A simple analytical method to estimate all exit parameters of a cross-flow air dehumidifier using liquid desiccant

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ABSTRACT

The dehumidifier is a key component in liquid desiccant air-conditioning systems. Analytical solutions have more advantages than numerical solutions in studying the dehumidifier performance parameters. This paper presents the performance results of exit parameters from an analytical model of an adiabatic cross-flow liquid desiccant air dehumidifier. Calcium chloride is used as desiccant material in this investigation. A program performing the analytical solution is developed using the engineering equation solver software. Good accuracy has been found between analytical solution and reliable experimental results with a maximum deviation of +6.63% and −5.65% in the moisture removal rate. The method developed here can be used in the quick prediction of the dehumidifier performance. The exit parameters from the dehumidifier are evaluated under the effects of variables such as air temperature and humidity, desiccant temperature and concentration, and air to desiccant flow rates. The results show that hot humid air and desiccant concentration have the greatest impact on the performance of the dehumidifier. The moisture removal rate is decreased with increasing both air inlet temperature and desiccant temperature while increases with increasing air to solution mass ratio, inlet desiccant concentration, and inlet air humidity ratio.

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column with LiBr solutions was developed by Factor and Grossman [3]. The interface temperature and concentration were assumed to be the bulk liquid temperature and concentration. Overall, heat and mass transfer coefficients were utilized. The model was validated with the experimental results. For CaCl₂, LiCl and cost effective liquid desiccant solutions (CELD), the individual phase heat and mass transfer coefficients were calculated and correlated for various packing materials [4,5]. Analytical expressions of the air and desiccant parameters in the counter flow dehumidifier are provided by Stevens et al. [6]. Within the model, the analytical solution of the air enthalpy and liquid desiccant equivalent enthalpy, which expressed the capability of the combined heat and mass transfer process, is first calculated. Then, the solutions of the air humidity ratio and desiccant equivalent humidity ratio, which expresses the capability of moisture transfer, are given. Finally, the air and liquid desiccant temperature can be calculated according to the above enthalpy and humidity ratio calculated result. A method for finding the analytical solution of the coupled heat and mass transfer performance for the dehumidifier and regenerator was reported before [7,8]. Analytical solutions of the air enthalpy and desiccant equivalent enthalpy field within the cross-flow dehumidifier/regenerator were given [9,10], where the air and desiccant are not mixed breadthwise (which means the transfer processes of the air and desiccant are both two dimensional). The enthalpy field gained from the analytical solutions compares well with numerical solutions, and the analytical enthalpy efficiency compares well with experimental results of the cross-flow dehumidifier.

Researchers [11–13] have developed mathematical models of the coupled heat and mass transfer processes in the dehumidifier or regenerator, and most of the models were solved numerically. In Liu et al. [14], an experimental study of the performance of the cross-flow dehumidifier was done, which has been less studied than the counter flow dehumidifier, although it is more applicable in practice. The moisture removal rate and dehumidifier effectiveness were adopted as the dehumidifier performance indices. The effects of the dehumidifier inlet parameters on the two indices were investigated. Correlations have been proposed to predict the cross-flow dehumidifier performance, which give results in good agreement with the present experimental findings. The results from studying the performance of a counter flow liquid desiccant dehumidifier were presented by Koronaki et al. [15]. A heat and mass transfer theoretical model of an adiabatic packed column has been developed, based on the Runge–Kutta fixed step method, to predict the performance of the device under various operating conditions. Good agreement was found between experimental tests and the theoretical model. Davoud and Meysam [16] presented a new analytical solution of heat and mass transfer processes in a packed bed liquid desiccant dehumidifier. They results revealed that design variables such as desiccant concentration, desiccant temperature, air flow rate, and air humidity ratio have the greatest impact on the performance of the dehumidifier. The liquid flow rate and the air temperature have not a significant effect. Furthermore, the effects of air and liquid desiccant flow rate have been reported on the humidity effectiveness of the column.

Heat and mass transfer coefficients were used to numerically solve most models in the literature. This paper proposed a simple analytical model of the bulk heat and mass transfer processes in a cross-flow liquid desiccant air dehumidifier. An empirical correlation for calculating the dehumidifier effectiveness introduced by Moon et al. [17] is used to perform the analytical solution of the presented model with acceptable accuracy. Comprehensively, this model is used for studying the effect of operating parameters on the whole dehumidifier performance. The analytical solution shows good accuracy when compared with reliable experimental data available in the literature.

**System description**

Based on energy and mass laws of conservations, the proposed analytical model has been developed as a tool for evaluating the performance of a cross-flow liquid desiccant air dehumidifier. This model describes rationally the bulk coupled heat and mass transfer processes taking place inside the dehumidifier. From Fig. 1, the strong desiccant solution is supplied to the dehumidifier at suitable concentration and temperature. At the same time, process air flows continuously across the dehumidifier. Due to the vapor pressure difference between air and desiccant solution, the process air is dehumidified. The dryer the exit air is, the higher the rate of water vapor absorbed by the strong desiccant solution which leads to weak desiccant solution at the dehumidifier exit. The following initial parameters should be assumed during calculation procedure: concentration and temperature of desiccant solution at dehumidifier inlet, mass flow rate of inlet desiccant, humidity ratio, temperature, and mass flow rate of inlet air.
A simple analytical method based on the nominal effectiveness values of a cross-flow liquid desiccant air dehumidifier is introduced in this investigation. The schematic diagram of the control volume of the desiccant dehumidifier is shown in Fig. 1. In order to simplify the complexity of the governing equations, the following assumptions are used in the calculations based on the available heat and mass transfer models: adiabatic cross-flow air dehumidifier, steady-state operation, the dehumidifier effectiveness is used as a controlling variable in the calculation procedure, equilibrium properties of air are calculated at the same conditions of the desiccant solution in the interface area. The desiccant solution properties at the interface area are calculated at the average conditions across the dehumidifier. The bulk heat and mass transfer balance equations which link air and desiccant solution properties across the dehumidifier are introduced as follows:

**Energy balance equation across the dehumidifier**

\[
\dot{m}_d (h_{a1} - h_{a2}) = \dot{m}_a (h_{s2} - h_{s1}) + \dot{m}_a (y_{a1} - y_{a2}) h_{fg} \tag{1}
\]

where \(\dot{m}_d\) is the mass flow rate of air in kg/s, \(h_{a1}\) and \(h_{a2}\) are the inlet and exit air enthalpy across the dehumidifier, respectively, in kJ/kg, \(h_{s1}\) and \(h_{s2}\) are the inlet and exit solution enthalpy across the dehumidifier, respectively, in kJ/kg, \(h_{fg}\) is the latent heat of vaporization in kJ/kg, \(y\) is the humidity ratio across the dehumidifier, respectively, in kg H2O/kg air.

**Mass balance equation for the desiccant solution**

Since the mass of the desiccant material is constant during the absorption process, the following equation can be written as:

\[
\dot{m}_a X_1 = \dot{m}_a X_2 \tag{5}
\]

where \(\dot{m}_a\) and \(\dot{m}_a\) are the inlet and exit solution mass flow rate across the dehumidifier, respectively, in kg/s, and \(X_1\) and \(X_2\) are the inlet (strong) and exit (weak) concentration across the dehumidifier, respectively, in kg H2O/kg.

**Mass balance equation for air water vapor**

The rate of water vapor condensed from the process air and absorbed by the strong desiccant solution inside the dehumidifier, referred as moisture removal rate (MRR), is given by:

\[
\dot{m}_{water} = MRR = \dot{m}_a (y_{a1} - y_{a2}) = \dot{m}_a \Delta y_a \tag{6}
\]

where \(\dot{m}_{water}\) is the rate of water condensed by the dehumidifier in kg/s. The rate of water vapor condensed from the process air is transferred to the desiccant solution by process known as absorption. Simply, the condensation rate represents the amount by which the desiccant solution is diluted. So, Eq. (5) can be formulated as follows:

\[
\dot{m}_a X_1 = (\dot{m}_a + \dot{m}_a (y_{a1} - y_{a2})) X_2 \tag{7}
\]

With little arrangements, Eq. (7) can be written as follows:

\[
X_2 = \frac{1}{(1 + \frac{\dot{m}_a}{\dot{m}_a} \Delta y_a)} X_1 \tag{8}
\]

The most common performance measures for evaluating the dehumidifier potential to dehumidify the process air are both humidity and temperature effectiveness. An empirical correlation of the humidity effectiveness (\(e_y\)) has been given by Moon et al. [17]. Also, \(e_y\) is introduced as follows:

\[
e_y = \frac{y_{a1} - y_{a2}}{y_{a1} - y_{eq}} \tag{9}
\]

where \(e_y\) is the dehumidifier humidity effectiveness based on the air humidity ratio change, and \(y_{eq}\) is the humidity ratio of air in equilibrium with CaCl2 solution at the interface area. It is calculated from the following equation:

\[
y_{eq} = \frac{0.622 p_r}{1.013 \times 10^5 - p_r} \tag{10}
\]

where \(p_r\) is the partial vapor pressure on the desiccant solution surface in Pa. Also, the dehumidifier thermal effectiveness (\(e_T\)) based on air temperature change across the dehumidifier is given as follows:

\[
e_T = \frac{T_{a1} - T_{a2}}{T_{a1} - T_{eq}} \tag{11}
\]

where \(T_{a1}\) and \(T_{a2}\) are the inlet and exit air temperature across the dehumidifier, respectively, in °C and \(T_{eq}\) is the temperature
of air which in thermal equilibrium with CaCl₂ solution at the interfacial area in °C, and it is assumed to be equal to the desiccant solution temperature \( T_s \).

The partial vapor pressure on the surface of CaCl₂ solution \( (p_a) \) in mm Hg is calculated using the correlations introduced by Gad et al. [19]. Constants of Eq. (12) and its operating range are shown in Table 1.

\[
\ln(p_a) = (a_o + a_1 X) - \frac{(b_o + b_1 X)}{T_s + C} 
\]  

(12)

The above mentioned analysis shows the dependence of the absorption process, air dehumidification, on operational parameters such as air inlet humidity and temperature, inlet concentration and temperature of the desiccant solution, and air to desiccant solution mass flow rates. The proposed mathematical model is constituted from coupled algebraic equations integrated with the correlation from Moon et al. [17]. A program for the analytical solution is developed using the engineering equation solver software. The inlet parameters for both air and desiccant solutions are introduced into the program, and then, the exit parameters of the desiccant solution and process air are calculated.

Validation of mathematical model

Before evaluating the effect of various operating parameters on the performance of the adiabatic air dehumidifier, the validation of the developed analytical model should be achieved. For this purpose, reliable experimental data from Moon et al. [17] were selected. A plot digitizer program is used to extract point data from Moon et al. [17]. The obtained inlet desiccant concentrations from the plot digitizer are fed to the presented model, and the results are shown in Fig. 2. According to these results, good agreement between the experimental data of Moon et al. [17] and the analytical results of present study is achieved. In all cases, the most of predicted values for MRR are higher than the experimental values, and the discrepancy may be due to the assumptions made in the analysis. However, the maximum deviation in MRR is +6.63% and −5.65%.

Results and discussion

After the validation of the analytical model with the experimental results, an extensive theoretical investigation was conducted to examine the effect of various operating parameters on the adiabatic dehumidifier performance. The parametric study includes the effect of air inlet humidity ratio and temperature, air to solution mass ratio, inlet desiccant concentration, and temperature on the exit dehumidifier parameters. Table 2 provides the operating conditions considered for all cases in the parametric analysis. The effect of each five parameter is studied, while the other parameters are held constant.

Effect of inlet air humidity ratio

The effect of inlet air humidity ratio \( (y_{a1}) \) on the moisture removal rate, dehumidifier effectiveness (MRR, \( \varepsilon_e \); respectively), and the exit parameters from the dehumidifier; air humidity ratio, air temperature, solution concentration, and solution temperature \( (y_{a2}, T_{a2}, X_{s2}, \text{and } T_{s2}, \text{respectively}) \) is shown in Fig. 3. As illustrated, when the inlet air humidity ratio is increased, MRR, \( y_{a2} \), and \( T_{a2} \) are increased, while \( \varepsilon_e \), \( T_{s2} \), and \( X_{s2} \) show no significant effect. To a great extent, the partial vapor pressure is the governing factor of the mass transfer occurs between process air and desiccant solution. As the inlet air humidity ratio increases, the partial vapor pressure of air also increases which in turn enhances the difference between the partial vapor pressure in the inlet air-stream and that on the desiccant solution surface resulting in an increase in the moisture absorbing capacity of desiccant solution. This increase leads to high moisture removing capacity. On the other hand, as \( y_{a1} \) is increased the increase in the numerator of Eq. (9) offsets, the increase in the denominator of the same equation results in slight decrease in the dehumidifier effectiveness. This in turn increases the exit humidity ratio \( y_{a2} \). Increasing \( y_{a1} \) in turn increases the enthalpy of air at the dehumidifier inlet which rises the temperature of the solution at the exit. When \( y_{a1} \) is increased from 0.016 to 0.024 kgv/kgda, MRR, \( y_{a2} \), and \( T_{a2} \) are increased by 67.29%, 39.22%, and 13.39%, respectively.

Effect of inlet air temperature

Fig. 4 shows the effect of inlet air temperature \( (T_{a1}) \) on the MRR, \( \varepsilon_e \) and the exit parameters from the dehumidifier; \( y_{a2} \), \( T_{a2}, X_{s2}, \text{and } T_{s2} \). As \( T_{a1} \) is increased, both MRR and \( \varepsilon_e \) are

| Equation constants | Operating range |
|--------------------|-----------------|
| \( a_o = 10.0624, a_1 = 4.4674, b_o = 739.828, b_1 = 1450.96, C = 111.96 \) | \( T = 10–65 \, (^{\circ}C) \); \( X = 0.2–0.5 \, (\text{kgv/kgda}) \) |
| \( a_o = 19.786, a_1 = 1.21507, b_o = 4758.1735, b_1 = 1492.5857, C = 273 \) | \( T = 60–100 \, (^{\circ}C) \); \( X = 0.2–0.5 \, (\text{kgv/kgda}) \) |

Fig. 2 Comparison of MRR at different \( X_1 \) between present study and Moon et al. [17].

Fig. 3 Comparison of MRR, \( \varepsilon_e \), and air parameters at different inlet air humidity ratio and temperature between present study and Moon et al. [17].
Table 2  Operating conditions for the cases considered in the parametric analysis.

| Cases      | $m_a/m_s$ | $y_{a1}$ (kg$_v$/kg$_{da}$) | $T_{a1}$ (°C) | $X_1$ (kg$_d$/kg$_s$) | $T_{s1}$ (°C) |
|------------|-----------|-----------------------------|---------------|------------------------|---------------|
| Fig. 2     | 0.64      | 0.02157                     | 30            | 0.33–0.43              | 30            |
| Fig. 3     | 1         | 0.016–0.026                 | 40            | 0.43                   | 20            |
| Fig. 4     | 1         | 0.018                       | 26–40         | 0.43                   | 20            |
| Fig. 5     | 1         | 0.018                       | 40            | 0.33–0.43              | 20            |
| Fig. 6     | 1         | 0.018                       | 40            | 0.43                   | 26–36         |
| Fig. 7     | 0.25–2    | 0.018                       | 40            | 0.43                   | 20            |

Fig. 3  Effect of $y_{a1}$ on dehumidifier parameters (MRR, $\varepsilon_y$, $y_{a2}$, $T_{a2}$, $T_{s2}$, $X_2$).

Fig. 4  Effect of $T_{a1}$ on dehumidifier parameters (MRR, $\varepsilon_y$, $y_{a2}$, $T_{a2}$, $T_{s2}$, $X_2$).
decreased, but $T_{a2}$, $y_{a2}$, and $T_{s2}$ are increased, while $X_2$ has no significant change. This may be explained as follows: as the inlet process air temperature is increased, the temperature of the desiccant solution inside the dehumidifier is increased which in turn increases $T_{s2}$, $Ta_2$ and the partial vapor pressure on the desiccant surface. When the desiccant surface vapor pressure increases, the potential of the absorption process is decreased causing air to become more humid (i.e., low $D_{ya}$) and in turn low MRR. On the other hand, the reduction in $D_{ya}$ is greater than the decrease in $(y_{a1} - y_{eq})$ which leads to low $e_y$. When $T_{a1}$ is increased from 26°C to 40°C, both MRR and $e_y$ are decreased by about 11.6% and 11.8%, respectively, but $T_{a2}$, $y_{a2}$, and $T_{s2}$ are increased by a percentage of 31.58%, 9.5%, and 6.8%, respectively.

Effect of inlet desiccant concentration

Fig. 5 shows the effect of inlet desiccant concentration ($X_1$) on the MRR, $e_y$ and the exit parameters from the dehumidifier; $y_{a2}$, $T_{a2}$, $X_2$, and $T_{s2}$. When $X_1$ is increased, MRR, $T_{a2}$, and $X_2$ are increased, but the dehumidifier effectiveness and $T_{a2}$ are slightly changed. When $X_1$ increases, vapor pressure on the desiccant surface is reduced leading to low $y_{a2}$ which in turn increases MRR. As shown from Table 2, the inlet air temperature is higher than the inlet temperature of the desiccant solution resulting in high $T_{s2}$. Both $y_{a2}$ and $y_{eq}$ are decreased but at different rates which means that the numerator of Eq. (9) is to some extent smaller than its denominator; so, $e_y$ is slightly reduced. Increasing $X_1$ from 0.33 to 0.43, MRR, $T_{a2}$,
and $X_2$ are increased by 39.13%, 10.33%, and 30%, respectively. On the other hand, both $y_{a2}$ and $\varepsilon_2$ are decreased by a percentage of 15.64% and 1.66%, respectively.

Effect of inlet desiccant temperature

Fig. 6 shows the effect of inlet desiccant temperature ($T_{a1}$) on the MRR, $\varepsilon_1$, and the exit parameters from the dehumidifier; $y_{a2}$, $T_{a2}$, $X_2$, and $X_2$. When $T_{a1}$ is increased, $y_{a2}$, $T_{a2}$, $\varepsilon_2$, and $X_2$ are increased, however, the MRR is decreased and $X_2$ is unaffected. Increasing $T_{a1}$ increases the vapor pressure on the desiccant surface which in turn decreases the moisture absorption from the process air, and hence, MRR is decreased but $y_{a2}$ increases. When $T_{a1}$ increases, the difference between $(y_{a1} - y_{a2})$ is more than that of $(y_{a1} - y_{a2})$ which in turn increases $\varepsilon_2$ (see Eq. (9)). Increasing $T_{a1}$ from 26°C to 36°C results in increasing $y_{a2}$, $T_{a2}$, $\varepsilon_2$, and $X_2$ by 28.61%, 16.92%, 22.88%, and 12.2%, respectively, but MRR is decreased by about 48.92%.

Effect of air to solution mass ratio

Fig. 7 shows the effect of air to solution mass ratio ($m_a/m_s$) on the MRR, $\varepsilon_1$, and the exit parameters from the dehumidifier; $y_{a2}$, $T_{a2}$, $X_2$, and $X_2$. When $m_a/m_s$ is increased, both MRR and $T_{a2}$ are increased, but $\varepsilon_1$ is decreased, while $T_{a2}$ and $y_{a2}$ are slightly increased, however, $X_2$ is slightly decreased. The potential capacity of the desiccant solution to carry over moisture from the process air is reduced by increasing $m_a/m_s$ results in higher outlet $y_{a2}$ which in turn reduces $\varepsilon_2$. Increasing the mass flow rate of air leads to high heat capacity of air compared to solution which offset the temperature increase in air-stream. Increasing $m_a/m_s$ by 400% results in an increase in both MRR and $T_{a2}$ by 611% and 81.6%, respectively. On the other hand, $\varepsilon_1$ is decreased by about 11.1%.

Conclusions

Air dehumidification by using CaCl$_2$ desiccant solution in a cross-flow liquid desiccant dehumidifier is studied by proposing a simple analytical model. The developed analytical model shows an excellent agreement with the available experimental data from Moon et al. [17]. Thus, for a detailed study of the absorption process, this model gives accurate performance prediction, minimizing the use of calculation and assumptions. Operating variables found to have the greatest impact on the dehumidifier performance. The following conclusions from the analytical results can be summarized: The moisture removal rate is decreased with increasing both air inlet temperature and desiccant temperature while increases with increasing $m_a/m_s$, $X_1$, and $y_{a1}$. The dehumidifier effectiveness increases with the increase of $T_{a1}$, while it decreases with the increase of $T_{a1}$ and $m_a/m_s$. Increasing $T_{a1}$, $T_{a2}$, and $y_{a2}$ results in higher $y_{a2}$, however, low exit humidity ratio is obtained at lower inlet desiccant concentration. The exit desiccant solution concentration remains unaffected by changing different operating parameters except $X_1$.

Conflict of interest

The author has declared no conflict of interest.

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