system. International Journal of Refrigeration 32, 524-533.

Oberg, V., Goswami, D.Y., 1998. Experimental study of the heat and mass transfer in a packed bed liquid desiccant air dehumidifier. J. Solar .Energy. Eng. 120(4), 289-97.

Potnis, S.V., Lenz, T.G., 1996. Dimensionless mass-transfer correlations for packed-bed liquid-desiccant contactors. Industrial & Engineering Chemistry Research 35, 4185-4193.

Wang, R.Z., Ge, T.S., Chen, C.J., Ma, Q., Xiong, Z.Q., 2009. Solar sorption cooling systems for residential applications: Options and guidelines. International Journal of Refrigeration 32, 638-660.

Wang, X.L., Cai, W.J., Lu, J.G., Sun, Y.X., Ding, X.D., 2013. A hybrid dehumidifier model for real-time performance monitoring, control and optimization in liquid desiccant dehumidification system. Applied Energy 111, 449-455.

Xiao, F., Ge, G.M., Niu, X.F., 2011. Control performance of a dedicated outdoor air system adopting liquid desiccant dehumidification. Applied Energy 88, 143-149.

Yin, Y.G., Zhang, X.S., Chen, Z.Q., 2007. Experimental study on dehumidifier and regenerator of liquid desiccant cooling air conditioning system. Building and Environment 42, 2505-2511.

Yin, Y.G., Zheng, B.J., Chen, T.T., Shao, B., Zhang, X.S., 2016. Investigation on coupled heat and mass transfer coefficients between compressed air and liquid desiccant in a packed dryer. International Journal of Heat and Mass Transfer 93, 1218-1226.

Zhang, L., Hihara, E., Matsuoka, F., Dang, C.B., 2010. Experimental analysis of mass transfer in adiabatic structured packing dehumidifier/regenerator with liquid desiccant. International Journal of Heat and Mass Transfer 53, 2856-2863.

Zhang, L., Hihara, E., Saikawa, M., 2012. Combination of air-source heat pumps with liquid desiccant dehumidification of air. Energy Conversion and Management 57, 107-116.

Zurigat, Y.H., Abu-Arabi, M.K., Abdul-Wahab, S.A., 2004. Air dehumidification by triethylene glycol desiccant in a packed column. Energy Conversion and Management 45, 141-155.