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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.

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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 [5–8]. 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

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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 $\triangle\omega$))
h_l	liquid phase mass transfer coefficient (kg/(m^2 s $\triangle\omega$))
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 $\triangle\omega$))
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 (°C/W)
$T_{a,i}$	ambient air temperature (°C)
$T_{a,cc}$	outlet air temperature of cooling coil (°C)
$T_{a,o}$	outlet air temperature of dehumidifier (°C)
$T_{s,co}$	inlet solution temperature of cooler (°C)
$T_{s,i}$	inlet solution temperature of dehumidifier (°C)
$T_{s,o}$	outlet solution temperature of dehumidifier (°C)
$T_{w,i}$	inlet chilled water temperature (°C)
$T_{w,cc}$	outlet chilled water temperature of cooling coil (°C)
$T_{w,co}$	outlet chilled water temperature of cooler (°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
cc	cooling coil
co	cooler
de	dehumidifier
i	inlet
o	outlet
s	desiccant solution
sat	saturated
w	chilled water

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 effectiveness model of liquid desiccant heat/mass exchanger. Ren [26] developed analytical expressions to improve the ε-NTU model by applying perturbation technique. Langroud 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

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