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Dynamic modeling and validation of a liquid desiccant cooling and dehumidification system

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Abstract

In this study, a simplified dynamic model for the Liquid Desiccant Cooling and Dehumidification System (LDCDS) is developed from a control viewpoint based on the laws of conservation of energy and mass. The complete LDCDS consists of three subsystems, namely the cooling coil, dehumidifier and cooler in which the models can be estimated separately and combined to obtain the model of LDCDS. The heat and mass transfer rates in model are derived through effectiveness-NTU and hybrid modeling approaches. The parameters of the thermal and moisture dynamic models are pre-identified by using the Levenberg-Marquardt method with static experimental data from the LDCDS pilot plant and then refined by adopting an unscented Kalman filter algorithm with dynamic experimental data. Detailed experimental tests on a pilot plant reveal that the proposed model accurately predicts the system performance under different operating conditions. The proposed model is expected to be applied in further research on the effects of more advanced control and optimization algorithms with the system energy efficiency.

Keywords: Cooling coil, dehumidifier, cooler, dynamic model, UKF

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1. Introduction

The major task of the Heating, Ventilating and Air Conditioning (HVAC) systems in buildings corresponds to providing a comfortable indoor environment, and consumes a significant amount of energy. In the United States, approximately 50% of building energy is consumed by HVAC systems, while this percentage reaches 70% in Singapore [1, 2]. It is inevitable to improve energy efficiency through technology upgrading. Air cooling and dehumidification is essential for air-conditioning, especially in hot and humid regions. The traditional method includes forcing the moist air through a low temperature chilled water coil system. However, this involves several disadvantages such as low dehumidification efficiency, energy waste due to the need to reheat overcooled air, and inability to achieve a lower humidity [3, 4].

The Liquid Desiccant Dehumidification System (LDDS) is considered as an alternative method for air dehumidification since it exhibits a strong ability to deal with latent loads, high energy efficiency and can be driven by low grade thermal energy [5, 6, 7, 8]. However, a system that uses only LDDS to achieve air cooling and dehumidification will have a faster dilution rate of the desiccant solution and require a higher desiccant regeneration frequency. As a result, the total energy consumption of LDDS increases due to the high energy consuming desiccant solution regeneration process in regenerator [9, 10]. To solve the fore-mentioned problems and realize high efficiency independent air temperature and humidity control, a novel Liquid Desiccant Cooling and Dehumidification System (LDCDS) is developed. The system combines a cooling coil and a liquid desiccant dehumidifier. The performance of LDCDS is directly affected by the dynamic characteristics of heat and mass transfer process [11]. Therefore, the dynamic model of LDCDS is required for a better understanding of the transient and steady-state characteristics of LDCDS and to design and simulate various control strategies of LDCDS under different operating conditions.

Several studies examined the modeling of the cooling coil. These models are mainly classified into reference models and simplified models [12]. Chow

[13] developed a fundamental dynamic model of a cooling coil based on dry and wet coil models [14] by using a finite difference control volume conservation approach. Yao et al. [15, 16] developed a dynamic model and a state-space model of a cooling coil, the models were validated by experiments and the effects of the air and water inlet parameters on transient behaviors of heat exchanger were investigated. Kusiak et al. [17] established an air handling unit model by adopting a data mining algorithm, and the function mapping between the system output and input variables was identified by a neural network algorithm. Abdul and Farrokh [18] developed thermal dynamics of the air handling unit through the energy conservation equations. The parameters were estimated from experimental data. They also established some air handling unit models by using artificial neural network, transfer function, state-space and autoregressive exogenous methods [19]. The reference models are generally complex and detailed and often involve partial differential equations that introduce high computational efforts. These models are only suitable for numerical simulation and system design. The simplified models include fewer ordinary differential equations, but some key parameters are often ignored or need to be determined in advance.

A wide range of literatures about modeling of the dehumidifier in LDDS also have been published. These models can be divided into finite difference models, effectiveness-NTU (Number of Transfer Units, ε -NTU) models and empirical models [20]. Oberg and Goswami [21] developed a finite difference model of the dehumidifier to research the system performance, and conducted a detailed experimental validation. Fumo and Goswami [22] improved this model by using a modified transfer surface. Chen et al. [23] modified the $NTU - Le$ model of the dehumidifier by setting a few parameter ranges of the process air and desiccant solution, and proposed a approach to calculate the coupled coefficients of the heat and mass transfer. Wang et al. [24] proposed a dynamic model of a dehumidifier by using a quantified thermal mass of packing and a new correlation for the heat and mass transfer coefficients. Steven et al. [25] modified the cooling coil effectiveness model and defined the NTU and derived an ef-

fectiveness model of liquid desiccant heat/mass exchanger. Ren [26] developed analytical expressions to improve the ε -NTU model by applying perturbation technique. Langrond et al. [27] adopted the ε -NTU model of a dehumidifier to study the heat and mass transfer performance. Liu et al. [28] fitted two empirical correlations of enthalpy and moisture effectiveness of a cross-flow liquid desiccant dehumidifier to calculate the system outputs. Rahimi et al. [29] developed a non-isothermal model of a packed-bed dehumidifier to research the effect of different empirical correlations on the model prediction. Both the former models require cumbersome iterative computation and system geometric information, and these disadvantages lead to the lack of these models in application of real-time control and optimization. Although empirical models do not need iterative calculations, but the model parameters are fitted under certain operating conditions, and thus the generalization capability of this model is low.

To improve the shortcomings of these modeling methods, the hybrid modeling approach [30, 31] is adopted. The hybrid model is derived from physical or thermodynamic principles while the model parameters are identified by operation or catalogue data. In this study, to research the heat and mass transfer process in LDCDS, a dynamic model is proposed by using hybrid modeling and the ε -NTU modeling approach [32, 33]. The dynamic model of the whole system is expressed by six energy and mass conservation equations. The system dimensional terms and fluids' thermodynamic parameters are lumped as constant coefficients, and thus the proposed model contains fewer unknown model parameters and is determined by using the Levenberg–Marquardt Method (LM-M) and Unscented Kalman Filter (UKF) algorithm with respect to the experimental data. Once the parameters are determined, the outputs of the system are directly calculated from the system inputs. Furthermore, the dynamic model of LDCDS is easily represented as a state-space model that is helpful in future optimization and fault diagnosis research.

2. Working principles of LDCDS

The LDCDS is to cool and dry the process air. Additionally, to maintain the continuous dehumidification in LDCDS, a desiccant regeneration system is required to concentrate the dilute desiccant solutions from the LDCDS. The schematic diagrams of a basic LDCDS and desiccant regeneration system are shown in Fig. 1. The complete LDCDS mainly consists of cooling coil, dehumidifier, cooler, pump and fan, and the desiccant regeneration system consists of heat pipe heat-exchanger, regenerator, heater, pump and fan.

Figure 1: Schematic diagram of a basic (a) LDCDS, (b) desiccant regeneration system.

The cooling and dehumidification processes in LDCDS represented in psychrometric chart are shown in Fig. 2(a), and the variations in the surface water vapor pressure of desiccant solution in the LDCDS are depicted in Fig. 2(b). The operating principles of LDCDS are briefly described below:

- The ambient air (state 1 in Fig. 2(a)) and chilled water are forced by the centrifugal fan and water pump, respectively, and they follow a counter-flow configuration in cooling coil. The process air is cooled and dehumidified to state 3. The cooling load is transferred through the metal pipe from the process air into chilled water. The process is described as

Figure 2: (a) Cooling and dehumidification processes in LDCDS represented in psychrometric chart, (b) variations in the desiccant surface water vapor pressure in LDDS.

curve line 1-2-3 in Fig. 2(a) and can be divided into the following two sub-processes: (a) sensible cooling (process 1–2), the air temperature decreases while its humidity ratio remains unaltered (relative humidity rises); (b) simultaneous sensible cooling and dehumidification (process 2–3), the air temperature continues to decrease and the humidity ratio begins to decrease (the relative humidity equals 100%), and the moisture in the air then condenses.

- The high concentrated desiccant solution (state A) is cooled by chilled water in cooler prior to entering the dehumidifier, and the surface vapor pressure and temperature of the desiccant solution decreases to state B. Following this, the solution is sprayed from the top of dehumidifier.
- In the dehumidifier, the dripping down desiccant solution wets the structured packing and forms a liquid film. The cooled air flows in a counter-flow configuration with the desiccant solution in direct contact with the liquid film. The heat and mass transfer process occurs due to the temperature and water vapor pressure difference between the process air and desiccant solution. The process air is cooled and dehumidified again and leaves from the top of dehumidifier, while the desiccant solution becomes

dilute and warm and drops down into the working solution tank. The process is depicted by line 3–4 in Fig. 2(a) and line B-C in Fig. 2(b).

- To maintain the continuous dehumidification process in dehumidifier, the diluted desiccant solution is need to be concentrated. In the regenerator, the weak desiccant solution is pumped into a heater and heated (C–D) to obtain a higher surface vapor pressure. Subsequently the hot desiccant solution and process air enters the regenerator in a counter-flow configuration. The water vapor pressure of the process air is lower than that of the desiccant solution, and thus the moisture moves from the desiccant solution into the process air. The weak desiccant solution is concentrated. The process is depicted by line D-A in Fig. 2(b).

3. Model development

The objective of LDCDS is to provide fresh air with specified temperature and humidity through remove the cooling load from the process air into chilled water and desiccant solution. Additionally, the system consumes high energy, and the energy efficiency can be improved by regulating the mass flow rates of working fluids. Therefore, an accurate model is indispensable. The model of LDCDS is divided into two categories: the heat transfer model and the mass transfer model. The cooling coil, dehumidifier and cooler are treated as independent control volumes, and a schematic diagram of heat and mass transfer process in LDCDS is illustrated in Fig. 3.

Figure 3: Schematic diagram of heat and mass transfer processes in the LDCDS.

3.1. Heat transfer model

In the LDCDS, the heat transfer process includes the following three parts: heat transfer between the process air and chilled water in the cooling coil, heat transfer between the process air and desiccant solution in the dehumidifier and heat transfer between the desiccant solution and chilled water in the cooler. Meanwhile, the chilled water, desiccant solution and process air are mechanically moved by pump and fan, respectively. Thus, the heat transfer mechanisms correspond to forced convection.

3.1.1. Cooling coil

In the cooling coil, the cooling load transfers from the process air into the chilled water. The forced chilled water flows from inlet to outlet of cooling coil with mass flow rate $\dot{m}_w$ and inlet temperature $T_{w,i}$, and its outlet temperature rises to $T_{w,cc}$. The forced process air passing through the cooling coil with mass flow rate $\dot{m}_a$, inlet temperature $T_{a,i}$ and humidity $\omega_{a,i}$, and its outlet temperature and humidity drops to $T_{a,cc}$ and $\omega_{a,cc}$, respectively. According to the $\varepsilon - NTU$ method, the heat transfer rate Q_{cc} between air and chilled water can readily be determined by the heat transfer effectiveness ε_{cc} and the maximum possible heat transfer rate Q_{max} as

$$\begin{aligned} Q_{cc} &= \varepsilon_{cc} Q_{max} = \varepsilon_{cc} C_{cc,min} (T_{a,i} - T_{w,i}) \\ C_{cc,min} &= \min\{\dot{m}_a c_a, \dot{m}_w c_w\} \end{aligned} \quad (1)$$

where $C_{cc,min}$ is the minimum heat capacity rate between air and chilled water, c_w and c_a are the specific heat capacity of the chilled water and process air, respectively.

The energy conservation diagram of the cooling coil is given in Fig. 4. The thermal dynamics of the process air and chilled water can be formulated as

$$M_{w,cc} c_w \dot{T}_{w,cc} = \varepsilon_{cc} C_{cc,min} (T_{a,i} - T_{w,i}) + \dot{m}_w c_w (T_{w,i} - T_{w,cc}) \quad (2a)$$

$$\begin{aligned} M_{a,cc} c_a \dot{T}_{a,cc} &= \varepsilon_{cc} C_{cc,min} (T_{w,i} - T_{a,i}) + \dot{m}_a c_a (T_{a,i} - T_{a,cc}) \\ &+ H_{fg} (\omega_{a,i} - \omega_{a,cc}) \end{aligned} \quad (2b)$$

Figure 4: Energy conservation diagram of the cooling coil.

where $M_{w,cc}$ and $M_{a,cc}$ are the mass constants of the chilled water and process air, respectively, H_{fg} is the latent heat of vaporization.

In the cooling coil, the heat transfer effectiveness ε_{cc} can be calculated by

$$\begin{aligned} \varepsilon_{cc} &= 1 - \exp(-NTU_{cc}) \\ NTU_{cc} &= \frac{H_{cc}A_{cc}}{C_{cc,min}} \end{aligned} \quad (3)$$

where H_{cc} and A_{cc} are the overall heat transfer coefficient and heat transfer effective area of cooling coil, respectively.

As reported in [34], the overall heat transfer coefficient H_{cc} can be obtained by Eq. (4)

$$H_{cc}A_{cc} = \left(\frac{1}{h_a A_a} + \frac{1}{h_w A_w} + \frac{1}{R} \right)^{-1} \approx \left(\frac{1}{h_a A_a} + \frac{1}{h_w A_w} \right)^{-1} \quad (4)$$

where h_a , h_w , A_a and A_w are the heat transfer coefficient and area of the process air and chilled water, respectively. R is the heat resistance of the metal wall conduction and can be considered as constant due to its good thermal conductivity of the metal wall, the conduction effect can be lumped into the unknown coefficients and neglected in the following development. The heat transfer coefficients h_a and h_w can be determined by Prandtl number and Reynolds number as [35]

$$h = \frac{kC}{D} (Re)^e (Pr)^f = \frac{kC}{D} \left(\frac{\rho \nu D}{\mu} \right)^e \left(\frac{C_p \mu}{k} \right)^f \quad (5)$$

where k , D , ρ , ν and μ are the thermal conductivity, characteristic length, density, velocity and viscosity, respectively, C , e and f are unknown parameters. In

the cooling coil, the fluid properties can approximately correspond to constants when the temperature variation is small. Simplifying Eq. (5), we then obtain

$$h = b\dot{m}^e \quad (6)$$

where $b = (kC/D) (4/\pi\mu D)^e (C_p\mu/k)^f$.

From Eqs. (4) and (6), we have

$$\begin{aligned} H_{cc}A_{cc} &= \frac{b_1A_a\dot{m}_a^{e_1}b_2A_w\dot{m}_w^{e_1}}{b_1A_a\dot{m}_a^{e_1} + b_2A_w\dot{m}_w^{e_1}} \\ &= \frac{a_1\dot{m}_a^{a_3}}{1 + a_2(\dot{m}_a/\dot{m}_w)^{a_3}} \end{aligned} \quad (7)$$

where $a_1 = b_1A_a$, $a_2 = b_1A_a/b_2A_w$ and $a_3 = e_1$; b_1 , b_2 and e_1 are the unknown parameters.

Substituting Eq. (7) into Eq. (3) yields the following expression:

$$\varepsilon_{cc} = 1 - \exp\left(\frac{-a_1\dot{m}_a^{a_3}}{(1 + a_2(\dot{m}_a/\dot{m}_w)^{a_3})C_{cc,min}}\right) \quad (8)$$

In the LDCDS, since the dew point temperature of the process air is greater than the chilled water temperature, and thus the air outlet humidity of cooling coil reaches saturation ($\varphi_a = 100\%$). Subsequently, the air humidity ratio $\omega_{a,cc}$ is a function of temperature at constant pressure and can be approximated as linear to air temperature at a low temperature variation [36], we have

$$\omega_{a,cc} = f(T_{a,cc}) = \frac{0.622\varphi_a P_{a,sat}}{101.325 - \varphi_a P_{a,sat}} \quad (9)$$

where $P_{a,sat}$ and φ_a are the air saturated vapor pressure and relative humidity, respectively. The equation for calculating $P_{a,sat}$ is given in Appendix A.

3.1.2. Dehumidifier

In the dehumidifier, the inlet temperature of the desiccant solution is typically lower than the temperature of the air. Hence, the cooling load moves from the air into the desiccant solution through the phase interface. The desiccant solution flows from the top to bottom of dehumidifier, driven by the solution pump, with mass flow rate $\dot{m}_s$ and inlet temperature $T_{s,i}$, and its outlet temperature increases to $T_{s,o}$. While the process air flows from the bottom to top with inlet temperature $T_{a,cc}$, and its outlet temperature decreases to $T_{a,o}$.

Figure 5: Energy conservation diagram of the dehumidifier.

The energy conservation diagram of the dehumidifier is shown in Fig. 5. The energy balance equations of the air and desiccant solution are formulated as

$$M_{a,de} c_a \dot{T}_{a,o} = \varepsilon_{de} C_{de,min} (T_{s,i} - T_{a,cc}) + \dot{m}_a c_a (T_{a,cc} - T_{a,o}) \quad (10a)$$

$$M_{s,de} c_s \dot{T}_{s,o} = \varepsilon_{de} C_{de,min} (T_{a,cc} - T_{s,i}) + \dot{m}_s c_s (T_{s,i} - T_{s,o}) + \dot{m}_a L_w (\omega_{a,cc} - \omega_{a,o}) \quad (10b)$$

where $C_{de,min} = \min\{\dot{m}_a c_a, \dot{m}_s c_s\}$ is the minimum heat capacity rate between air and desiccant solution; $M_{s,de}$ and c_s are the desiccant solution mass constant in dehumidifier and specific heat capacity, respectively, $M_{a,de}$ and L_w are the process air mass constants in dehumidifier and water heat of evaporation, respectively.

The derivation manner detailed for the cooling coil is followed, and the overall heat transfer coefficient H_{de} is given as

$$\begin{aligned} H_{de} A_{de} &= \frac{b_3 A_{a,de} \dot{m}_a^{e_2} b_4 A_{s,de} \dot{m}_s^{e_2}}{b_3 A_{a,de} \dot{m}_a^{e_2} + b_4 A_{s,de} \dot{m}_s^{e_2}} \\ &= \frac{c_1 \dot{m}_a^{c_3}}{1 + c_2 (\dot{m}_a / \dot{m}_s)^{c_3}} \end{aligned} \quad (11)$$

where $c_1 = b_3 A_{a,de}$, $c_2 = b_3 A_{a,de} / b_4 A_{s,de}$, $c_3 = e_2$; b_3 , b_4 and e_2 are unknown parameters.

According to Eq. (11), we obtain the heat transfer effectiveness ε_{de}

$$\begin{aligned} \varepsilon_{de} &= 1 - \exp\left(-\frac{H_{de} A_{de}}{C_{de,min}}\right) \\ &= 1 - \exp\left(\frac{-c_1 \dot{m}_a^{c_3}}{(1 + c_2 (\dot{m}_a / \dot{m}_s)^{c_3}) C_{de,min}}\right) \end{aligned} \quad (12)$$

3.1.3. Cooler

In the cooler, the desiccant solution is driven by the solution pump with inlet temperature $T_{s,co}$ and its outlet temperature drops to $T_{s,i}$. The chilled water is driven by a water pump with mass flow rate $\dot{m}_{w,co}$ and inlet temperature $T_{w,i}$, and its outlet temperature rises to $T_{w,co}$.

Figure 6: Energy conservation diagram of the cooler.

The energy conservation diagram of the cooler is given in Fig. 6. The thermal dynamics of desiccant solution and chilled water are expressed as

$$M_{s,co} c_s \dot{T}_{s,i} = \varepsilon_{co} C_{co,min} (T_{w,i} - T_{s,co}) + \dot{m}_s c_s (T_{s,co} - T_{s,i}) \quad (13a)$$

$$M_{w,co} c_w \dot{T}_{w,co} = \varepsilon_{co} C_{co,min} (T_{s,co} - T_{w,i}) + \dot{m}_{w,co} c_w (T_{w,i} - T_{w,co}) \quad (13b)$$

where $C_{co,min} = \min\{\dot{m}_s c_s, \dot{m}_{w,co} c_w\}$ is the minimum heat capacity rate between desiccant solution and chilled water; $M_{s,co}$ and $M_{w,co}$ are the desiccant solution and chilled water mass constant in cooler, respectively.

The overall heat transfer coefficient H_{co} is similarly determined as H_{de} by using the same process follows

$$\begin{aligned} H_{co} A_{co} &= \frac{b_5 A_{s,co} \dot{m}_s^{e_3} b_6 A_{w,co} \dot{m}_{w,co}^{e_3}}{b_5 A_{s,co} \dot{m}_s^{e_3} + b_6 A_{w,co} \dot{m}_{w,co}^{e_3}} \\ &= \frac{d_1 \dot{m}_s^{d_3}}{1 + d_2 (\dot{m}_s / \dot{m}_{w,co})^{d_3}} \end{aligned} \quad (14)$$

where $d_1 = b_5 A_{s,co}$, $d_2 = b_5 A_{s,co} / b_6 A_{w,co}$, $d_3 = e_3$; b_5 , b_6 and e_3 are the unknown parameters.

From Eq. (14), the heat transfer effectiveness ε_{co} is expressed as

$$\begin{aligned}\varepsilon_{co} &= 1 - \exp\left(-\frac{h_{co}A_{co}}{C_{co,min}}\right) \\ &= 1 - \exp\left(\frac{-d_1\dot{m}_s^{d_3}}{(1 + d_2(\dot{m}_s/\dot{m}_{w,co})^{d_3})C_{co,min}}\right)\end{aligned}\quad (15)$$

3.2. Mass transfer model

In the LDCDS, the mass transfer process includes two parts: mass transfer between the air and chilled water in the cooling coil and mass transfer between the process air and desiccant solution in the dehumidifier. The ambient air pass through the cooling coil, the humidity of the air is saturated and can be determined by Eq. (9). In the dehumidifier, the process air flows from bottom to top of dehumidifier with inlet humidity $\omega_{a,cc}$, and the outlet humidity descends to $\omega_{a,o}$. In a manner similar to the heat transfer model derivation process, the mass transfer rate N_{de} can be determined through the input variables as

$$N_{de} = \varepsilon_{de,m}N_{max} = \varepsilon_{de,m}\dot{m}_a(\omega_{a,cc} - \omega_{equ}) \quad (16)$$

where $\varepsilon_{de,m}$ and N_{max} are the mass transfer effectiveness and the maximum mass transfer rate in dehumidifier, respectively, ω_{equ} is the equilibrium humidity ratio and it is a function of the inlet temperature and concentration of the desiccant solution.

Figure 7: Mass conservation diagram of the dehumidifier.

The mass conservation diagram of the dehumidifier the is shown in Fig. 7, the moisture dynamic equation of the process air is given as

$$M_{a,de}\dot{\omega}_{a,o} = \varepsilon_{de,m}\dot{m}_a(\omega_{equ} - \omega_{a,cc}) + \dot{m}_a(\omega_{a,cc} - \omega_{a,o}) \quad (17)$$

In mass transfer, the NTU_m is defined as [25]

$$NTU_m = \frac{H_G A_{de}}{\dot{m}_a} \quad (18)$$

where H_G is the overall gas-phase mass transfer coefficient.

According to the two-film theory, H_G can be expressed as

$$H_G = \frac{1}{1/h_g + H/k_l} \quad (19)$$

where H , h_g and h_l are the Henrys law constant, gas-phase and liquid-phase mass transfer coefficient, respectively.

In the previous study [35], the gas-phase and liquid-phase mass transfer coefficient were derived and simplified as follows:

$$h_g = b_7 \dot{m}_a^{e_4} / T_{a,cc} \quad (20a)$$

$$h_l = b_8 \dot{m}_s^{e_5} \quad (20b)$$

where b_7 , b_8 , e_4 and e_5 are the constant coefficients.

Substituting Eq. (20) into Eq. (19), we obtain

$$H_G = \frac{b_7 \dot{m}_a^{e_4}}{T_{a,cc} + \frac{b_7}{b_8} H \dot{m}_a^{e_4} \dot{m}_s^{-e_5}} \quad (21)$$

and

$$\begin{aligned} \varepsilon_{de,m} &= 1 - \exp \left(- \frac{b_7 \dot{m}_a^{e_4} A_{de}}{(T_{a,cc} + \frac{b_7}{b_8} H \dot{m}_a^{e_4} \dot{m}_s^{-e_5}) \dot{m}_a} \right) \\ &= 1 - \exp \left(\frac{-c_4 \dot{m}_a^{c_6}}{(T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7}) \dot{m}_a} \right) \end{aligned} \quad (22)$$

where $c_4 = b_7 A_{de}$, $c_5 = b_7 H / b_8$, $c_6 = e_4$, $c_7 = -e_5$.

The dynamic model of LDCDS is integrated by Eqs. (2), (10), (13) and (17). For the purpose of researching the effects of relevant parameters on the system cooling and dehumidification performance and realizing real-time control and optimization, the dynamic LDCDS model is formulated into a continuous time state-space form as:

$$\begin{aligned} \dot{X} &= A_c X + B_c u + E_c d \\ Y &= C_c X \end{aligned} \quad (23)$$

where $X = [T_{a,cc}, T_{w,cc}, T_{a,o}, T_{s,i}, T_{w,co}, \omega_{a,o}]^T$ is the system state variable, $u = [\dot{m}_a, \dot{m}_s, \dot{m}_w, \dot{m}_{w,co}]^T$ is the system control variable, $d = [T_{a,i}, T_{w,i}, T_{s,co}, \omega_{a,i}, \omega_{equ}]^T$ is the system measurable disturbance, $Y = [T_{a,o}, \omega_{a,o}]^T$ is the system output, A_c, B_c, C_c, E_c are the state matrix, input matrix, output matrix and disturbance matrix, respectively, and these matrixes can be obtained from the linearization of the dynamic differential equations. The expressions of the matrices A_c, B_c, C_c and E_c are given in APPENDIX B.

4. Parameters identification

The developed dynamic model of LDCDS contains thirteen unknown model parameters and can be determined through experimental data. In the experiment, the inputs and outputs of LDCDS are measurable. Therefore, we can enhance modeling accuracy using an advanced parameter identification algorithm. Given that the equations are nonlinear, the model parameters can be determined using an UKF algorithm [37, 38, 39] with the commissioning information. In the UKF algorithm, it is not necessary to calculate the Jacobian matrix, and the posterior mean and covariance estimation can achieve second or higher order accuracy.

In the UKF based parameters identification algorithm, the unknown model parameters are selected as the state variables, and thus the state transition matrix corresponds to an identity matrix. The state vectors are equal during the adjacent sampling time. The discrete-time state equation is given as

$$x_{k+1} = x_k + w_k \quad (24)$$

where $x_k = [a_1^k, a_2^k, a_3^k, c_1^k, \dots, c_7^k, d_1^k, d_2^k, d_3^k]^T$ is the system state vector at time k , w_k is the system process noise vector.

The input and output variables of the cooling coil, cooler and dehumidifier are selected as the input and measurement variables. The UKF algorithm directly adopts the nonlinear system model without linearization, and thus the measurement equation is formulated as:

$$y_k = h(x_k, u_k) + v_k \quad (25)$$

where $u_k = [T_{a,i}^k, T_{w,i}^k, T_{s,co}^k, \omega_{a,i}^k, \dot{m}_a^k, \dot{m}_s^k, \dot{m}_w^k, \dot{m}_{w,co}^k]^T$ is the system input vector, $y_k = [T_{a,cc}^k, T_{a,o}^k, T_{w,cc}^k, T_{w,co}^k, T_{s,i}^k, \omega_{a,o}^k]^T$ is the system measurement vector, $h(\cdot)$ is the observation function, v_k is the measurement noise vector.

The explicit finite difference method is applied solve the measurement equation, and the finite difference equations are given as follows:

$$M_{w,cc}c_w \frac{T_{w,cc} - T_{w,cc}^0}{\Delta t} = \varepsilon_{cc}C_{cc,min}(T_{a,i}^0 - T_{w,i}^0) + \dot{m}_w c_w (T_{w,i}^0 - T_{w,cc}^0) \quad (26)$$

$$M_{a,cc}c_a \frac{T_{a,cc} - T_{a,cc}^0}{\Delta t} = \varepsilon_{cc}C_{cc,min}(T_{w,i}^0 - T_{a,i}^0) + \dot{m}_a c_a (T_{a,i}^0 - T_{a,cc}^0) + H_{fg}(\omega_{a,i}^0 - \omega_{a,cc}^0) \quad (27)$$

$$M_{s,co}c_s \frac{T_{s,i} - T_{s,i}^0}{\Delta t} = \varepsilon_{co}C_{co,min}(T_{w,i}^0 - T_{s,co}^0) + \dot{m}_s c_s (T_{s,co}^0 - T_{s,i}^0) \quad (28)$$

$$M_{w,co}c_w \frac{T_{w,co} - T_{w,co}^0}{\Delta t} = \varepsilon_{co}C_{co,min}(T_{s,co}^0 - T_{w,i}^0) + \dot{m}_{w,co} c_w (T_{w,i}^0 - T_{w,co}^0) \quad (29)$$

$$M_{a,de}c_a \frac{T_{a,o} - T_{a,o}^0}{\Delta t} = \varepsilon_{de}C_{de,min}(T_{s,i}^0 - T_{a,cc}^0) + \dot{m}_a c_a (T_{a,cc}^0 - T_{a,o}^0) \quad (30)$$

$$M_{a,de} \frac{\omega_{a,o} - \omega_{a,o}^0}{\Delta t} = \varepsilon_{de,m} \dot{m}_a (\omega_{equ}^0 - \omega_{a,cc}^0) + \dot{m}_a (\omega_{a,cc}^0 - \omega_{a,o}^0) \quad (31)$$

where the superscript 0 denotes the value at the previous time.

Furthermore, a good initial condition is especially helpful in enhancing the optimization convergence speed and determining the function global minima. Hence, all unknown model parameters are pre-identified by the LMM [34] through static experimental data. The entire parameters identification procedure is shown in Fig. 8.

5. Validation and discussion

In order to validate the proposed model, experimental tests were conducted on a LDCDS and the system working with lithium chloride (LiCl) as desiccant. The experimental rig is shown in Fig. 9. The equipment shell is composed of polypropylene and is covered with a layer of rubber insulation cotton with a height of 1600 mm. The cooling coil is installed at the air inlet of equipment, in front of the dehumidifier tower with face dimensions corresponding to 500 mm

Figure 8: The flow chart of the identification procedure.

$\times 250$ mm, six rows and the aluminium fin pitch is 2 mm. The dehumidifier tower is filled with structured packing materials with a section area of 0.20 m^2 ($500 \text{ mm} \times 400 \text{ mm}$) and a height of 500 mm. A distributor and demister are

equipped at the top of the tower to offer a reasonable solution distribution and eliminate the carryover of the desiccant, respectively. Chilled water is provided by an independent chiller system. The flow rates of the air, desiccant solution and chilled water are regulated by the VSD fan, VSD pump and electrical ball valve, respectively.

Figure 9: Pilot plant of LDCDS: 1. cooling coil; 2. pumps; 3. cooler; 4. fan; 5. dehumidifier; 6. flow meters; 7. temperature/humidity transmitter; 8. temperature sensors; 9. electrical ball valve.

The air temperature and relative humidity at the inlet and outlet of the cooling coil and dehumidifier are measured by the probe type temperature/humidity transmitters. The chilled water temperature at the inlet and outlet of the cooling coil and cooler and desiccant solution temperature at the inlet and outlet of the cooler are obtained by PT-1000 temperature sensors. The volume flow rates of the process air, chilled water and desiccant solution are evaluated by a blade type, turbine and magnetic flow meter, respectively. The solution density is measured by a glass hydrometer, and it be used to determine the concentration

of solution with the corresponding temperature. A Data Acquisition (DAQ) system is adopted to collect sensor signals and samples data every 1 s. The specifications of sensors are listed in Table 1.

Table 1: Specifications of different measuring sensors.

Sensors	Type	Accuracy	Range
Fluid temperature sensor	PT-1000, 3-Wire	0.15 °C	0–100 °C
Chilled water flow meter	Turbine flow meter	±0.5%	0–50 L/min
Solution flow meter	Magnetic flow meter	±0.5%	0–30 L/min
Solution density meter	Glass hydrometer	1 kg/m ³	1.1–1.3 kg/L
Air temperature/humidity transmitter	Probe, EE210	0.1°C, ±0.5%	0–60 °C, 0–100%
Air flow meter	Blade, C310-BO	±5%	0–600 m ³ /h

To evaluate the effectiveness of the proposed model, the following four error indexes are applied: Mean Absolute Error (MAE), Relative Error (RE), Mean Relative Error (MRE) and Root Mean Square of Relative Error (RMSRE). The expressions are as follows:

$$MAE = \frac{|D_{real} - D_{calc}|}{N} \quad (32)$$

$$RE = \frac{|D_{real} - D_{calc}|}{D_{real}} \times 100\% \quad (33)$$

$$MRE = \frac{\sum_{i=1}^N RE_i}{N} \quad (34)$$

$$RMSRE = \sqrt{\frac{\sum_{i=1}^N (RE_i)^2}{N}} \quad (35)$$

where N , D_{real} and D_{calc} are the number of fitted points, experimental data and data calculated from the model, respectively.

According to the parameters identification procedure, the model parameters can be identified from both static and dynamic experimental data while the fluid mass constants are estimated from experimental step responses. Therefore, one of mass flow rates of the three working fluids is sequentially changed to produce a step response, the other two mass flow rates are kept constant, each operating condition lasts for approximately 20 min. Experimental data covering different operating condition is collected, and thus the model parameters are identified

through LMM and UKF methods. The identification results are listed in Table 2, the mass constants ($M_{a,cc}$, $M_{w,cc}$, $M_{a,de}$, $M_{s,co}$ and $M_{w,co}$) correspond to 7.9941, 12.4271, 10.7980, 15.6571 and 13.5102, respectively.

Table 2: Summary of parameters identification results.

Model	Identified model parameters
Cooling coil	$a_1 = 4.6879$, $a_2 = 15.6315$, $a_3 = 0.4720$
Dehumidifier	$c_1 = 1.9930$, $c_2 = 1.7540$, $c_3 = 1.1324$, $c_4 = 38.4538$ $c_5 = 6.8699$, $c_6 = 0.9937$, $c_7 = -1.3008$
Cooler	$d_1 = 12.8299$, $d_2 = 4.8880$, $d_3 = 0.8505$

Following the determination of the model parameters, the proposed model can be validated through experiments. The model validation includes two parts: the static performance validation and dynamic performance validation.

5.1. Static performance

The main focus of this study is on the LDCDS modeling, and thus it is necessary to obtain a model that is precise over a wide range of operating conditions. In addition to the ambient air temperature and humidity, all other independent variables that consist of the volume flow rates of the air, chilled water and desiccant solution, chilled water inlet temperature and desiccant solution concentration are varied during the experiments. The experiments cover most of designed capacity (10-90%) of the LDCDS. The heat transfer rates in the cooler, cooling coil and dehumidifier are varied between 0.3-2.3 kW, 0.3-2 kW, and 0.2-1.5 kW, respectively. And the mass transfer rate in the dehumidifier ranges 0.3-1.6 g/s. For each sub-model, 200 data sets under different steady state are collected, including the inlet and outlet temperatures and volume flow rates of the process air, desiccant solution and chilled water, the air inlet and outlet humidity and desiccant solution concentration. Based on the determined model parameters, the model static performance validation results are shown in Figs. 10-13 and the error indexes for the static performance prediction of LDCDS are summarized in Table 3.

(a) Heat transfer rate

(b) Relative error

Figure 10: Model validation of the heat transfer rate and relative error for the cooler.

(a) Heat transfer rate

(b) Relative error

Figure 11: Model validation of the heat transfer rate and relative error for cooling coil.

(a) Heat transfer rate

(b) Relative error

Figure 12: Model validation of the heat transfer rate and relative error for dehumidifier.

Figure 13: Model validation of the mass transfer rate and relative error for dehumidifier.

Table 3: Summary of the error indexes for static performance predictions.

Error index	Q_{co}	Q_{cc}	Q_{de}	N_{de}
MRE	2.20%	1.61%	2.15%	2.06%
RMSRE	2.61%	2.01%	2.46%	2.44%

As shown in Figs. 10-13, the model predictions are in accordance with the measured data in a wide working ranges. Specifically, 100% of 200 data points fall within a RE of 5% with the largest RE of 4.849%, 4.903%, 4.758% and 4.636%. And from Table 3, all error indexes are minimal. The static model is validated, and it is sufficiently precise for further control and optimization applications.

5.2. Dynamic performance

The dynamic performance validation of the LDCDS is carried out under the real-time operating conditions. The time varying inlet temperature and mass flow rates of the desiccant solution and chilled water to the cooler are shown in Fig. 14. Similarly, for cooling coil and dehumidifier, the time varying experimental operating conditions are shown in Fig. 15. In cooler, the model prediction results in terms of a comparison of the desiccant solution and chilled water outlet temperatures with those of measured data are shown in Fig. 16.

In the cooling coil, the results of comparing the outlet air and chilled water temperature in model predictions and practical system outputs are shown in Fig. 17. The results of comparing outlet air temperature and humidity of the dehumidifier in the model predictions and experimental data are depicted in Fig. 18. The error indexes for predicting the dynamic performance of LDCDS are listed in Table 4.

Figure 14: Time varying the fluid inlet temperature and mass flow rates to the cooler.

Figure 15: Time varying the fluid inlet properties and mass flow rates to the cooling coil and dehumidifier.

Figure 16: Comparison of the cooler outputs of model predictions with experimental data.

Figure 17: Comparison of the cooling coil outputs of model predictions with experimental data.

Figure 18: Comparison of the dehumidifier outputs of model predictions with experimental data.

Table 4: Summary of the error indexes for dynamic performance predictions.

Error index	$T_{s,i}$	$T_{w,co}$	$T_{a,cc}$	$T_{w,cc}$	$T_{a,o}$	$\omega_{a,o}$
MRE	0.67%	0.87%	0.92%	1.06%	2.02%	2.56%
RMSRE	0.83%	1.00%	1.17%	1.42%	2.50%	2.98%
MAE	0.073 °C	0.098 °C	0.177 °C	0.112 °C	0.277 °C	0.161 g/kg

As shown in Figs. 16-18 and Table 4, the proposed model accurately reflects the details of transient performance of LDCDS. The model accuracy is sufficient for the real-time applications, although the error will increase at the transients. The prediction error of cooler is the smallest among those errors of cooling coil and dehumidifier. The reason may be due to the measurement error of the solution fluid is much less than that of air fluid. For example, as in Table 1, the measuring precision of the air flow rate is less than that of the liquid flow rate. Moreover, the difference between the measurements of air temperature at the adjacent time may reaches 0.3 °C, while that of the liquid fluids is less than 0.06 °C. Another reason could be the changing flow rate of the chilled water in

the cooling coil. The chilled water in the cooling coil and dehumidifier is driven by the same water pump. Therefore, the flow rate of the chilled water in dehumidifier would change correspondingly to maintain constant inlet temperature of desiccant solution. As a result of that the flow rate of chilled water in cooling coil also change. After the cooling coil, the ambient air temperature gradually drops by approximately 8 °C and the humidity ratio reduces by 25% such that the inlet temperature difference for the dehumidifier between air and desiccant solution decreases. As a result, in the dehumidifier, both the latent and sensible cooling loads that should be handled are reduced. Given the same operating conditions, when compared with the traditional LDDS, the usage period of the desiccant solution in LDCDS is extended. Accordingly, the regeneration frequency of the desiccant solution and the thermal energy consumed by the desiccant regeneration system decreases.

6. Conclusions

In this study, an accurate and simplified dynamic model of LDCDS is proposed. The entire system is composed of three subsystems, including a cooling coil, a dehumidifier and a cooler. The thermal and moisture dynamic equations of each subsystem are established based on the energy and mass balance principles, and the heat and mass transfer rates are derived through the ε -NTU and hybrid modeling approach. All unknown model parameters are pre-identified through LMM and then refined through UKF algorithm from experimental data. The validation results indicate that the model predictions in terms of heat and mass transfer rates and output variables of each subsystem in accordance with the experimental data. The present model does not require complex iterative computation following the determination of model parameters. Additionally, the dynamic model is easily formulated into state-space form and the output is precisely calculated through the explicit finite difference method. These advantages are conducive to future control and optimization applications.

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APPENDIX A

The moisture air saturated vapor pressure can be calculated by using Hyland-Wexler equation under air temperature ranging from 0 °C to 200 °C as[40].

$$\ln P_{a,sat} = C_1/T + C_2 + C_3T + C_4T^2 + C_5T^3 + C_6T \ln T \quad (36)$$

where $P_{a,sat}$ and T are the air saturated vapor pressure and absolute temperature, respectively, $C_1 \sim C_6$ are constants $C_1 = -5.8002206E + 03$, $C_2 = 1.3914993$, $C_3 = -4.8640239E - 02$, $C_4 = 4.1764768E - 05$, $C_5 = -1.4452093E - 08$ and $C_6 = 6.5459673$.

APPENDIX B

The state matrix A_c , input matrix B_c , output matrix C_c and disturbance matrix E_c are expressed as following:

$$A_c = \begin{pmatrix} A11 & 0 & 0 & 0 & 0 & 0 \\ 0 & A22 & 0 & 0 & 0 & 0 \\ A31 & 0 & A33 & A34 & 0 & 0 \\ 0 & 0 & 0 & A44 & 0 & 0 \\ 0 & 0 & 0 & 0 & A55 & 0 \\ A61 & 0 & 0 & 0 & 0 & A66 \end{pmatrix} \quad (37)$$

$$B_c = \begin{pmatrix} B11 & 0 & B13 & 0 \\ B21 & 0 & B23 & 0 \\ B31 & B32 & 0 & 0 \\ 0 & B42 & 0 & B44 \\ 0 & B52 & 0 & B54 \\ B61 & B62 & 0 & 0 \end{pmatrix} \quad (38)$$

$$C_c = \begin{pmatrix} 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{pmatrix} \quad (39)$$

$$E_c = \begin{pmatrix} E11 & E12 & 0 & E14 & 0 \\ B21 & E22 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & E42 & E43 & 0 & 0 \\ 0 & E52 & E53 & 0 & 0 \\ 0 & 0 & 0 & 0 & E65 \end{pmatrix} \quad (40)$$

where $A11 = -\dot{m}_a/M_{a,cc}$,

$$A22 = -\dot{m}_w/M_{w,cc},$$

$$A31 = \frac{\dot{m}_a}{M_{a,de}} + \frac{C_{de,\min}}{M_{a,de}} \left(e^{\frac{-c_1 \dot{m}_a^{c_3}}{\left(c_2 \left(\frac{\dot{m}_a}{\dot{m}_s}\right)^{c_3} + 1\right)^{C_{de,\min}}}} - 1 \right),$$

$$A33 = -\dot{m}_a/M_{a,de},$$

$$A34 = -\frac{C_{de,\min}}{M_{a,de}} \left(e^{\frac{-c_1 \dot{m}_a^{c_3}}{\left(c_2 \left(\frac{\dot{m}_a}{\dot{m}_s}\right)^{c_3} + 1\right)^{C_{de,\min}}}} - 1 \right),$$

$$A44 = -\dot{m}_s/M_{s,co},$$

$$A55 = -\dot{m}_{w,co}/M_{w,co},$$

$$A61 = \frac{c_4 \dot{m}_a^{c_6} (w_{a,cc} - w_{equ})}{M_{a,de} (T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7})^2} e^{\frac{-c_4 \dot{m}_a^{c_6}}{\left(T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7}\right)^2}},$$

$$A66 = -\dot{m}_a/M_{a,de},$$

$$B11 = -\frac{T_{a,cc} - T_{a,i}}{M_{a,cc}} - \frac{a_1 a_3 \dot{m}_a^{a_3} (T_{a,i} - T_{w,i})}{M_{a,cc} c_a \left(a_2 \left(\frac{\dot{m}_a}{\dot{m}_w}\right)^{a_3} + 1\right)^2} e^{\frac{-a_1 \dot{m}_a^{a_3}}{\left(a_2 \left(\frac{\dot{m}_a}{\dot{m}_w}\right)^{a_3} + 1\right)^{C_{cc,\min}}}},$$

$$B13 = -\frac{a_1 a_2 a_3 \dot{m}_a^{2a_3} (T_{a,i} - T_{w,i})}{M_{a,cc} c_a \dot{m}_w^{a_3+1} \left(a_2 \left(\frac{\dot{m}_a}{\dot{m}_w}\right)^{a_3} + 1\right)^2} e^{\frac{-a_1 \dot{m}_a^{a_3}}{\left(a_2 \left(\frac{\dot{m}_a}{\dot{m}_w}\right)^{a_3} + 1\right)^{C_{cc,\min}}}},$$

$$B21 = \frac{a_1 a_3 \dot{m}_a^{a_3-1} (T_{a,i} - T_{w,i})}{M_{a,cc} c_a \left(a_2 \left(\frac{\dot{m}_a}{\dot{m}_w}\right)^{a_3} + 1\right)^2} e^{\frac{-a_1 \dot{m}_a^{a_3}}{\left(a_2 \left(\frac{\dot{m}_a}{\dot{m}_w}\right)^{a_3} + 1\right)^{C_{cc,\min}}}},$$

$$B23 = -\frac{T_{w,cc} - T_{w,i}}{M_{w,cc}} + \frac{a_1 a_2 a_3 \dot{m}_a^{2a_3} (T_{a,i} - T_{w,i})}{M_{w,cc} c_w \dot{m}_w^{a_3+1} \left(a_2 \left(\frac{\dot{m}_a}{\dot{m}_w}\right)^{a_3} + 1\right)^2} e^{\frac{-a_1 \dot{m}_a^{a_3}}{\left(a_2 \left(\frac{\dot{m}_a}{\dot{m}_w}\right)^{a_3} + 1\right)^{C_{cc,\min}}}},$$

$$B31 = \frac{T_{a,cc} - T_{a,o}}{M_{a,de}} - \frac{c_1 c_3 \dot{m}_a^{c_3-1} (T_{a,cc} - T_{s,i})}{M_{a,de} c_a \left(c_2 \left(\frac{\dot{m}_a}{\dot{m}_s}\right)^{c_3} + 1\right)^2} e^{\frac{-c_1 \dot{m}_a^{c_3}}{\left(c_2 \left(\frac{\dot{m}_a}{\dot{m}_s}\right)^{c_3} + 1\right)^{C_{de,\min}}}},$$

$$B32 = -\frac{c_1 c_2 c_3 \dot{m}_a^{2c_3} (T_{a,cc} - T_{s,i})}{M_{a,de} c_a \dot{m}_s^{c_3+1} \left(c_2 \left(\frac{\dot{m}_a}{\dot{m}_s}\right)^{c_3} + 1\right)^2} e^{\frac{-c_1 \dot{m}_a^{c_3}}{\left(c_2 \left(\frac{\dot{m}_a}{\dot{m}_s}\right)^{c_3} + 1\right)^{C_{de,\min}}}},$$

$$B42 = \frac{T_{s,co} - T_{s,i}}{M_{s,co}} - \frac{d_1 d_3 \dot{m}_s^{d_3-1} (T_{s,co} - T_{w,i})}{M_{s,co} c_s \left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}}\right)^{d_3} + 1\right)^2} e^{\frac{-d_1 \dot{m}_s^{d_3}}{\left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}}\right)^{d_3} + 1\right)^{C_{co,\min}}}},$$

$$\begin{aligned}
B44 &= -\frac{d_1 d_2 d_3 \dot{m}_s^{2d_3} (T_{s,co} - T_{w,i})}{M_{s,co} c_s \dot{m}_{w,co}^{d_3+1} \left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right)^2} e^{\frac{-d_1 \dot{m}_s^{d_3}}{\left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right) C_{co,\min}}}, \\
B52 &= \frac{d_1 d_3 \dot{m}_s^{d_3-1} (T_{s,co} - T_{w,i})}{M_{w,co} c_w \left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right)^2} e^{\frac{-d_1 \dot{m}_s^{d_3}}{\left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right) C_{co,\min}}}, \\
B54 &= -\frac{T_{w,co} - T_{w,i}}{M_{w,co}} + \frac{d_1 d_2 d_3 \dot{m}_s^{2d_3} (T_{s,co} - T_{w,i})}{M_{w,co} c_w \dot{m}_{w,co}^{d_3+1} \left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right)^2} e^{\frac{-d_1 \dot{m}_s^{d_3}}{\left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right) C_{co,\min}}}, \\
B61 &= \frac{\omega_{equ} - \omega_{a,o}}{M_{a,de}} + \frac{\omega_{a,cc} - \omega_{equ}}{M_{a,de}} e^{\frac{-c_4 \dot{m}_a^{c_6}}{\dot{m}_a (T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7})}} \\
&\quad + \frac{c_4 \dot{m}_a^{c_6-1} (\omega_{a,cc} - \omega_{equ}) (T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7} - c_6 T_{a,cc})}{M_{a,de} (T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7})^2} e^{\frac{-c_4 \dot{m}_a^{c_6}}{\dot{m}_a (T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7})}}, \\
B62 &= \frac{c_4 c_5 c_7 \dot{m}_a^{2c_6} \dot{m}_s^{c_7-1} (\omega_{a,cc} - \omega_{equ})}{M_{a,de} (T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7})^2} e^{\frac{-c_4 \dot{m}_a^{c_6}}{\dot{m}_a (T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7})}}, \\
E11 &= \frac{\dot{m}_a}{M_{a,cc}} + \frac{C_{cc,\min}}{M_{a,cc} c_a} \left(e^{\frac{-a_1 \dot{m}_a^{a_3}}{a_2 \left(\frac{\dot{m}_a}{\dot{m}_w} \right)^{a_3} + 1} C_{cc,\min}} - 1 \right), \\
E12 &= -\frac{C_{cc,\min}}{M_{a,cc} c_a} \left(e^{\frac{-a_1 \dot{m}_a^{a_3}}{a_2 \left(\frac{\dot{m}_a}{\dot{m}_w} \right)^{a_3} + 1} C_{cc,\min}} - 1 \right), \\
E21 &= -\frac{C_{cc,\min}}{M_{w,cc} c_w} \left(e^{\frac{-a_1 \dot{m}_a^{a_3}}{a_2 \left(\frac{\dot{m}_a}{\dot{m}_w} \right)^{a_3} + 1} C_{cc,\min}} - 1 \right), \\
E22 &= \frac{\dot{m}_w}{M_{w,cc}} + \frac{C_{cc,\min}}{M_{a,cc} c_a} \left(e^{\frac{-a_1 \dot{m}_a^{a_3}}{a_2 \left(\frac{\dot{m}_a}{\dot{m}_w} \right)^{a_3} + 1} C_{cc,\min}} - 1 \right), \\
E42 &= -\frac{C_{co,\min}}{M_{s,co} c_s} \left(e^{\frac{-d_1 \dot{m}_s^{d_3}}{\left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right) C_{co,\min}}} - 1 \right), \\
E43 &= \frac{\dot{m}_s}{M_{s,co}} + \frac{C_{co,\min}}{M_{s,co} c_s} \left(e^{\frac{-d_1 \dot{m}_s^{d_3}}{\left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right) C_{co,\min}}} - 1 \right), \\
E52 &= \frac{\dot{m}_{w,co}}{M_{w,co}} + \frac{C_{co,\min}}{M_{w,co} c_w} \left(e^{\frac{-d_1 \dot{m}_s^{d_3}}{\left(d_2 \left(\frac{\dot{m}_s}{\dot{m}_{w,co}} \right)^{d_3} + 1 \right) C_{co,\min}}} - 1 \right), \\
E65 &= -\frac{\dot{m}_a}{M_{a,de}} \left(e^{\frac{-c_4 \dot{m}_a^{c_6}}{\dot{m}_a (T_{a,cc} + c_5 \dot{m}_a^{c_6} \dot{m}_s^{c_7})}} - 1 \right).
\end{aligned}$$

Nomenclature

$a_1 - a_3$	lumped parameters of the cooling coil (dimensionless)
$A_{cc/de/co}$	heat or mass transfer area of cooling coil, dehumidifier or cooler (m ²)
$b_1 - b_8$	constant parameters (dimensionless)
$c_1 - c_7$	lumped parameters of the dehumidifier (dimensionless)

$c_{a/w/s}$	special heat capacity of air, chilled water or solution (J/(kg °C))
C	constant (dimensionless)
$C_{cc/de/co,min}$	fluid specific heat rate in cooling coil, dehumidifier or cooler (W/(°C))
$d_1 - d_3$	lumped parameters of the cooler (dimensionless)
D	geometric parameter of cooling coil (m)
D_{calc}	calculated data (dimensionless)
D_{real}	experimental data (dimensionless)
e	constant (dimensionless)
f	constant (dimensionless)
h	heat transfer coefficient (W/(m ² °C))
h_g	gas-phase mass transfer coefficient ($kg/(m^2 s \Delta \omega)$)
h_l	liquid phase mass transfer coefficient ($kg/(m^2 s \Delta \omega)$)
h_a	heat transfer coefficient of air (W/(m ² °C))
h_w	heat transfer coefficient of chilled water (W/(m ² °C))
H	Henry's law constant (dimensionless)
H_{cc}	overall heat transfer coefficient in cooling coil (W/(m ² °C))
H_{co}	overall heat transfer coefficient in cooler (W/(m ² °C))
H_{de}	overall heat transfer coefficient in dehumidifier (W/(m ² °C))
H_{fg}	latent heat of vaporization (J/kg)
H_G	overall gas-phase mass transfer coefficient ($kg/(m^2 s \Delta \omega)$)
k	thermal conductivity (W/(m ² °C))
L_w	water heat of evaporation (J/kg)
$\dot{m}$	fluid mass flow rate (kg/s)
$\dot{m}_a$	air mass flow rate (kg/s)
$\dot{m}_s$	solution mass flow rate (kg/s)
$\dot{m}_w$	chilled water mass flow rate in cooling coil (kg/s)
$\dot{m}_{w,co}$	chilled water mass flow rate in cooler (kg/s)
M	fluid mass constant (kg)
N_{max}	maximum mass transfer rate in dehumidifier (kg/s)
N_{de}	mass transfer rate in dehumidifier (kg/s)

NTU	number of transfer units (dimensionless)
$P_{a,sat}$	saturated vapor pressure of air (Pa)
Q_{cc}	heat transfer rate in cooling coil (W)
Q_{co}	heat transfer rate in cooler (W)
Q_{de}	heat transfer rate in dehumidifier (W)
Q_{max}	maximum heat transfer rate (W)
R	heat resistance of the metal wall conduction ($^{\circ}\text{C}/\text{W}$)
$T_{a,i}$	ambient air temperature ($^{\circ}\text{C}$)
$T_{a,cc}$	outlet air temperature of cooling coil ($^{\circ}\text{C}$)
$T_{a,o}$	outlet air temperature of dehumidifier ($^{\circ}\text{C}$)
$T_{s,co}$	inlet solution temperature of cooler ($^{\circ}\text{C}$)
$T_{s,i}$	inlet solution temperature of dehumidifier ($^{\circ}\text{C}$)
$T_{s,o}$	outlet solution temperature of dehumidifier ($^{\circ}\text{C}$)
$T_{w,i}$	inlet chilled water temperature ($^{\circ}\text{C}$)
$T_{w,cc}$	outlet chilled water temperature of cooling coil ($^{\circ}\text{C}$)
$T_{w,co}$	outlet chilled water temperature of cooler ($^{\circ}\text{C}$)
ε_{cc}	heat transfer effectiveness of cooling coil (dimensionless)
ε_{co}	heat transfer effectiveness of cooler (dimensionless)
ε_{de}	heat transfer effectiveness of dehumidifier (dimensionless)
$\varepsilon_{de,m}$	mass transfer effectiveness of dehumidifier (dimensionless)
ω_{cc}	outlet air humidity of cooling coil (g/kg (dry air))
ω_{equ}	equilibrium humidity ratio (g/kg (dry air))
ω_i	inlet air humidity of cooling coil (g/kg (dry air))
ω_o	outlet air humidity of dehumidifier (g/kg (dry air))
φ_a	relative humidity of air (%)
ν	velocity of fluid (m/s)
ρ	density of fluid (kg/m^3)
μ	viscosity of fluid (Pa s)

Subscripts

a	air
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<i>cc</i>	cooling coil
<i>co</i>	cooler
<i>de</i>	dehumidifier
<i>i</i>	inlet
<i>o</i>	outlet
<i>s</i>	desiccant solution
<i>sat</i>	saturated
<i>w</i>	chilled water

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