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Efficiency comparison and performance analysis of internally-cooled liquid desiccant dehumidifiers using LiCl and $CaCl_2$ aqueous solutions[*]

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Abstract: Internally-cooled dehumidifiers are efficient liquid desiccant dehumidifiers, whose performance is mainly determined by the device structure and operating conditions. Based on energy and mass conservation in the air, solution, and cooling water in the device, mathematical models are built and their theoretical performance is simulated and analyzed in this paper. A novel measure of dehumidification efficiency is introduced to evaluate the performance of internally-cooled dehumidifiers, in which the equilibrium humidity ratio of the inlet solution is calculated according to the minimum temperature in the inlet solution and the cooling water. Numerical simulations show that a counter flow between air and solution is always the most efficient, followed by cross flow, and parallel flow is the least efficient. Cooling water with the same flow direction as the solution performs better than that with a counter flow, with approximately a 5% improvement in efficiency. Compared with $CaCl_2$, the dehumidification efficiency of a LiCl solution is greater by 60%, while its exergy efficiency is less by 16%. Dehumidification efficiency can be improved with the number of air-solution heat transfer units ($NTU_{a\text{-}s}$) increasing, and reduced with the air mass flow rate raised. With $NTU_{a\text{-}s}$ increasing, exergy efficiency can be improved, and an increase in mass flow rate of cooling water results in a decrease of efficiency. Higher solution concentration and lower inlet temperature of solution and air can achieve both higher dehumidification efficiency and exergy efficiency.

Key words: Liquid desiccant; Internally-cooled dehumidifiers; Performance comparison; Dehumidification efficiency; Exergy efficiency
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1 Introduction

Condensation dehumidification is generally used in conventional air conditioning systems, which remove the sensible heat load and the latent heat load of a building simultaneously, so as to assure the appropriate indoor air conditions. However, there are some disadvantages in the dehumidifying method, such as a waste of energy, poor indoor thermal comfort, and molds growing on wet surfaces.

As one of the alternative technologies for conventional air conditioning, liquid desiccant dehumidification air conditioning is increasingly attracting attention from scholars, because of its environmental friendliness, low-grade driven energy availability, and high density energy-stored ability (Jain et al., 2000, 2011; Jain and Bansal, 2007; Lowenstein, 2008; Tu et

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al., 2010; Yin et al., 2014; Fazilati et al., 2017). In a liquid desiccant air conditioning system, the liquid desiccant dehumidifier, such as a packed dehumidifier, is important (Oberg and Goswami, 1998; Longo and Gasparella, 2005; Bassuoni, 2011). It not only has a simple structure but can be used for liquid desiccant dehumidification and low-grade energy driven solution regeneration.

The packed dehumidifier first appeared as an adiabatic type (Zhang et al., 2010; Koronaki et al., 2013), which was divided into parallel flow, counter flow, and cross flow according to the relative flow directions between air and the solution. It is well-known that a counter flow shows the best dehumidification performance, while, under the same conditions, a parallel flow performs least well. However, the expected dehumidification effect failed to be achieved because of the rise in the solution temperature in an adiabatic dehumidification process.

Internally-cooled dehumidifiers (Chang et al., 2007; Yin et al., 2008; Gao et al., 2013; Luo et al., 2014; Turgut and Coban, 2016) can solve the above problems and consequently, have good performance and high dehumidification efficiency. They have thus received more attention in recent years. Chang et al. (2009) established a theoretical model to study the heat and mass exchange performance and the effect of flow pattern on an internally-cooled dehumidifier. Zhang et al. (2013) conducted an experimental study on an internally-cooled dehumidifier using waste heat and found that the coefficient of performance (COP) can reach 4.2 to 6.5. Qi et al. (2013) established a mathematical model to study the enthalpy efficiency, temperature efficiency, and humidity efficiency of an internally-heated/cooled liquid desiccant dehumidification system. Then Qi and Lu (2014) built an internally-cooled/heated liquid desiccant dehumidification air conditioning system and found that the electricity consumed was reduced by 22%–47%. Liu et al. (2015) designed an internally-heated/cooled liquid desiccant regenerator/dehumidifier made of conductive plastics which showed good performance.

The driving force of the dehumidification process is the humidity ratio of air and the equilibrium humidity ratio of the solution. As the dehumidification progresses, the equilibrium humidity ratio of the solution increases, and the humidity ratio of the air continuously decreases. Finally, the humidity ratios of air and solution reach equilibrium. Dehumidification efficiency was investigated (Chung et al., 1995; Yang and Lian, 2014) to evaluate the performance of a desiccant dehumidifier, and the evaluation indicator has been widely used. Ding et al. (2010) experimented and calculated the dehumidification efficiency of a cross-flow adiabatic dehumidifier and obtained a formula for calculating the dehumidification efficiency of a mixed salt solution in a solar dehumidification system. Peng and Zhang (2016) used dehumidification efficiency and isenthalpic effectiveness to predict the performance of a solution dehumidifier.

Exergy, a thermodynamic quantity, represents useful energy, and exergy analysis as a tool for designing, analyzing, and optimizing thermal systems can truly describe the loss of energy quality and be a guide to technical improvement (Wang et al., 2010; Gonçalves et al., 2014). In exergy analysis, exergy efficiency is usually chosen as an index to reflect the thermodynamic perfection of a certain process, such as a dehumidification process. Xiong et al. (2010) proposed the adjustable liquid desiccant concentration method and the calcium chloride pre-dehumidification method to improve the performance of the system, with an improvement of the system exergy efficiency of 20%. Hürdoğan et al. (2011) experimented with a system, and adopted exergy efficiencies to assess the individual performances within the improvement. Uçkan et al. (2014) proposed exergy transfer and destruction between the components of an evaporative air conditioning system, and obtained a sustainability assessment and relative irreversibility of components. Zhang et al. (2017) proposed a detailed exergy analysis and indicated that the limited transfer capability and the unmatched properties resulting in exergy destruction occurred during the heat and mass transfer processes. Thus, in this paper, exergy analysis is adopted to evaluate the performance of dehumidifiers.

At present, much research on internally-cooled liquid desiccant dehumidifiers is based on a specific device structure and flow pattern and that restricts its general application. In this paper, dimensionless heat and mass transfer mathematical models based on the number of transfer units are proposed to study the effects of a variety of structural and fluid parameters by numerical simulation. A novel definition of dehumidification efficiency which overcomes the disadvantages of the old one is introduced in order to better evaluate the dehumidification performance. Dehumidification efficiency and exergy efficiency models are built to analyze the performance of dehumidifiers under different flow types and inlet parameters and several universal conclusions, not limited by specific device and fluid parameters, are obtained. The results provide suggestions for the design optimization of internally-cooled dehumidifiers, and show that dehumidification efficiency and exergy efficiency can be used to predict their dehumidification performance.

2 Models of internally-cooled dehumidifiers

In a large number of studies on dehumidifiers (Ren et al., 2007; Liu et al., 2009), common flow types of air and solution include parallel, counter, and cross flow, and, in internally-cooled dehumidifiers, cooling water usually flows from top to bottom or from bottom to top without changing the cooling water pipe. According to the flow directions of the three fluids, six mathematical models of internally-cooled dehumidifiers are constructed in this study to study the effects of different air, solution, and cooling water flow types according to the schematic diagrams in Fig. 1. The flow types between air and solution include parallel (Pa.), counter (Co.), and cross (Cr.) flow, while the flow types between solution and cooling water are parallel (I) and counter (II) flow.

In order to build mathematic models successfully, some reasonable hypotheses are proposed as follows:

(1) Assuming the dehumidification process of the device is a steady state.

(2) Assuming that the entire device is adiabatic and ignoring heat and mass exchange with the external environment.

(3) Assuming the solution is evenly distributed in the packed bed with perfect wettability.

(4) Ignoring the heat and mass transfer of solution along the width direction, so that the parallel and counter flow dehumidification process can be simplified into a 1D model, while the cross flow dehumidification process is simplified into a 2D model.

(5) Assuming cooling water is distributed uniformly.

(6) Only considering convective heat and mass transfer and ignoring the thermal conduction and mass diffusion along the flow direction of the fluid itself.

According to the above assumptions, the mathematical models of heat and mass transfer in the dehumidification process of parallel and counter flow are expressed as follows (Liu et al., 2016):

The energy conservation equation among air, solution, and cooling water is given as

$$\mathrm{d}\left(m_{\mathrm{s}}h_{\mathrm{s}}\right)=\delta_{\text{a-s}}m_{\mathrm{a}}\mathrm{d}h_{\mathrm{a}}+\delta_{\text{s-w}}C_{\mathrm{pw}}m_{\mathrm{w}}\mathrm{d}t_{\mathrm{w}}, \tag{1}$$

where m_{s}, m_{a}, and m_{w} are the mass flow rates of solution, air, and cooling water, respectively, kg/s; h_{s} and h_{a} are the enthalpy of solution and air, respectively, kJ/kg; $\delta_{\text{a-s}}$ is the flow coefficient of air and solution, with parallel flow as −1 and counter flow as 1; $\delta_{\text{s-w}}$ is the flow coefficient between solution and cooling water, which is −1 in I flow and 1 in II flow; C_{pw} is the specific heat capacity of cooling water, kJ/(kg·K); t_{w} is the temperature of cooling water, °C.

The mass conservation equation between air and solution is written as

$$m_{\text{solute}}\mathrm{d}R=\delta_{\text{a-s}}m_{\mathrm{a}}\mathrm{d}\omega_{\mathrm{a}}, \tag{2}$$

where m_{solute} is the mass flow rate of solute in solution, kg/s; R is the ratio of water to solute; ω_{a} is the humidity ratio of the air, g/kg.

Energy and mass transfer equations on the air side are expressed as

$$\begin{aligned}&\mathrm{d}h_{\mathrm{a}}\\&=\delta_{\text{a-s}}\left[\left(h_{\mathrm{a}}-h_{\mathrm{e}}\right)+r\left(\frac{1}{Le}-1\right)\left(\omega_{\mathrm{a}}-\omega_{\mathrm{e}}\right)\right]\times\mathrm{d}\left(\mathrm{NTU}_{\text{a-s}}\right),\end{aligned} \tag{3}$$

$$\mathrm{d}\omega_{\mathrm{a}}=\delta_{\text{a-s}}\frac{1}{Le}\left(\omega_{\mathrm{a}}-\omega_{\mathrm{e}}\right)\mathrm{d}\left(\mathrm{NTU}_{\text{a-s}}\right), \tag{4}$$

where h_{e} is the enthalpy of air in equilibrium with the solution, kJ/kg; r is the latent heat of water vaporization, kJ/kg; ω_{e} is the humidity ratio of the air in equilibrium with the solution, g/kg; $\mathrm{NTU}_{\text{a-s}}$ is the number of transfer units between air and solution.

The heat transfer equation between cooling water and solution is shown as

$$\mathrm{d}t_{\mathrm{w}} = \delta_{\text{s-w}}\left(t_{\mathrm{w}} - t_{\mathrm{s}}\right)\mathrm{d}\left(\mathrm{NTU}_{\text{s-w}}\right), \tag{5}$$

where t_{s} is the temperature of solution, °C; $\mathrm{NTU}_{\text{s-w}}$ is the number of transfer units between solution and cooling water.

The dimensionless parameter Lewis number (Le) is defined as

$$Le = \frac{K}{K_{\mathrm{m}} C_{\mathrm{pa}}}, \tag{6}$$

and d($\mathrm{NTU}_{\text{a-s}}$) and d($\mathrm{NTU}_{\text{s-w}}$) can be specified as

$$\mathrm{d}\left(\mathrm{NTU}_{\text{a-s}}\right) = \frac{K A_1 \mathrm{d}x}{H m_{\mathrm{a}} C_{\mathrm{pa}}}, \tag{7}$$

$$\mathrm{d}\left(\mathrm{NTU}_{\text{s-w}}\right) = \frac{K_{\mathrm{w}} A_2 \mathrm{d}x}{H m_{\mathrm{w}} C_{\mathrm{pw}}}, \tag{8}$$

where K is the heat transfer coefficient between air and solution; K_{w} is the heat transfer coefficient between solution and water, kW/(m^2·K); K_{m} is the mass transfer coefficient between air and solution, kg/(m^2·s); A_1 is the contact area between air and solution; A_2 is the contact area between solution and cooling water, m^2; C_{pa} is the specific heat capacity of air, kJ/(kg·K); H is the height of the device, m; x denotes the solution flow direction.

Eqs. (9)–(13) are mathematical models of heat and mass transfer in cross flow.

The energy conservation equation among air, solution, and cooling water is given as

$$\frac{\partial\left(m_{\mathrm{s}} h_{\mathrm{s}}\right)}{L\partial x} = -\frac{m_{\mathrm{a}}\partial h_{\mathrm{a}}}{H\partial y} + \frac{\delta_{\text{s-w}} C_{\mathrm{pw}} m_{\mathrm{w}} \partial t_{\mathrm{w}}}{L\partial x}, \tag{9}$$

where L is the length of the device, m; y donates the air flow direction.

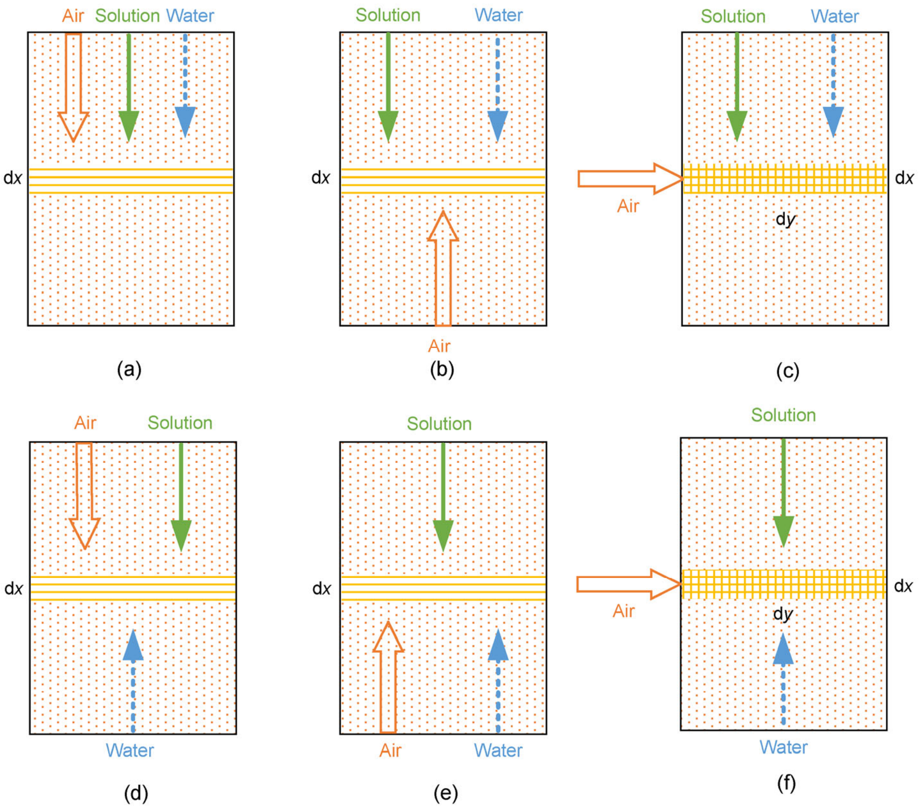


Fig. 1 Physical models of internally-cooled dehumidifiers
(a) Pa.-I; (b) Co.-I; (c) Cr.-I; (d) Pa.-II; (e) Co.-II; (f) Cr.-II

The mass conservation equation between solution and air is written as

$$\frac{m_{\mathrm{solute}}\partial R}{L\partial x}=-\frac{m_{\mathrm{a}}\partial \omega_{\mathrm{a}}}{H\partial y}. \tag{10}$$

The energy and mass control equations on the air side are expressed as

$$\frac{\partial h_{\mathrm{a}}}{\partial y}=\left[\left(h_{\mathrm{e}}-h_{\mathrm{a}}\right)+r\left(\frac{1}{Le}-1\right)\left(\omega_{\mathrm{e}}-\omega_{\mathrm{a}}\right)\right]\frac{\mathrm{NTU}_{\mathrm{a\text{-}s}}}{L}, \tag{11}$$

$$\frac{\partial \omega_{\mathrm{a}}}{\partial y}=\frac{\mathrm{NTU}_{\mathrm{a\text{-}s}}}{Le\times L}\left(\omega_{\mathrm{e}}-\omega_{\mathrm{a}}\right). \tag{12}$$

The heat transfer equation between cooling water and solution is shown as

$$\frac{\partial t_{\mathrm{w}}}{\partial x}=\delta_{\mathrm{s\text{-}w}}\left(t_{\mathrm{w}}-t_{\mathrm{s}}\right)\frac{\mathrm{NTU}_{\mathrm{s\text{-}w}}}{H}. \tag{13}$$

3 Validation of models

To verify the accuracy of internally-cooled liquid desiccant dehumidification models, the moisture removal rate, outlet air humidity ratio and temperature, and outlet solution temperature calculated by mathematical models are compared with experimental results from the literature. The calculation methods of the shape parameters of structured and random packing are given in (Al-Farayedhi et al., 2002) with the solution physical properties in (Conde, 2004). Heat and mass transfer coefficients are calculated from the heat and mass transfer quantity and temperatures. The parameters of dehumidifiers in the compared references are shown in Table 1 while Table 2 presents the initial parameters of air, solution, and water.

The comparisons of moisture removal rate, outlet air humidity ratio and temperature, and outlet solution temperature determined in this study with the experimental data in (Bansal et al., 2011; Zhang et al., 2013; Liu et al., 2015) are exhibited in Fig. 2. The figures show that the range of relative errors between the analog data and the comparison data is within 10%. In general engineering, an error within 10%, considering experimental system and measurement errors, is acceptable and can be taken as verifying the accuracy of the simulation programs.

4 Mathematical models of efficiency analysis

4.1 Dehumidification efficiency

The dehumidification efficiency is commonly defined as the ratio between the difference between the inlet humidity ratio ($\omega_{\mathrm{a,in}}$) and the outlet humidity ratio ($\omega_{\mathrm{a,out}}$) of dehumidified air and the difference between the inlet air humidity ratio and the inlet solution equilibrium humidity ratio ($\omega_{\mathrm{e,in}}$) (Gandhidasan, 2004), expressed by η_{D}:

$$\eta_{\mathrm{D}}=\frac{\omega_{\mathrm{a,in}}-\omega_{\mathrm{a,out}}}{\omega_{\mathrm{a,in}}-\omega_{\mathrm{e,in}}}. \tag{14}$$

However, the result for dehumidification efficiency expressed here has three disadvantages because Eq. (14) fails to consider the cost of cooling water manufactured in internally-cooled dehumidifiers. Specifically:

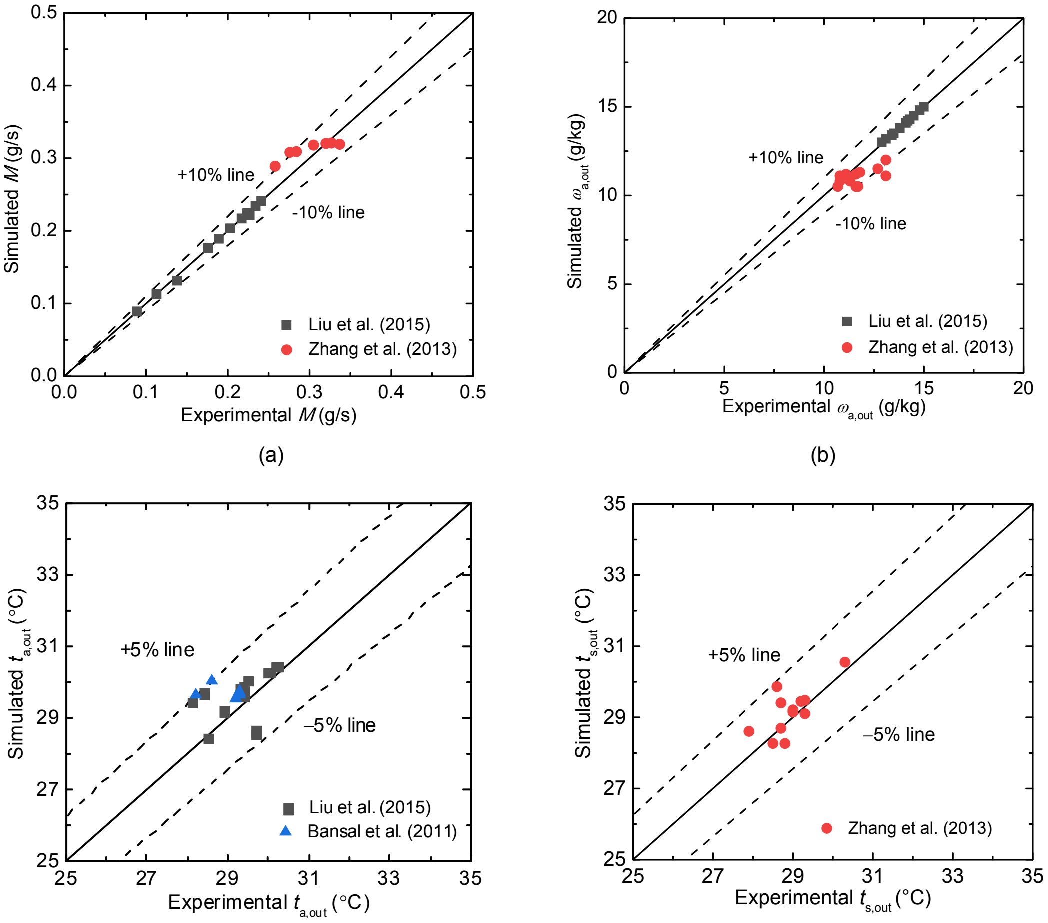
Table 1 Experimental parameters of dehumidifiers in the compared references

Reference	Volume (m^3)	Specific surface area ($\mathrm{m}^2/\mathrm{m}^3$)	$\mathrm{NTU}_{\mathrm{a\text{-}s}}$	$\mathrm{NTU}_{\mathrm{s\text{-}w}}$
Liu et al., 2015	0.0318	342	1.48–2.48	0.120–0.258
Zhang et al., 2013	0.0318	790	0.48	0.70–2.45
Bansal et al., 2011	0.0270	608	0.138–0.200	0.150–0.174

Table 2 Experimental inlet parameters of fluids in the compared references

Reference	m_{a} (kg/s)	$t_{\mathrm{a,in}}$ (°C)	$\omega_{\mathrm{a,in}}$ (g/kg)	m_{s} (kg/s)	$t_{\mathrm{s,in}}$ (°C)	$X_{\mathrm{s,in}}$ (%)	m_{w} (kg/s)	$t_{\mathrm{w,in}}$ (°C)
Liu et al., 2015	0.121–0.180	33.5–36.3	15.2–18.9	0.044–0.134	28.6–31.5	38.9–42.1	0.126–0.280	11.9–16.8
Zhang et al., 2013	0.358–0.370	29.6–33.4	15.4–17.0	0.041–0.178	24.2–28.5	43.0–45.5	0.101–0.359	16.4–25.5
Bansal et al., 2011	0.1	29.9–30.3	20.35–20.70	0.045–0.227	30.3–30.4	37.0	0.083	4.8–8.7

Note: $t_{\mathrm{a,in}}$, $t_{\mathrm{s,in}}$, $t_{\mathrm{w,in}}$ are the inlet temperatures of air, solution, and cooling water, respectively, °C; $X_{\mathrm{s,in}}$ is the inlet solution concentration, %



(a) (b)

(c) (d)

Fig. 2 Accuracy verification of internally-cooled dehumidifier models

(a) Moisture removal rate (M); (b) Outlet air humidity ratio ($\omega_{a,out}$); (c) Outlet air temperature ($t_{a,out}$); (d) Outlet solution temperature ($t_{s,out}$)

1. The dehumidification efficiency exceeds the range of 0 to 1 in some cases. As shown in Fig. 3a, when an inlet $CaCl_2$ solution with a concentration of 36% and cooling water are at temperatures of 36 °C and 16 °C, respectively, dehumidification efficiencies of the internally-cooled dehumidifier are all greater than 1.0 in the parallel, cross, and counter flow types, reaching about 1.05, 1.12, and 1.17. Figs. 3c and 3e also reveal that the dehumidification efficiencies as defined above will lie out of that range when, under certain conditions in internally-cooled dehumidifiers, the solution temperature increases and the concentration of the solution decreases to some extent, and they can even be negative. A dehumidification efficiency outside the range of 0 to 1 violates the generally understood definition of efficiency.

2. The dehumidification efficiencies of a $CaCl_2$ solution may be greater than for LiCl when the solution temperature is higher than 30 °C under the same concentration, which is reflected in Fig. 3a. Because LiCl shows a better dehumidification performance than $CaCl_2$ in the same conditions, the calculated dehumidification efficiency does not reflect actual dehumidification performance.

3. The dehumidification efficiencies increase with increasing temperature of solution. In fact, more heat is carried to an internally-cooled dehumidifier as the solution temperature increases, resulting in a decrease in dehumidification performance, so the dehumidification efficiencies of solutions can be reduced.

Consequently, a novel dehumidification efficiency η_D^*, expressed in Eq. (15), is introduced in this study. In it the inlet solution equilibrium humidity ratio is calculated according to the minimum temperature among the inlet solution and cooling water.

$$\eta_{\mathrm{D}}^{*}=\frac{\omega_{\mathrm{a,in}}-\omega_{\mathrm{a,out}}}{\omega_{\mathrm{a,in}}-\omega_{\mathrm{e,in}}^{*}}, \tag{15}$$

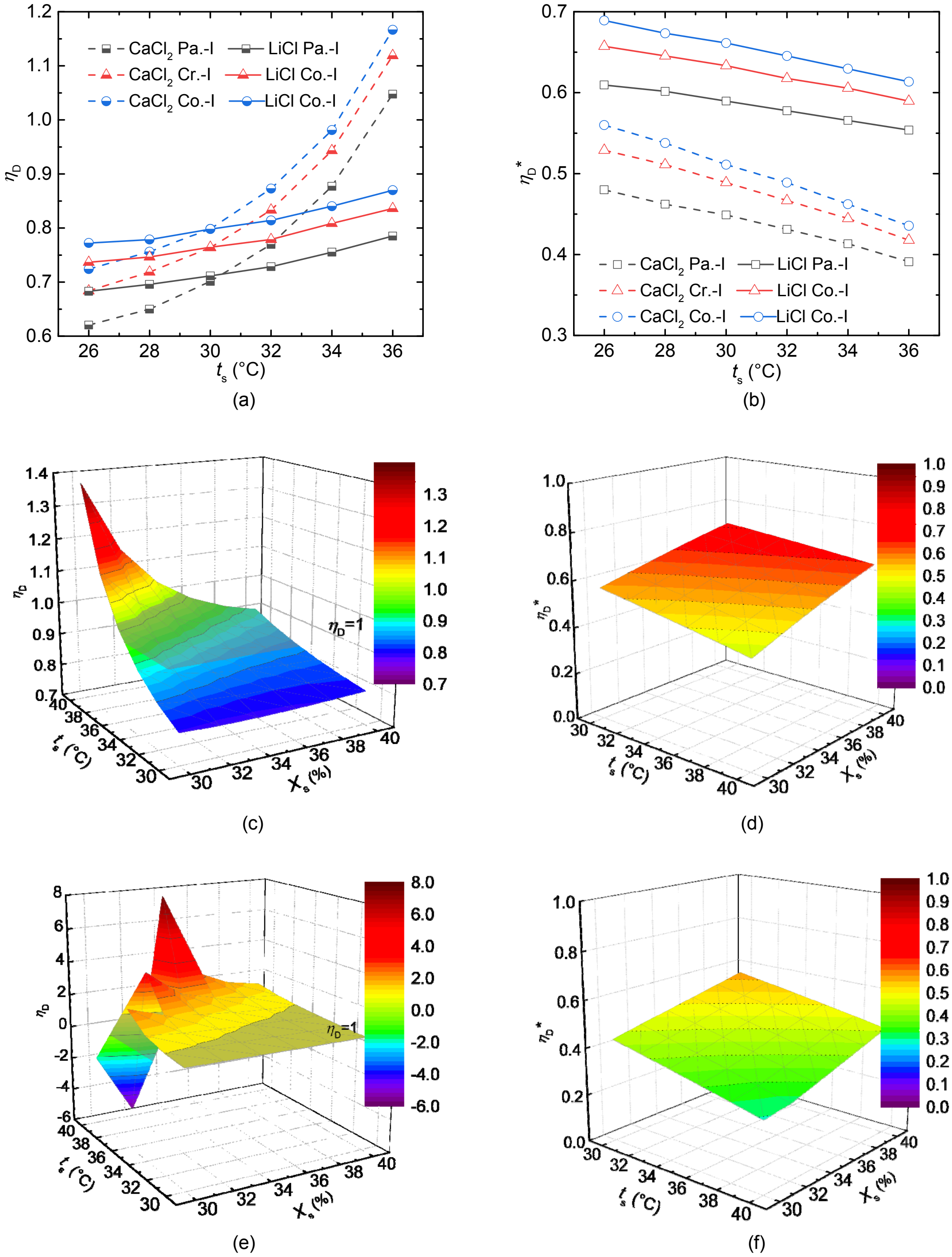


Fig. 3 Comparison of the two dehumidification efficiencies

(a) Effect of solution temperature on η_{D}; (b) Effect of solution temperature on η_{D}^{*}; (c) Effect of LiCl solution temperature and concentration on η_{D}; (d) Effect of LiCl solution temperature and concentration on η_{D}^{*}; (e) Effect of $\mathrm{CaCl_2}$ solution temperature and concentration on η_{D}; (f) Effect of $\mathrm{CaCl_2}$ solution temperature and concentration on η_{D}^{*}

$$\omega_{\mathrm{e,in}}^{*}=f\left[X,\min\left(t_{\mathrm{s,in}},t_{\mathrm{w,in}}\right)\right],\tag{16}$$

where X is the concentration of the liquid desiccant, %.

This novel dehumidification efficiency overcomes previous shortcomings by giving consideration of the cost of the cooling water. The new dehumidification efficiency η_{D}^{*} will lie within the easily understood range of 0 to 1. As shown in Fig. 3b, the new dehumidification efficiencies η_{D}^{*} of $CaCl_2$ solution under parallel, cross, and counter flow types now become 0.39, 0.42, and 0.44 instead of being above 1.0 in the same simulated conditions. The new dehumidification efficiencies of LiCl are greater than those of $CaCl_2$ which accurately reflects the dehumidification performance of the two solutions, and the novel dehumidification efficiencies decline with a rise in solution temperature but increase with increasing concentrations of the solutions. All these are shown in Figs. 3d and 3f. Therefore, all dehumidification efficiencies adopted in the following sections are defined according to the proposed new formula.

4.2 Exergy efficiency

Exergy, also known as usable energy or useful energy, refers to the part of energy that can be converted into useful work under certain referenced conditions. It reflects the level of energy grade. Exergy (E) includes physical exergy (E_{p}) and chemical exergy (E_{c}). Physical exergy can be further divided into heat exergy, cooling exergy, and enthalpy exergy.

The exergy of a system is related not only to the state parameters of the materials in the system, but also to the selected reference points. The correct choice of the reference point of exergy is of paramount importance for analysis of a system, because unreasonable choices can result in wrong analyses. The reference point selected in this study is the ultimate dead state point, the saturated state, which is the equilibrium state of the heat and mass source. In that state the material has no useful energy available, and it can be taken as the environmental reference point in the exergy analysis of moist air in air conditioning engineering. It is the point of zero exergy. In this study, the saturated state parameters of moist air under atmosphere pressure P_0=101325 Pa, and environmental temperature t_0=36 °C are selected as the reference point of zero exergy with a saturated humidity ratio ω_0=38.8 g/kg.

1. Exergy of moist air

The exergy of moist air is composed of physical exergy ($E_{\mathrm{a,p}}$) and chemical exergy ($E_{\mathrm{a,c}}$) (Zhu, 1988), and can be expressed as follows:

$$\begin{aligned}E_{\mathrm{a}}&=E_{\mathrm{a,p}}+E_{\mathrm{a,c}}=m_{\mathrm{a}}\left(C_{\mathrm{pa}}+\omega C_{\mathrm{pv}}\right)T_0\left(\frac{T_{\mathrm{a}}}{T_0}-1-\ln\frac{T_{\mathrm{a}}}{T_0}\right)\\&+m_{\mathrm{a}}R_{\mathrm{a}}T_0\left[\left(1+1.608\omega\right)\ln\frac{1+1.608\omega_0}{1+1.608\omega}+1.608\omega\ln\frac{\omega}{\omega_0}\right],\end{aligned}\tag{17}$$

where C_{pv} is the specific heat capacity of water vapor, kJ/(kg·K); T_{a} is the air temperature, K; R_{a} is the gas constant of air, kJ/(kg·K).

2. Exergy of liquid water

The exergy of liquid water in a closed cycle under normal pressure consists only of the enthalpy exergy ($E_{\mathrm{w,p}}$) (Peng et al., 2017), expressed as

$$\begin{aligned}E_{\mathrm{w}}&=E_{\mathrm{w,p}}=m_{\mathrm{w}}C_{\mathrm{pw}}\int_{T_{\mathrm{w}}}^{T_0}\left(\frac{T_0}{T_{\mathrm{w}}}-1\right)\mathrm{d}T\\&=m_{\mathrm{w}}C_{\mathrm{pw}}\left(T_{\mathrm{w}}-T_0+T_0\ln\frac{T_0}{T_{\mathrm{w}}}\right),\end{aligned}\tag{18}$$

where T is the temperature, K.

3. Exergy of desiccant solution

The exergy of the desiccant solution consists of physical exergy ($E_{\mathrm{s,p}}$) and chemical exergy ($E_{\mathrm{s,c}}$). The physical exergy (Eq. (19)) is related to temperature, while the chemical exergy (Eq. (20)) is determined by solution concentration (Zhu, 1988). The specific expressions of the exergy are:

$$E_{\mathrm{s,p}}=m_{\mathrm{s}}C_{\mathrm{ps}}\int_{T_0}^{T_{\mathrm{s}}}\left(1-\frac{T_0}{T_{\mathrm{s}}}\right)\mathrm{d}T=m_{\mathrm{s}}C_{\mathrm{ps}}\left(T_{\mathrm{s}}-T_0+T_0\ln\frac{T_0}{T_{\mathrm{s}}}\right),\tag{19}$$

$$E_{\mathrm{s,c}}=m_{\mathrm{s}}\sum_{i=1}^{k}x_i\left(\mu_{i,0}-\mu_i^0\right)=m_{\mathrm{s}}\left(\sum_{i=1}^{k}x_iR_{\mathrm{v}}T_0\ln\frac{x_{i,0}}{x_i^0}\right),\tag{20}$$

where C_{ps} is the specific heat capacity of solution, kJ/(kg·K); x_i donates the mole fraction of component i; μ_0 is the chemical potential of dead state; μ^0 is the chemical potential of non-binding dead state, kJ/kg; R_{v} is the gas constant of water vapor, kJ/(kg·K).

The total exergy of the desiccant solution is physical exergy plus chemical exergy, and is written as

$$E_{\mathrm{s}} = E_{\mathrm{s,p}} + E_{\mathrm{s,c}}. \tag{21}$$

4. Exergy conservation equation

The exergy conservation equation of an internally-cooled dehumidifier is shown as

$$E_{\mathrm{a,in}} + E_{\mathrm{s,in}} + E_{\mathrm{w,in}} = E_{\mathrm{a,out}} + E_{\mathrm{s,out}} + E_{\mathrm{w,out}} + I, \tag{22}$$

where $E_{\mathrm{a,in}}$, $E_{\mathrm{s,in}}$, and $E_{\mathrm{w,in}}$ are the inlet exergy of air, solution, and cooling water, and $E_{\mathrm{a,out}}$, $E_{\mathrm{s,out}}$, and $E_{\mathrm{w,out}}$ are the outlet exergy of air, solution, and cooling water, respectively, kJ; I is the exergy destruction, kJ.

5. Exergy evaluation index: exergy efficiency

Exergy efficiency, as an index of the thermodynamic perfection of a process in exergy analysis, refers to the ratio of exergy as a benefit to exergy as a cost, in the energy conversion process, which is given by

$$\eta_{\mathrm{E}} = \frac{\Delta E_{\mathrm{a}}}{\Delta E_{\mathrm{s}} + \Delta E_{\mathrm{w}}} = \frac{E_{\mathrm{a,out}} - E_{\mathrm{a,in}}}{\left(E_{\mathrm{s,in}} - E_{\mathrm{s,out}}\right) + \left(E_{\mathrm{w,in}} - E_{\mathrm{w,out}}\right)}. \tag{23}$$

5 Performance analysis of internally-cooled liquid desiccant dehumidifiers

The dehumidification performance of internally-cooled dehumidifiers is affected by various parameters, including the flow types of fluids, the number of transfer units (NTU) of fluids, the properties of liquid desiccants, and the inlet parameters of air, solution, and cooling water. As evaluation indexes, dehumidification efficiency and exergy efficiency are adopted to analyze the effects of above parameters on dehumidification performance. The LiCl and $\mathrm{CaCl_2}$ aqueous solutions are used as the liquid desiccants in this simulation as they are common dehumidification solutions (Dai and Zhang, 2004; Mei and Dai, 2008; Xiao et al., 2011), with their initial parameters shown in Table 3.

Table 3 Initial parameters of simulation

Item	$\mathrm{NTU_{a\text{-}s}}$	$\mathrm{NTU_{s\text{-}w}}$	m_{a} (kg/s)	m_{s} (kg/s)	m_{w} (kg/s)
Standard condition	2	0.5	0.3	0.15	0.3
Range	1–3	0.1–0.9	0.1–0.5	0.05–0.25	0.1–0.5

Item	$t_{\mathrm{a,in}}$ (°C)	$t_{\mathrm{s,in}}$ (°C)	$t_{\mathrm{w,in}}$ (°C)	$\omega_{\mathrm{a,in}}$ (g/kg)	$X_{\mathrm{s,in}}$ (%)
Standard condition	36	30	16	28	36
Range	32–40	26–34	12–20	24–32	32–40

5.1 Effects of device parameters on the efficiencies of internally-cooled dehumidifiers

5.1.1 Effects of flow types

The flow types between air and solution are divided into parallel, cross, and counter flow, and their effects on the two efficiencies are shown in Figs. 4a and 4b. As shown in Fig. 4a, with the efficiencies of parallel flow set as abscissa, dehumidification efficiency of cross flow increases by 16% at its maximum, while in the counter flow it is 22% higher than in the parallel type. The exergy efficiency of counter flow increases by 28% at most, while that of cross flow rises by 16%. Counter flow turns out to be the highest in both dehumidification and exergy efficiencies, and the efficiencies of cross flow are less, and parallel flow has the lowest efficiencies. The exergy efficiency of counter flow improves obviously at high values, while the exergy efficiency of cross flow increases evenly in the whole process. As a result of more even driving potentials of heat and mass transfer in counter and cross flow than in parallel flow, better heat and mass transfer performance, as reflected by dehumidification efficiency and exergy efficiency, are shown.

The effects of the flow types between solution and cooling water involving I and II conditions on the efficiencies are indicated in Figs. 4c and 4d, respectively. Different flow types of solution and cooling water, have small influences that do not exceed 4% and 6% respectively of the increases of dehumidification and exergy efficiencies. When the solution and cooling water flows under I condition, the temperature difference of heat transfer changes a little and stays stable during the process, so the effect of heat transfer is more ideal. All the following conditions are based on I flow, because type I performs a little better than type II.

5.1.2 Effects of the number of transfer units

The number of heat transfer units has an important role in dehumidification performance, which is affected by the heat transfer area of the device and the heat transfer coefficient between fluids. The number of heat transfer units between air and solution ($\mathrm{NTU_{a\text{-}s}}$) is dominated by the heat transfer area of packing and the heat transfer coefficient between air and solution, and the number of heat transfer units

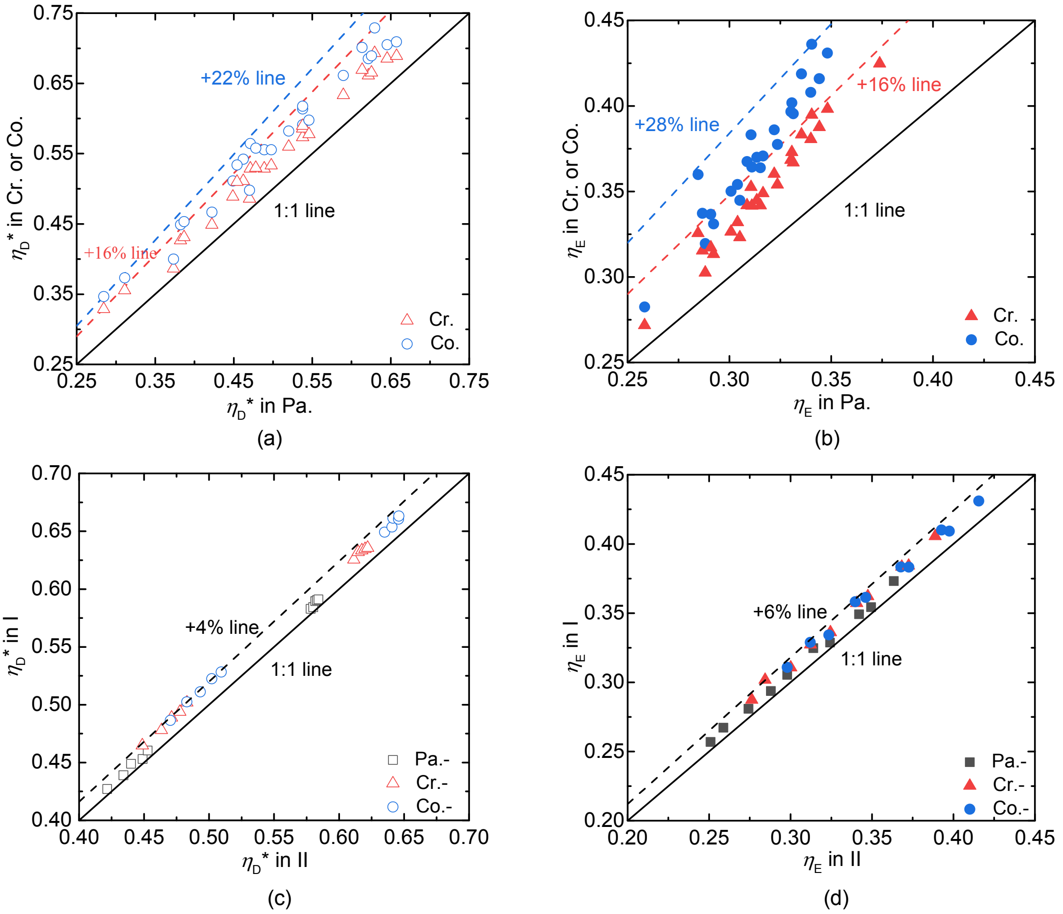


Fig. 4 Effects of flow types on the efficiencies of internally-cooled dehumidifiers

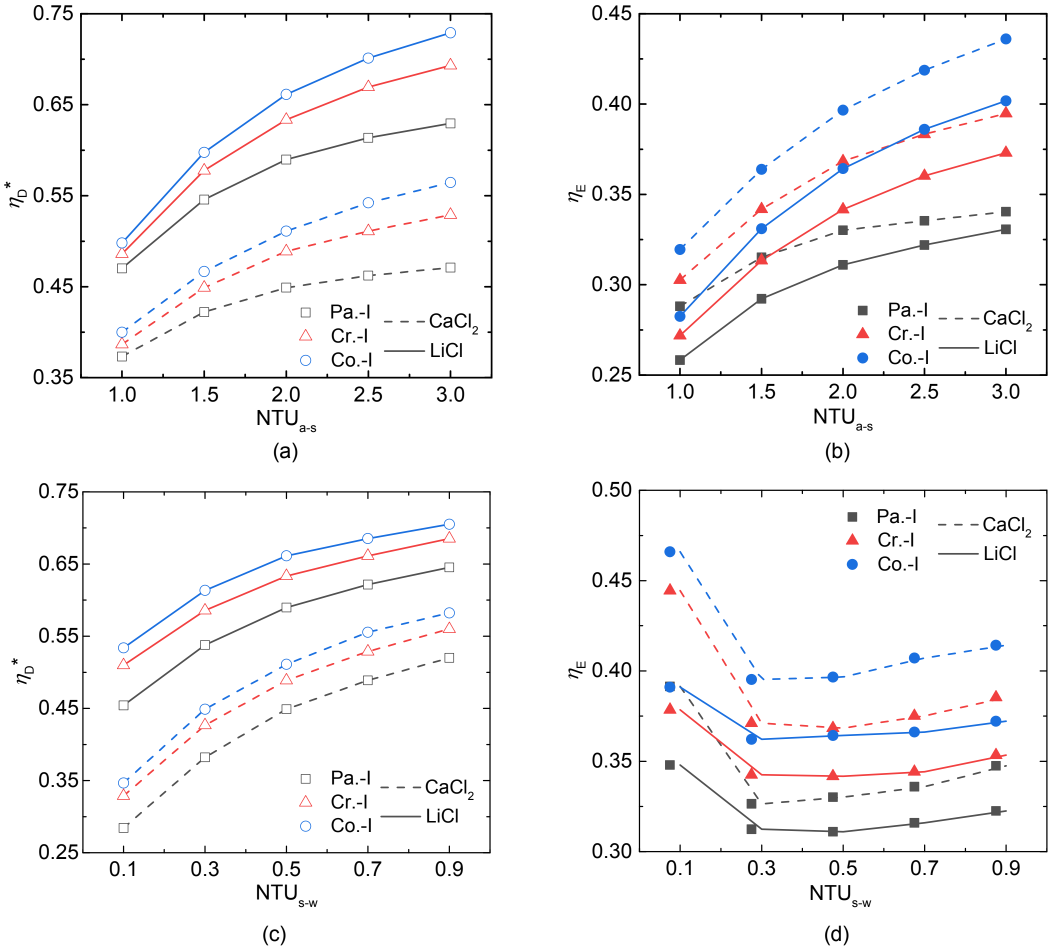
(a) Dehumidification efficiencies in parallel, cross, and counter flow; (b) Exergy efficiencies in parallel, cross, and counter flow; (c) Dehumidification efficiencies in I and II flow; (d) Exergy efficiencies in I and II flows

between the solution and the cooling water ($\mathrm{NTU}_{\mathrm{s\text{-}w}}$) is determined by the heat transfer area of the cooling water pipe and heat transfer coefficient between the solution and the cooling water, which are the two independent key factors in internally-cooled dehumidifiers whose effects are shown in Fig. 5.

1. Effect of the number of heat transfer units between air and solution ($\mathrm{NTU}_{\mathrm{a\text{-}s}}$)

With the $\mathrm{NTU}_{\mathrm{a\text{-}s}}$ varying from 1.0 to 3.0, the relationship between $\mathrm{NTU}_{\mathrm{a\text{-}s}}$ and dehumidification efficiency is shown in Fig. 5a. The dehumidification efficiency and the difference of the efficiencies grow with the increase in $\mathrm{NTU}_{\mathrm{a\text{-}s}}$, and the rates of increase in dehumidification efficiencies gradually decline. The dehumidification efficiency of LiCl solution increases by 33.9% in Pa., by 42.6% in Cr., and in Co. by 46.4%, while that of CaCl_2 solution in those three flow types shows sequential increases of 26.2%, 36.8%, and 41.1%. That means the increase in $\mathrm{NTU}_{\mathrm{a\text{-}s}}$ enhances the heat and mass transfer performance between air and solution, which leads to a higher dehumidification efficiency in the dehumidifiers.

Fig. 5b reveals the relationship between $\mathrm{NTU}_{\mathrm{a\text{-}s}}$ and exergy efficiency. Exergy efficiency shows the same trend as dehumidification efficiency with increases of 28.0%, 37.2%, and 42.2% for LiCl solutions, and 18.2%, 30.5%, and 36.5% for CaCl_2 solutions under flow types of Pa., Cr., and Co. Although exergy efficiencies of the three flow types increase with the growth of $\mathrm{NTU}_{\mathrm{a\text{-}s}}$, the rates of their increase drop. The increase of $\mathrm{NTU}_{\mathrm{a\text{-}s}}$ increases the moisture removal rate of the dehumidifier, and the outlet exergy of the air, which is also the benefit exergy, increases.



(a) (b) (c) (d)

Fig. 5 Effects of NTU on the efficiencies of internally-cooled dehumidifiers

(a) Effect of $NTU_{a\text{-}s}$ on dehumidification efficiency; (b) Effect of $NTU_{a\text{-}s}$ on exergy efficiency; (c) Effect of $NTU_{s\text{-}w}$ on dehumidification efficiency; (d) Effect of $NTU_{s\text{-}w}$ on exergy efficiency

2. Effect of the number of heat transfer units between solution and cooling water ($NTU_{s\text{-}w}$)

The relationship between $NTU_{s\text{-}w}$ and dehumidification efficiency is shown in Fig. 5c. The dehumidification efficiencies of the three flow types all rise with the increase of $NTU_{s\text{-}w}$ but the rates of rising gradually decline. When the $NTU_{s\text{-}w}$ increases from 0.1 to 0.9, the dehumidification efficiency of LiCl solution improves by 32.1%–42.1%, while $CaCl_2$ solution has 67.9%–82.8% of growth in dehumidification efficiency. The LiCl solution performs at a higher dehumidification efficiency, and a greater improvement is also shown by the $CaCl_2$ solution. The increase in dehumidification efficiency of the dehumidifier with the increase of $NTU_{s\text{-}w}$ is because the solution temperature decreases followed by enhancement of the heat transfer performance between the solution and cooling water, so that the dehumidification capacity of the solution is improved.

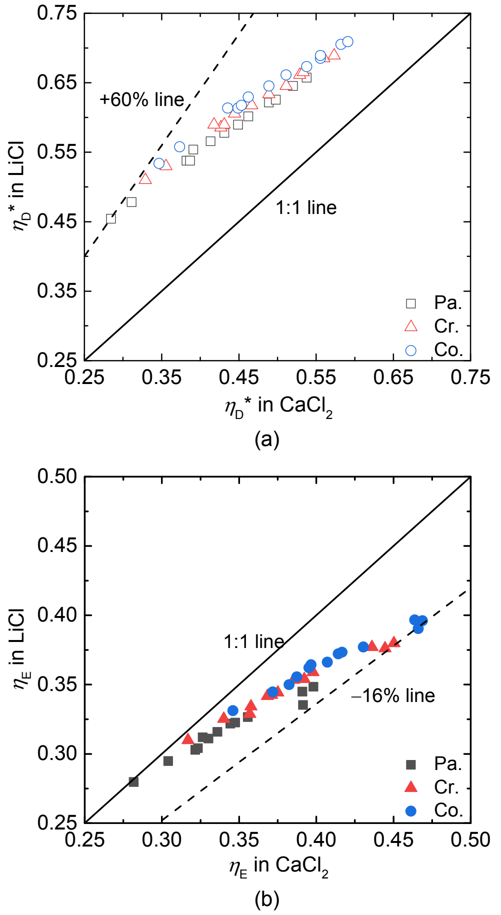
Fig. 5d illustrates the impact of $NTU_{s\text{-}w}$ on exergy efficiency. The exergy efficiencies of LiCl and $CaCl_2$ solutions firstly decline sharply and then climb slowly. With $NTU_{s\text{-}w}$ increasing from 0.1 to 0.3, LiCl exergy efficiency only decreases 0.03 while that of $CaCl_2$ decreases 0.07. The exergy efficiencies of $CaCl_2$ have greater variation than those of LiCl, which leads to the result that the exergy efficiencies of $CaCl_2$ are more susceptible to changes in $NTU_{s\text{-}w}$. This is due to the change in solution temperature having a larger effect on the water vapor partial pressure of the $CaCl_2$ solution.

The exergy efficiencies reach a minimum at $NTU_{s\text{-}w}$=0.3, and then increase slowly. When the $NTU_{s\text{-}w}$ increases from 0.1–0.3, the outlet temperature of the cooling water rises rapidly, and the inlet and outlet exergy difference of cooling water increases, which means the cost exergy increases, so the exergy efficiency decreases. When $NTU_{s\text{-}w}$ is above 0.3, the increase of $NTU_{s\text{-}w}$ has less influence on outlet cooling water temperature, but the lower solution temperature strengthens the dehumidification capacity and as a result the exergy efficiency tends to grow slowly.

5.2 Effects of the fluids parameters on the efficiencies of internally-cooled dehumidifiers

5.2.1 Effects of liquid desiccant types

A comparison of the efficiencies of internally-cooled dehumidifiers using LiCl and $CaCl_2$ aqueous solutions was carried out, with the relevant results shown in Fig. 6.



(a)

(b)

Fig. 6 Effects of liquid desiccant types on the efficiencies of internally-cooled dehumidifiers

(a) Effect of liquid desiccant types on dehumidification efficiency; (b) Effect of liquid desiccant types on exergy efficiency

As shown in Fig. 6a, the dehumidification efficiencies of a LiCl solution are higher than those of a $CaCl_2$ solution under the same conditions, and apparently increase by about 60% at most at low efficiencies. At the same concentration and temperature, the water vapor partial pressure of a LiCl solution is lower than that of a $CaCl_2$ solution, so the LiCl solution has a stronger ability to absorb moisture from air, which leads to higher dehumidification efficiency. However, the exergy efficiency of the LiCl solution is about 16% lower than that of a $CaCl_2$ solution, and as shown in Fig. 6b, higher efficiencies show sharper decreases. The value of exergy efficiency depends on the degree of imbalance. The water partial vapor pressure of $CaCl_2$ solution is closer to that of moist air than that of LiCl, leading to a smaller difference of water vapor partial pressure, which benefits exergy. Thus, due to its lower exergy loss, the $CaCl_2$ solution presents a better exergy efficiency.

5.2.2 Effects of inlet solution parameters

The effects of inlet solution parameters, including solution mass flow rate, inlet temperature, and concentration, on the dehumidification performance of the two solutions, are analyzed in this section and illustrated in Fig. 7.

1. Effect of inlet solution mass flow rate (m_s)

Fig. 7a shows the change of dehumidification efficiency as the solution mass flow rate increases from 0.05 kg/s to 0.25 kg/s. The dehumidification efficiency of the LiCl solution increases by about 15% and that of the $CaCl_2$ solution by 9.5%–14.3%. The dehumidification efficiency differences of LiCl and $CaCl_2$ solutions in three flow types increase with the increase in mass flow rates and grow in basically the same ratio. Taking the Co.-I flow, which possesses the highest dehumidification efficiency, as an example, when the solution mass flow rate is 0.05 kg/s, the dehumidification efficiency of the LiCl solution is 0.127 higher than that of the $CaCl_2$ one which increases by 27.2%. The dehumidification efficiency of a LiCl solution is higher than that of a $CaCl_2$ solution by 0.148 at a mass flow rate of 0.25 kg/s, the rising rate of that is 27.7%, which is similar to that with a low mass flow rate. In addition, the increasing trends of dehumidification efficiencies gradually slow down. These changes are due to smaller decreases in solution concentration and smaller increases in solution temperature in the dehumidification processes as the mass flow rate of the solutions increase and promote increases in dehumidification efficiency.

Fig. 7b is a graph of the change in exergy efficiency as the solution mass flow rate increases. It shows that a larger solution mass flow rate conduces to a higher exergy efficiency. In the process of

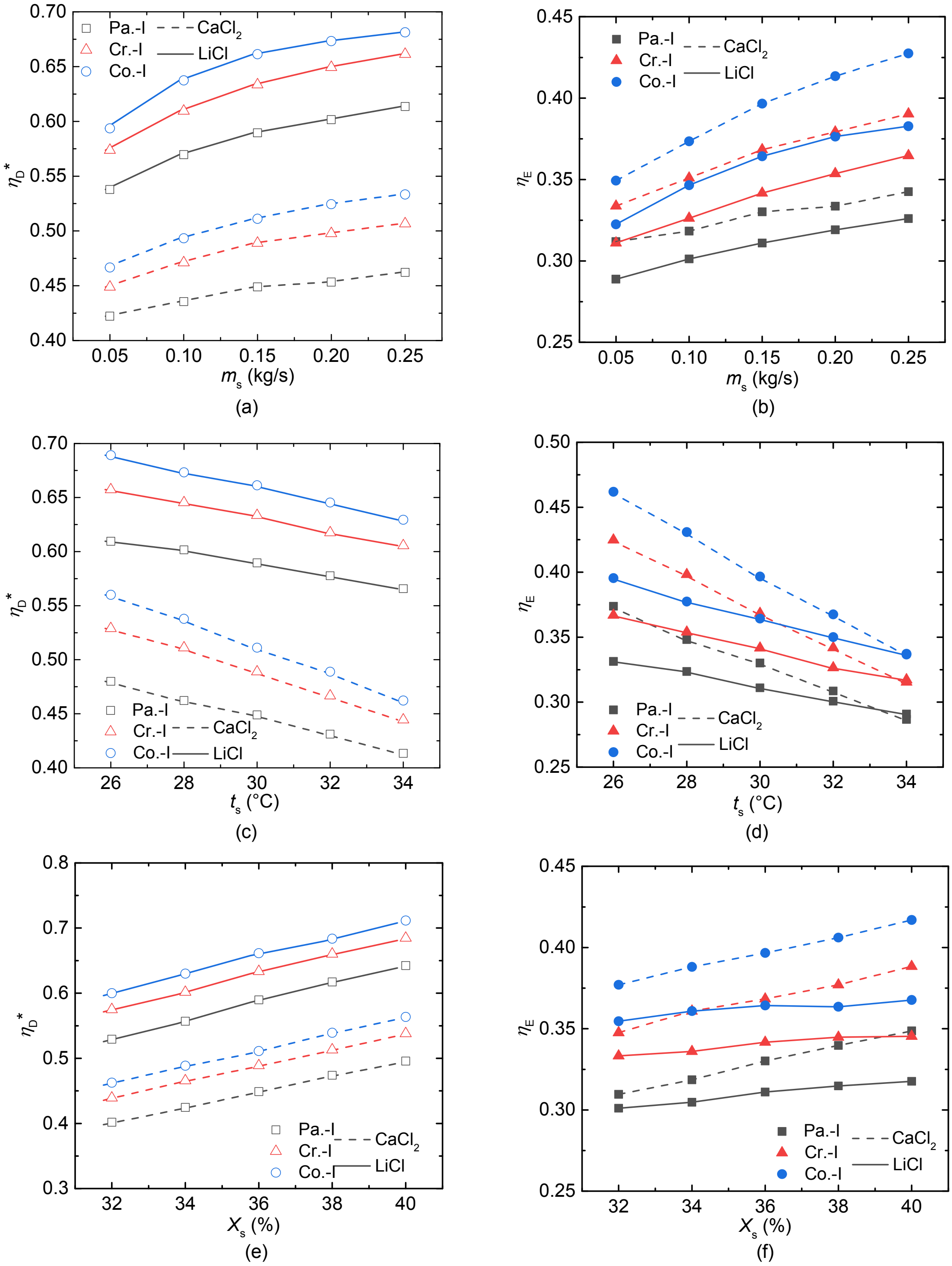


Fig. 7 Effects of inlet solution parameters on the efficiencies of internally-cooled dehumidifiers

(a) Effect of m_s on dehumidification efficiency; (b) Effect of m_s on exergy efficiency; (c) Effect of t_s on dehumidification efficiency; (d) Effect of t_s on exergy efficiency; (e) Effect of X_s on dehumidification efficiency; (f) Effect of X_s on exergy efficiency increasing solution mass flow rate, the exergy efficiency of LiCl solution in each flow type only increases in the range of 12.9%–18.7%, while the increasing ratio of different flow patterns in the $CaCl_2$ solution are distinct from each other. The exergy efficiency increases by 9.8% in Pa., by 17.0% in Cr., and by 22.4% in Co. Moreover, the exergy efficiency difference between the LiCl and $CaCl_2$ solutions in Pa. becomes smaller as the solution mass flow rate increases, but the difference in Cr. and Co. gets larger. The increase in the solution mass flow rate reduces the temperature difference between solution and cooling water, and the irreversible exergy loss decreases, leading to an increase in the exergy efficiency.

In addition, the graph shows that the exergy efficiency of the $CaCl_2$ solution is significantly higher than that of the LiCl solution because the surface water vapor pressure of $CaCl_2$ solution is higher than that of LiCl solution at the same concentration and temperature, which causes a smaller difference of water vapor pressure between a $CaCl_2$ solution and dehumidified air, and less irreversible damage during the dehumidification process.

2. Effect of inlet solution temperature (t_s)

Fig. 7c presents the relationship between dehumidification efficiency and inlet solution temperature. The dehumidification efficiency of the LiCl solution reduces by about 10% when the inlet solution temperature increases from 26 to 34 °C, whereas for the $CaCl_2$ solution, it increases by about 20%. The trend of decreasing efficiency is almost linear. The decrease in dehumidification efficiency is caused by an increase in the surface water vapor partial pressure of the solution due to the augmentation of the inlet solution temperature, and the mass transfer driving force between air and solution decreases, resulting in a decrease in dehumidification efficiency. Furthermore, the differences in dehumidification efficiency of different flow types with the same solution get smaller, thus the differences between the Pa. and Cr. and the Pa. and Co. of the LiCl solution decline from 0.08, 0.13 to 0.06, 0.11, and the dehumidification efficiency differences of different solutions in the same flow type become larger, for instance, the difference of dehumidification efficiency between LiCl and $CaCl_2$ increases from 0.13 to 0.18 in Cr.

Fig. 7d is an image of exergy efficiency and inlet solution temperature which illustrates that the exergy efficiency decreases as the inlet solution temperature increases. As the inlet solution temperatures increase from 26 °C to 34 °C, the exergy efficiency of the LiCl solution reduces by no more than 18.6%, while the $CaCl_2$ solution exergy efficiency decreases by as much as 33.2%. In Co.-I flow, the exergy efficiency of $CaCl_2$ drops from 0.46 to 0.34, compared with a decline from 0.39 to 0.34 for LiCl. The $CaCl_2$ solution is more susceptible to diverse inlet solution temperatures than the LiCl one. The exergy efficiency decreases as the inlet solution temperature increases, because of the higher inlet solution temperature which causes a greater heat transfer temperature difference between the solution and the cooling water, and thus the irreversible loss becomes larger.

3. Effect of inlet solution concentration (X_s)

The dehumidification efficiency gets higher with the increase of inlet solution concentration, as shown in Fig. 7e. As the inlet solution concentrations increase from 32% to 40%, the dehumidification efficiency of the LiCl solution increases by about 20%, while that for the $CaCl_2$ solution shows a growth of about 22%. Both solutions show similar linear growth trends. This is because a higher concentration solution has a smaller water vapor partial pressure so that the mass transfer driving force between solution and air increases, and the moisture transfer rate increases accordingly. Besides, the trends of dehumidification efficiency in the different flow types are basically similar.

Fig. 7f shows the relationship between exergy efficiency and inlet solution concentration. The exergy efficiency grows with the increase of inlet solution concentration, and the exergy efficiency of LiCl solution has a small increase of only about 5%, while the exergy efficiency of $CaCl_2$ solution, greatly affected by the concentration, increases by about 11%. This is because the water vapor partial pressure of $CaCl_2$ solution is affected more significantly by the solution concentration. The water vapor partial pressure of a solution decreases as its concentration rises and enhances the dehumidification performance and the exergy difference in dehumidified air. The exergy benefits increase more than the cost exergy, with a net growth in exergy efficiency.

5.2.3 Effects of inlet air parameters

It is assumed that the number of heat transfer unit NTU between air and solution is constant when analyzing the influences of inlet air parameters on dehumidification performance. The analyzed inlet air parameters include air mass flow rate, air temperature, and humidity ratio.

1. Effect of inlet air mass flow rate (m_a)

Fig. 8a is a graph showing the relationship between dehumidification efficiency and air mass flow rate. The dehumidification efficiency of the LiCl solution decreases by 25.1%–32.4% and that of the $CaCl_2$ solution by 37.1%–43.5%, as the air mass flow rate increases from 0.1 kg/s to 0.5 kg/s. The increase in air mass flow rate increases the heat removed by the high-temperature air and increases the solution temperature resulting in a decrease in the solution dehumidification capacity during the heat transfer process.

The relationship between air mass flow rate and exergy efficiency is shown in Fig. 8b. With the air mass flow rate increasing from 0.1 kg/s to 0.3 kg/s, the exergy efficiency of the LiCl solution increases by 9.9%–16.4%, while that of the $CaCl_2$ solution increases by 18.4%–24.3%. The exergy efficiency sharply increases at m_a=0.1–0.2 kg/s and reaches its highest value at m_a=0.3 kg/s. After that point, the efficiency slowly decreases and gradually stabilizes with no more than a 5% fall. This is because the dehumidification amount presents a small value in a low mass flow rate, and the exergy loss of cooling water is mainly applied to cool the solution and is not converted into a benefit exergy of dehumidified air.

2. Effect of inlet air temperature (t_a)

Fig. 8c shows the relationship between dehumidification efficiency and inlet air temperature. The dehumidification efficiency decreases slightly with an increase of inlet air temperature. When the inlet air temperature increases from 32 °C to 40 °C, the dehumidification efficiency reduces by 3.0%–4.6% for the LiCl solution, and by 5.9%–9.5% for the $CaCl_2$ solution. Generally, the increase of inlet air temperature has only a small influence on the dehumidification efficiency. The increase in inlet air temperature causes the solution temperature to rise during the heat transfer process, and so the water vapor partial pressure of the solution increases resulting in a decline in the dehumidification capacity of the solution. The dehumidification efficiency therefore drops.

The relationship between exergy efficiency and inlet air temperature is shown in Fig. 8d. It can be seen from the figure that the exergy efficiency of LiCl solution slightly reduces by 3.6%–7.4% as the air inlet temperature increases from 32 °C to 40 °C, while the $CaCl_2$ solution shows a drop of 13.5%–17.3% in exergy efficiency in the same circumstances. The exergy efficiencies of the LiCl solution are less affected by the inlet air temperature than those of the $CaCl_2$ solution, because of the smaller effect of LiCl solution temperature on the water vapor partial pressure compared with the $CaCl_2$ case. The difference in the exergy efficiency of the two solutions becomes smaller in three flow types. For the flow type Pa., the exergy efficiency of the LiCl solution exceeds that of the $CaCl_2$ solution as the air inlet temperature is over 38 °C. The increase in inlet air temperature causes a decline in the dehumidification rate, and a decrease in the air benefit exergy, while the outlet cooling water temperature rises, leading to an increase in cooling water exergy loss, and the combination leads to a slight drop in the exergy efficiency.

3. Effect of inlet air humidity ratio (ω_a)

Fig. 8e represents the relationship between dehumidification efficiency and the inlet air humidity ratio. The graph shows a 1.4%–2.1% increase in the dehumidification efficiency of the LiCl solution as the inlet air humidity ratio increases from 24 g/kg to 32 g/kg, while the efficiency of the $CaCl_2$ solution increases by 7.8%–8.6% showing the effect of inlet air humidity to be weak. The increase in humidity ratio of inlet air enhances the mass transfer driving force between the solution and air, thereby the dehumidification efficiency increases slightly.

As can be seen from Fig. 8f, which shows the relationship between exergy efficiency and humidity ratio of inlet air, as the inlet air humidity ratio increases from 24 g/kg to 32 g/kg, the exergy efficiencies of LiCl and $CaCl_2$ solution decline nearly linearly, and also have a similar rate of reduction of about 25%.

The augmentation of the inlet air humidity ratio increases the mass transfer driving force between air and solution, and the irreversible loss of the dehumidification process rises with the increased dehumidification of the solution, so that the exergy efficiency reduces. Figs. 8e and 8f also show that the effects of inlet air humidity ratio on dehumidification performance are similar in different flow types.

5.2.4 Effects of inlet cooling water parameters

The cooling capacity provided by cooling water has a significant effect on its dehumidification performance in internally-cooled dehumidifiers. The dehumidification performance influenced by cooling water mass flow rate and inlet temperature which are used for measuring the cooling capacity, are analyzed in this section.

1. Effect of inlet cooling water mass flow rate (m_w)

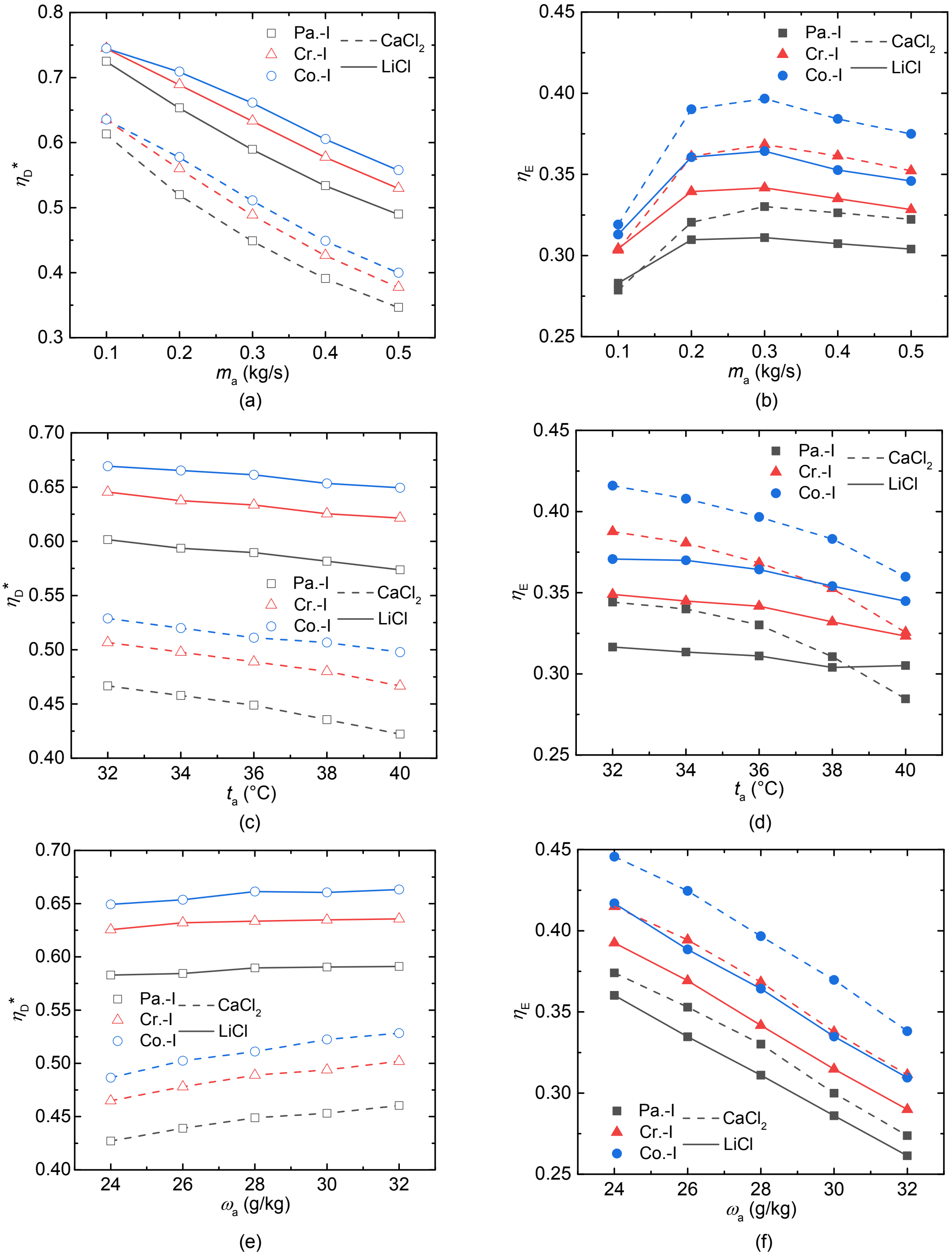


Fig. 8 Effects of inlet air parameters on the efficiencies of internally-cooled dehumidifiers

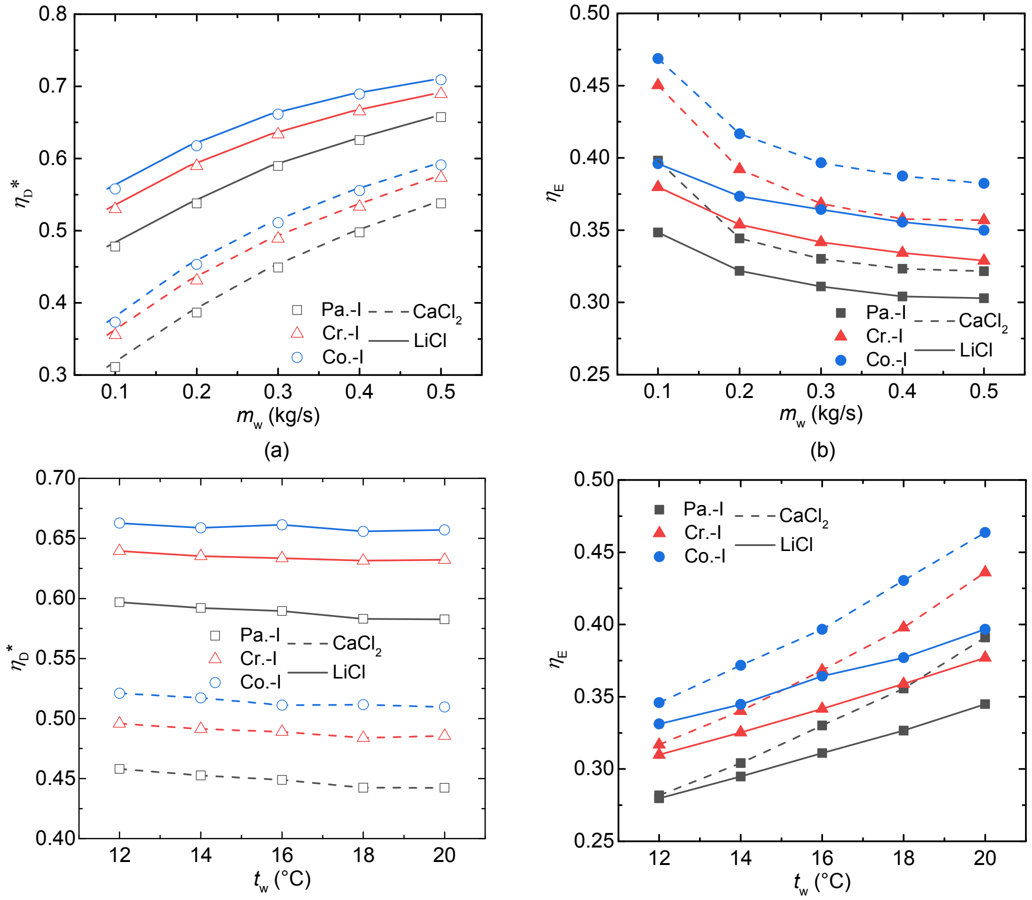
(a) Effect of m_a on dehumidification efficiency; (b) Effect of m_a on exergy efficiency; (c) Effect of t_a on dehumidification efficiency; (d) Effect of t_a on exergy efficiency; (e) Effect of ω_a on dehumidification efficiency; (f) Effect of ω_a on exergy efficiency

Figs. 9a and 9b show the effect of cooling water mass flow rate on two efficiencies. Shown in Fig. 9a, while the mass flow rate of cooling water increases from 0.1 kg/s to 0.5 kg/s, the dehumidification efficiency of the LiCl solution increases by 27.1%–37.5%, while that of the $CaCl_2$ solution increases by 58.3%–72.9%. The differences in dehumidification efficiency get smaller, whether it is different solutions of the same flow type or the same solution under different flow types. The larger cooling water mass flow rate behaves with a higher dehumidification efficiency owing to an increase in the cooling capacity provided into the device leading to a decrease in the solution temperature which increases the mass transfer driving force between air and solution.

Fig. 9b shows that exergy efficiency decreases with the increase of cooling water mass flow rate. There is a 11.6%–13.4% decrease in the exergy efficiency of the LiCl solution, while the efficiency of the $CaCl_2$ solution decreases by 18.4%–20.7%. Moreover, in parallel flow, the exergy efficiency of $CaCl_2$ drops significantly by about 13% from 0.398 to 0.344 between 0.1 and 0.2 kg/s of m_w; whereas that of LiCl drops only about 7.6%. The reducing trend of exergy efficiency becomes less as the mass flow of cooling water increases. The exergy efficiency decreases as the cooling water mass flow rate increases, because the increase in mass flow rate of cooling water restricts the temperature rise of the cooling water. The heat transfer temperature difference between solution and cooling water thus increases, resulting in an increase in the exergy loss, thereby reducing the exergy efficiency.

2. Effect of inlet cooling water temperature (t_w)

Fig. 9c shows dehumidification efficiency changes with inlet cooling water temperature. When the inlet cooling water temperature increases from 12 °C to 20 °C, the dehumidification efficiency of a



(a) (b) (c) (d)

Fig. 9 Effects of inlet cooling water parameters on efficiencies of internally-cooled dehumidifiers
(a) Effect of m_w on dehumidification efficiency; (b) Effect of m_w on exergy efficiency; (c) Effect of t_w on dehumidification efficiency; (d) Effect of t_w on exergy efficiency

LiCl solution reduces by 2.4% at most, while that of a $CaCl_2$ solution reduces by no more than 3.4%. As the inlet cooling water temperature increases, the dehumidification efficiency decreases slightly, because the increase of inlet cooling water temperature causes a reduction of the cooling capacity provided in the device and a rise in the solution temperature, so the dehumidification performance deteriorates, and dehumidification efficiency decreases slightly. The new definition of dehumidification efficiency takes the cost of cooling water into account.

Fig. 9d displays the influence of the inlet solution temperature on exergy efficiency. As the inlet cooling water temperature increases from 12 °C to 20 °C, the exergy efficiencies of the LiCl and $CaCl_2$ solutions increase by about 22% and 37%, respectively. The difference in exergy efficiency of different solutions in the same flow type grows with the rise of inlet cooling water temperature; in particular, the exergy efficiency difference between $CaCl_2$ and LiCl solution increases from 0.015 to 0.067, that is to say by 3.5 times in Co. The exergy efficiency gets higher, as the inlet solution temperature rises, because it leads to a decrease in the temperature difference between the solution and air and results in a lower exergy loss and thus an increase in exergy efficiency.

6 Conclusions

According to the energy and mass conservation among air, cooling water, and solution, mathematical models of internally-cooled dehumidifiers based on the number of transfer units under six flow types are built, and the influence of different device structures and operating parameters on dehumidification performance and efficiencies are analyzed. The main conclusions can be summarized as follows:

1. A novel definition of dehumidification efficiency, taking into account the cost of cooling water, is defined in this paper to avoid the possibility of the traditional one in internally-cooled dehumidifiers exceeding the range of 0 to 1 under certain conditions. The novel dehumidification efficiency, in which the inlet solution equilibrium humidity ratio is calculated according to the minimum temperature among the inlet solution and cooling water, is a better evaluation of internally-cooled dehumidifiers.

2. Counter flow between air and solution shows the highest dehumidification efficiency and exergy efficiency, followed by cross flow, and parallel flow shows the lowest. Flow types between solution and cooling water have little effect on efficiencies, and I flow (water flow parallel to the solution) shows an increase of about 5% compared with II flow (water flow counter to solution).

3. The dehumidification efficiency of the LiCl solution is higher than that of the $CaCl_2$ solution, and can be increased by 60%, while the exergy efficiency of the LiCl solution can be reduced by 16% compared with that of the $CaCl_2$ solution.

4. Higher $NTU_{a\text{-}s}$ should be used in the design of internally-cooled dehumidifiers. Higher solution concentration and lower inlet temperature of solution and air are recommended to achieve both superior dehumidification efficiency and exergy efficiency in internally-cooled dehumidifiers.

Contributors

Dong-gen PENG designed the research and edited the manuscript, Shun-yi LI processed the corresponding data and wrote the first draft of the manuscript, Dan-ting LUO conducted the research, Yu-ting FU revised the manuscript, Xiao-song CHENG helped organize the manuscript, and Yin LIU revised and edited the final version.

Conflict of interest

Dong-gen PENG, Shun-yi LI, Dan-ting LUO, Yu-ting FU, Xiao-song CHENG, and Yin LIU declare that they have no conflict of interest.

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中文概要

题　目：使用 **LiCl** 和 **$CaCl_2$** 溶液的内冷型溶液除湿器效率对比和性能分析

目　的：目前关于内冷型溶液除湿器的研究均基于特定的装置结构和流型，限制了其普遍应用。本文采用基于传热单元数的传热传质数学模型，通过数值模拟研究了各种流型、结构和流体参数的影响。

创新点：1. 提出一种新的除湿效率定义；2. 引入㶲效率评价内冷型除湿器的性能；3. 分析不同流型、结构和流体参数下除湿器除湿效率和㶲效率；4. 研究结论不受特定装置和流体参数的限制，为内冷型除湿器的设计优化提供了建议。

方　法：1. 建立内冷型除湿器的物理和数学模型；2. 通过实验数据与模拟数据的对比，验证模型的正确性；3. 建立除湿效率和㶲效率的数学模型；4. 通过不同流型、结构和流体参数的模拟数据，分析其对除湿效率和㶲效率的影响。

结　论：1. 本文提出了一种新的除湿效率定义，解决了内冷型除湿器在某些工况下除湿效率超出 0~1 范围的问题；溶液入口平衡含湿量根据入口溶液和冷却水的最低温度计算，并且新的除湿效率综合考虑了冷水的代价，可以更好地评价除湿器的除湿性能。2. 对于内冷型除湿器，三种空气与溶液间流型的除湿效率和㶲效率总是逆流最好，叉流其次，顺流最差；溶液与冷水之间的流型对两种效率影响不大，I 流型比 II 流型提升 5%左右。3. LiCl 溶液的除湿效率高于 $CaCl_2$ 溶液，可以提高 60%，但 LiCl 溶液的㶲效率低于 $CaCl_2$ 溶液，可降低 16%。4. 在设计内冷型除湿器时应采用较大的空气与溶液间传热单元数，并建议在其运行过程中使用浓度较高的溶液及入口温度较低的溶液和空气，以达到更高的除湿效率和㶲效率。

关键词：溶液除湿；内冷型除湿器；性能对比；除湿效率；㶲效率