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Desiccant wheel thermal performance modeling for indoor humidity optimal control ☆

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HIGHLIGHTS

- An optimal humidity control model is formulated to control the indoor humidity.
- MPC strategy is used to implement the optimal operation solution.
- Practical applications of the MPC strategy is illustrated by the case study.

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ABSTRACT

Thermal comfort is an important concern in the energy efficiency improvement of commercial buildings. Thermal comfort research focuses mostly on temperature control, but humidity control is an important aspect to maintain indoor comfort too. In this paper, an optimal humidity control model (OHCM) is presented. Model predictive control (MPC) strategy is applied to implement the optimal operation of the desiccant wheel during working hours of a commercial building. The OHCM is revised to apply the MPC strategy. A case is studied to illustrate the practical applications of the MPC strategy.

1. Introduction

Evidences indicate that indoor humidity environment is closely related to health problems. Low humidity environment is associated with dryness of skin and throats, mucous membrane, sensory irritation of eyes [1–4]. High humidity environment may induce the growth of mould leading to respiratory discomfort and allergies [5,6]. Therefore, keeping indoor humidity environment steady at the correct level is very important to ensure people's health in commercial buildings. For the dehumidification of commercial buildings in summer, existing research focuses on developing new energy efficient equipment, applying different control strategy in traditional air conditioning, or changing design parameters of dehumidifier. Chua et al. [7] presents a model to study the humidity control effects of different control strategies in traditional air conditioning system during the part-load condition. The results show that some strategies are more effective than others in sustaining acceptable indoor humidity under part-load condition, but none of these strategies provides the best performance under full load conditions. Mazzei et al. [8] studies the principles of mechanical and chemical dehumidification. The chemical dehumidification is widely used in recent years. According to the different desiccant materials, the chemical dehumidification can be classified in two classes. The first class is the liquid desiccant system. Zhang and Yoshino [9] presents the working principle of a liquid desiccant system combined refrigeration/heat grid system with the independent humidity control strategy, and the experiments show that the combined system can remove the entire latent load and part of the sensible load of the building. The second class is the solid desiccant system, such as desiccant wheel. Subramanyam et al. [10] presents a model of a desiccant wheel used for dehumidifying ventilation air. The simulation result in [10] shows that the model has good accuracy in certain working conditions. A review of the mathematical models for desiccant wheel is presented in [11]. Antonellis et al. [12] illustrates an optimal design problem for the desiccant wheel. The optimal wheel speed and area ratio of dehumidification section to regeneration section are determined under a range of regeneration temperatures from 60 °C to 150 °C. Panarasa et al. [13] presents an experimental validation of a simplified approach for a desiccant wheel model. This approach can be used to design the desiccant wheel and analyze its performance easily. Model predictive control (MPC) strategy has the ability to handle constraints, being able to use simple models and to change controls dynamically in terms

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Nomenclature

A_d	area of desiccant wheel (m^2)
A_{de}	area of dehumidification section (m^2)
A_p	area of processing section (m^2)
A_r	area of regeneration section (m^2)
C_a	specific heat of air (J kg^{-1} K^{-1})
C_d	specific heat of desiccant material (J kg^{-1} K^{-1})
C_{vl}	specific heat of water liquid (J kg^{-1} K^{-1})
C_v	specific heat of water vapour (J kg^{-1} K^{-1})
f_d	mass fraction of desiccant in the wheel
G	moisture generation rate (kg s^{-1})
h	convective heat transfer coefficient (W m^{-2} K^{-1})
h_m	mass transfer coefficient (kg m^{-2} s^{-1})
H_a	humidity ratio of air (kg kg^{-1})
H_{ap}	humidity ratio of inlet air in the dehumidification section (kg kg^{-1})
H_{ar}	humidity ratio of inlet air in the regeneration section (kg kg^{-1})
H_{dp}	humidity ratio of desiccant material in the dehumidification section (kg kg^{-1})
H_{dr}	humidity ratio of desiccant material in the regeneration section (kg kg^{-1})
H_i	indoor humidity ratio (kg kg^{-1})
H_{in}	initial indoor humidity ratio (kg kg^{-1})
H_o	outdoor humidity ratio (kg kg^{-1})
H_s	indoor humidity ratio set point (kg kg^{-1})
H_{sa}	humidity ratio of supply air (kg kg^{-1})
L	humidity ratio of supply air (m)
M_{sor}	heat of adsorption (J kg^{-1})
P	perimeter of flow channel (m)
Pr	pressure (Pa)
Q_i	air flow rate of infiltration (m^3 s^{-1})
Q_{lb}	lower bound of flow rate of supply air (m^3 s^{-1})
Q_{sa}	flow rate of supply air (m^3 s^{-1})
Q_{ub}	upper bound of flow rate of supply air (m^3 s^{-1})
t	time (s)
t_p	time required for the dehumidification per one wheel revolution (s)
t_r	time required for the regeneration per one wheel revolution (s)
T_a	temperature of air (K)
T_{ap}	temperature of inlet air in the dehumidification section (K)
T_{ar}	temperature of inlet air in the regeneration section (K)
T_d	temperature of desiccant wheel (K)
T_{dp}	temperature of desiccant material in the dehumidification section (K)
T_{dr}	temperature of desiccant material in the regeneration section (K)
T_{lb}	lower bound of temperature of inlet air in the regeneration section (K)
T_s	indoor temperature set point (K)
T_{sa}	temperature of supply air (K)
T_{ub}	upper bound of temperature of inlet air in the regeneration section (K)
u_a	velocity of air (m s^{-1})
u_{ap}	velocity of inlet air in the dehumidification section (m s^{-1})
u_{ar}	velocity of inlet air in the regeneration section (m s^{-1})
$u_{ar,lb}$	lower bound of velocity of inlet air in the regeneration section (m s^{-1})
$u_{ar,ub}$	upper bound of velocity of inlet air in the regeneration section (m s^{-1})
u_d	wheel speed (rph)
$u_{d,lb}$	lower bound of wheel speed (rph)
$u_{d,ub}$	upper bound of wheel speed (rph)
V	volume of building (m^3)
W	water content of the desiccant wheel material (kg kg^{-1})
z	axial coordinate (m)
β	weight factor
ϕ_d	relative humidity of desiccant material
ρ_a	density of air (kg m^{-3})
ρ_d	density of desiccant material (kg m^{-3})

of system changes, which makes it very practical to use in various energy problems. Zhang and Xia [14] proposes an MPC approach to the periodic implementation of the optimal solutions of a class of resource allocation problems. Privara et al. [15] presents the MPC strategy applied to the temperature control of a building. None of the existing research applies MPC strategies in the desiccant wheel of commercial buildings to maintain a required indoor humidity. In this paper, an optimal humidity control model (OHCM) for a commercial building with a desiccant wheel is presented. MPC strategy is applied to implement the optimal operation of the desiccant wheel during working hours of the commercial building. The OHCM is revised to apply the MPC strategy. To illustrate the practical applications of the MPC strategy, the optimization of the desiccant wheel in a commercial building of South Africa is studied. The layout of the paper is as follows. In Section 2, the backgrounds of the desiccant wheel model and indoor humidity model are recalled. The formulation of OHCM and the procedure of the MPC strategy can be found in Section 3. Section 4 illustrates the MPC strategy by a case study. The last section is concluding remarks.

2. Background

2.1. Desiccant wheel model

The desiccant wheel system to be considered in this paper is shown in Fig. 1. The desiccant wheel is driven by an engine and moves at a given rotary velocity. It consists of a large number of channels whose walls are constituted by supporting material and coated with desiccant material. The desiccant wheel system is divided into two sections: dehumidification section (position 1–2) and regeneration section (position 3–5). Water vapour of outdoor air is absorbed by the desiccant wheel when the outdoor air is moving from position 1–2. Note that the outdoor air temperature is changed when it moves from position 1–2. While the indoor hot air is heated up by the heater (position 3–4) and is flowing through the desiccant wheel, water is desorbed out from the desiccant and the desiccant material is regenerated (position 4–5). The assumptions of desiccant wheel are as follows:

(1) The air flow is one-dimensional.
(2) The axial heat conduction and mass diffusion in the fluid are neglected.

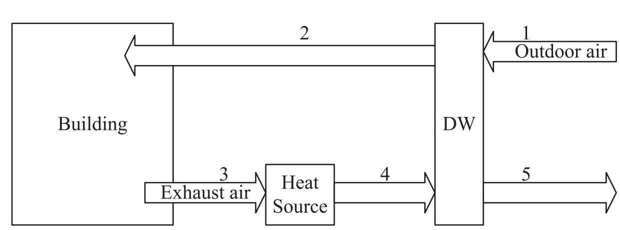

Fig. 1. Typical desiccant wheel system set-up.

(3) There is no leakage of fluid in the desiccant wheel.
(4) All ducts are impermeable and adiabatic.
(5) The thermodynamic properties are constant and uniform.
(6) The heat and mass transfer coefficients between the air flow and the desiccant wall are constant along the channel.
(7) The velocity of air in the dehumidification sections is the same with it in the regeneration section.

Based on the above assumptions, the energy and mass conservation equations of desiccant wheel can be written as follows [16].

Mass conservation for the processing air:

$$\frac{\partial H_a}{\partial z} = \frac{h_m P}{u_a \rho_a A_p}(H_d - H_a). \tag{1}$$

Energy conservation for the processing air:

$$(C_a + C_v H_a)\frac{\partial T_a}{\partial z} = \frac{hP}{u_a \rho_a A_p}(T_d - T_a). \tag{2}$$

Conservation of water content for the absorbent:

$$\frac{\partial W}{\partial t} = \frac{h_m P}{f_d \rho_d A_d}(H_a - H_d). \tag{3}$$

Conservation of energy for the absorbent:

$$(C_d + C_{vl} W f_d)\frac{\partial T_a}{\partial t} = \frac{hP}{\rho_a A_p}(T_a - T_d) + \frac{h_m P M_{sor}}{\rho_a A_d}(H_a - H_d). \tag{4}$$

The above equations are subject to the following boundary and initial conditions.

For the dehumidification section

$$T_a(t,0) = T_{ap}, \tag{5}$$

$$H_a(t,0) = H_{ap}, \tag{6}$$

$$T_{dp}(0,z) = T_{dr}(t_r, L - z), \tag{7}$$

$$H_{dp}(0,z) = H_{dr}(t_r, L - z). \tag{8}$$

For the regeneration section

$$T_a(t,0) = T_{ar}, \tag{9}$$

$$H_a(t,0) = H_{ar}, \tag{10}$$

$$T_{dr}(0,z) = T_{dp}(t_p, L - z), \tag{11}$$

$$H_{dr}(0,z) = H_{dp}(t_p, L - z). \tag{12}$$

Eqs. (1)–(4) have five unknowns T_a, T_d, H_a, H_d and W. In order to solve this set of equations, the relation between W and H_d is determined as

$$H_d = f(W). \tag{13}$$

This relation is determined by the physical attribute of desiccant material. The silica gel is used as a desiccant material in this paper. Therefore, the relative humidity of the desiccant material can be calculated as [18]

$$\phi_d = 0.0078 - 0.0576W + 24.2W^2 - 124W^3 + 204W^4. \tag{14}$$

The relation between W and H_d is calculated as [14]

$$f(W) = \frac{0.622\phi_d Pr_{vs}}{Pr - \phi_d Pr}. \tag{15}$$

The saturated pressure Pr_{vs} can be calculated as [11]

$$Pr_{vs} = \exp\left(23.196 - \frac{3816.44}{T_d - 46.13}\right). \tag{16}$$

The properties of desiccant wheel are given in [16].

2.2. Indoor humidity model

Humidity in the building comes from three sources. The first source is the water vapour of air generated by infiltration from outdoor to indoor. Infiltration is the flow of outdoor air into a conditioned space through doors, windows, cracks, etc. Infiltration is also known as air leakage into a building. The second source is the water vapour of air generated by the human skin. The third source is the water vapour added by the ventilation of desiccant wheel. Therefore, the variation of humidity in the commercial building is given by [17]

$$\frac{dH_i}{dt} = \frac{Q_{sa}}{V}(H_{sa} - H_i) + \frac{Q_i}{V}(H_o - H_i) + \frac{G}{\rho_a V}. \tag{17}$$

The variation of humidity in the commercial building without the ventilation of desiccant wheel is calculated as [17]

$$\frac{dH_i}{dt} = \frac{Q_i}{V}(H_o - H_i) + \frac{G}{\rho_a V}. \tag{18}$$

Eqs. (17) and (18) are subject to the initial condition $H_i(0) = H_{in}(0)$ and solved by Matlab function Ode23 with 360s step size.

3. Optimal humidity control model and MPC strategy

3.1. Optimal humidity control model

The commercial building to be studied in this paper has several storeys and each storey has the same height, single glazing glass windows and concrete walls. The building has no other devices for indoor moisture generation except the desiccant wheel. None of chimneys or any other natural ventilation devices is applied in the building. It can be assumed that all the rooms of the building have the same humidity ratio. The water absorbed and desorbed by walls and furniture are ignored.

To keep the indoor humidity ratio steady at the required level, the desiccant wheel will remove the water vapour of air generated by infiltration and occupants. The temperature of supply air of desiccant wheel is required to be close to the temperature set point of the building so as to reduce the indoor sensible cooling load. Therefore, the optimal humidity control model (OHCM) has two objectives. The first objective of the OHCM is to minimize the difference between the actual indoor humidity ratio and the humidity ratio set point, and the second objective is to minimize the difference between the actual temperature of the supply air and the indoor temperature set point. Consider a fixed time interval $[t_0, t_N]$, for example, 5 h, 1 day or 1 week. The objective function of the OHCM is defined as follow

$$\min \sum_{m=1}^{N} (1-\beta)(H_i(t_m) - H_s)^2 + \beta(T_{sa}(t_m) - T_s)^2, \tag{19}$$

where $\beta \in [0,1]$ is the weight factor that indicate the importance of sub objective. The following constraints are used in the OHCM.

(1) Indoor humidity ratio constraints: According to the solution of ordinary differential Eq. (17), the indoor humidity ratio at time is given by

$$H_i(t_m) = F_1(H_{sa}(t_m), H_{in}(t_m), Q_{sa}(t_m), Q_i, H_o(t_m), G). \tag{20}$$

(2) Supply air constraints: The velocity of supply air at time t_m must be between the maximum and minimum specified velocity. The temperature and humidity ratio of the supply air must be satisfied Eqs. (1)–(4). Eqs. (1)–(4) are discretized into finite difference equations by forward difference method. The finite difference equations are given as follows

$$\frac{H_a(t,z+\Delta z)-H_a(t,z)}{\Delta z}=C_1(H_d(t,z)-H_a(t,z)), \tag{21}$$

$$(C_a+C_vH_a(t,z))\frac{T_a(t,z+\Delta z)-T_a(t,z)}{\Delta z} = C_2(T_d(t,z)-T_a(t,z)), \tag{22}$$

$$\frac{W(t+\Delta t,z)-W(t,z)}{\Delta t}=C_3(H_a(t,z)-H_d(t,z)), \tag{23}$$

$$(C_d+C_{vi}W(t,z)f_d)\frac{T_a(t+\Delta t,z)-T_a(t,z)}{\Delta t} = C_4(T_a(t,z)-T_d(t,z))+C_5(H_a(t,z)-H_d(t,z)), \tag{24}$$

where

$$C_1=\frac{h_mP}{u_a\rho_aA_p},\quad C_2=\frac{hP}{u_a\rho_aA_p},\ C_3=\frac{h_mP}{f_a\rho_dA_d},\ C_4=\frac{hP}{\rho_dA_p},$$
$$C_5=\frac{h_mPM_{sor}}{\rho_aA_d},\ \Delta z=\frac{L}{20},$$
$$\Delta t=\begin{cases} t_p/20, \textit{Dehumidification section}\\ t_r/20, \textit{Regeneration section}\end{cases},$$
$$t_p=\frac{3600A_{de}}{u_dA_d}\ \text{and}\ t_r=\frac{3600A_r}{u_dA_d}.$$

According to the solution of the finite difference equations, the temperature and humidity ratio of the supply air at time t_m are given by

$$H_{sa}(t_m)=H_a(t_p,L) = F_2(T_{ap}(t_m),T_{ar}(t_m),H_{ap}(t_m),u_{ar}(t_m),u_{ap}(t_m),u_d(t_m)), \tag{25}$$

$$T_{sa}(t_m)=T_a(t_p,L) = F_3(T_{ap}(t_m),T_{ar}(t_m),u_{ar}(t_m),u_{ap}(t_m),u_d(t_m)). \tag{26}$$

(3) Initial indoor humidity ratio constraints: For $m\in[1,N]$, the initial indoor humidity ratio at time t_m should be equal to the indoor humidity ratio at time t_{m-1}.
(4) Desiccant wheel constraints: The velocity and temperature of inlet air in the regeneration section at time t_m must be between the maximum and minimum specified velocity and temperature respectively. The wheel speed at time t_m must be between the maximum and minimum specified speed.

According to the description and assumption of desiccant wheel in Section 2, it can be assumed that $u_{ar}=u_{ap}$, $H_{ap}=H_a$. Therefore, the OHCM is defined as

$$\begin{aligned}\min_{u_{ar},u_d,T_{ar},Q_{sa}}\ &\sum_{m=1}^{N}(1-\beta)(H_i(t_m)-H_s)^2+\beta(T_{sa}(t_m)-T_s)^2\\
\text{s.t.}\ &H_i(t_m)=F_1(H_{sa}(t_m),H_{in}(t_m),Q_{sa}(t_m),Q_i,H_o(t_m),G),\\
&H_{sa}(t_m)=F_2(T_{ap}(t_m),T_{ar}(t_m),H_o(t_m),u_{ar}(t_m),u_{ap}(t_m),u_d(t_m)),\\
&T_{sa}(t_m)=F_3(T_{ap}(t_m),T_{ar}(t_m),u_{ar}(t_m),u_{ap}(t_m),u_d(t_m)),\\
&H_{in}(t_m)=H_i(t_{m-1}),\\
&u_{ar}(t_m)=u_{ap}(t_m),\\
&u_{ar,lb}\leqslant u_{ar}(t_m)\leqslant u_{ar,ub},\\
&u_{d,lb}\leqslant u_d(t_m)\leqslant u_{d,ub},\\
&T_{lb}\leqslant T_{ar}(t_m)\leqslant T_{ub},\\
&Q_{lb}\leqslant Q_{sa}(t_m)\leqslant Q_{ub},\\
&1\leqslant m\leqslant N,\beta\in[0,1].\end{aligned} \tag{27}$$

The above OHCM is a quadratic linear programming problem. It is solved by Matlab genetic algorithm (GA) toolbox in this paper. The population size of the GA method is set as 100, and the number of generations is chosen as 2000.

3.2. MPC strategy

In a commercial building, it is important to maintain the required humidity ratio during daily working hours, while the humidity at non-office hours is often ignored. Assume that the period of daily working hours is from 8:00 to 18:00. Then the control period is 10 h. If the sampling period of the OHCM is 1 h, the number of the variables in the OHCM is 40. The corresponding computing time is 4 min on a personal computer with CPU Intel Core2 2.5 GHz. If the sampling period of the OHCM is 0.5 h, then the number of the variables is 80, and the corresponding computing time is about 16 min on the same computer. Following the usual requirement for load profile and the computational complexity, the sampling period is chosen as 1 h in this paper and (19) can be rewritten as follow.

$$\min\sum_{m=1}^{10}(1-\beta)(H_i(t_m)-H_s)^2+\beta(T_{sa}(t_m)-T_s)^2. \tag{28}$$

The closed-loop objective function of MPC strategy is defined as follows.

$$\min\sum_{m=1+k}^{10+k}(1-\beta)(H_i(t_m)-H_s)^2+\beta(T_{sa}(t_m)-T_s)^2. \tag{29}$$

Assume that the MPC strategy is applied in 2 days. When $k=1$, the control horizon of the MPC strategy is over $[t_2,t_{11}]$. The control interval $[t_2,t_{10}]$ is the working hours of the first day which is from 9:00 to 18:00, and the control interval $[t_{10},t_{11}]$ is the first working hour of the second day. The desiccant wheel is switched off during the nonworking hours. Therefore,

$$H_{in}(t_{11})=F_4(H_i(t_{10}),Q_i,G,Hu_1), \tag{30}$$

where $F_4(H_i(t_{10}), Q_i, G, Hu_1)$ is given by the solution of ordinary differential Eq. (18). Hu_1 is the vector of outdoor humidity ratio during the first period of nonworking hours. When $k=n$, $n=2,3,\ldots,20$, the control horizon of the MPC strategy is over $[t_{1+n},t_{10+n}]$. The initial indoor humidity ratio is rewritten as

$$H_{in}(t_m)=\begin{cases}H_i(t_{m-1}) & \lceil m/11\rceil\neq 0,\\ F_4(H_i(t_{m-1}),Q_i,G,Hu_j) & \text{others},\ j=m/11.\end{cases} \tag{31}$$

To apply the MPC strategy, the OHCM is revised as follows

$$\begin{aligned}\min_{u_{ar},u_d,T_{ar},Q_{sa}}\ &\sum_{m=1+k}^{10+k}(1-\beta)(H_i(t_m)-H_s)^2+\beta(T_{sa}(t_m)-T_s)^2\\
\text{s.t.}\ &H_i(t_m)=F_1(H_{sa}(t_m),H_{in}(t_m),Q_{sa}(t_m),Q_i,H_o(t_m),G),\\
&H_{sa}(t_m)=F_2(T_{ap}(t_m),T_{ar}(t_m),H_o(t_m),u_{ar}(t_m),u_{ap}(t_m),u_d(t_m)),\\
&T_{sa}(t_m)=F_3(T_{ap}(t_m),T_{ar}(t_m),u_{ar}(t_m),u_{ap}(t_m),u_d(t_m)),\\
&H_{in}(t_m)=\begin{cases}H_i(t_{m-1}) & \lceil m/11\rceil\neq 0,\\ F_4(H_i(t_{m-1}),Q_i,G,Hu_j) & \textit{others},\end{cases}\\
&u_{ar}(t_m)=u_{ap}(t_m),\\
&u_{ar,lb}\leqslant u_{ar}(t_m)\leqslant u_{ar,ub},\\
&u_{d,lb}\leqslant u_d(t_m)\leqslant u_{d,ub},\\
&T_{lb}\leqslant T_{ar}(t_m)\leqslant T_{ub},\\
&Q_{lb}\leqslant Q_{sa}(t_m)\leqslant Q_{ub},\\
&1+k\leqslant m\leqslant 10+k,\quad \beta\in[0,1],\qquad j=m/11.\end{aligned} \tag{32}$$

Table 1
Settings of the OHCM.

Settings	Value	Settings	Value
H_s	0.0081481	Q_i	0.833
Q_{lb}	0	Q_{ub}	6
T_s	296.15	T_{lb}	293.15
T_{ub}	423.15	$u_{ar,lb}$	0
$u_{ar,ub}$	5	$u_{d,lb}$	0
$u_{d,ub}$	15	β	0.8

Table 2
Properties of desiccant wheel.

Parameters	Value	Parameters	Value
A_d	1 m^2	A_{de}/A_r	3:1
C_d	921 J/kg K	C_a	1007 J/kg K
C_v	1872 J/ kg K	C_{vl}	4186 J/kg K
f_d	0.7	h	38.3248 W/m^2 K
h_m	0.0381 kg/m^2 s	L	0.2 m
M_{sor}	2700 kJ/kg	ρ_a	1.1614 kg/m^3
ρ_d	720 kg/m^3		

Now the following MPC strategy is obtained.

(1) Initialize the conditions of desiccant wheel and indoor environment at time instant t_0, and let $k = 0$.
(2) Measure the outdoor temperature and humidity ratio at time instant t_k and solve the revised OHCM over time interval $[t_{k+1}, t_{k+10}]$ to find its optimal solution.
(3) The optimal solution of the revised OHCM at time instant t_{k+1} is denoted by α_{k+1}. Implement α_{k+1} to control the desiccant wheel, let $k = k + 1$ and go to Step (2).

4. Results and discussion

To illustrate the accuracy of the desiccant wheel model, the actual data from [16] is used. The maximal difference between the actual data and the estimated data of desiccant wheel model is 10.25%. The indoor humidity model is taken from and also validated in [17]. Consider a commercial building in South Africa, the volume of the building is 1500 m^3. During the period of working hours and nonworking hours it can be assumed that $G = 0.0060479$ and $G = 0$ respectively. The settings of the OHCM are given in Table 1. According to the ASHRAE standard [19], the usual requirement of indoor environment is determined by the volume of the building. To maintain the desiccant wheel working properly, the specifications of the desiccant wheel are given in Table 1. The properties of desiccant wheel are shown in Table 2 [16]. The conditions of desiccant wheel and indoor environment at time instant t_0 can be initialized as follows:

$$H_i(t_0) = H_s, \tag{33}$$

$$T_{air,in}(t_0) = T_s, \tag{34}$$

$$T_{ar,in}(t_0) = T_{lb}, \tag{35}$$

$$T_{ap,in}(t_0) = T_o, \tag{36}$$

$$u_{ar}(t_0) = u_{ar,lb}, \tag{37}$$

$$u_d(t_0) = u_{d,lb}, \tag{38}$$

$$Q_{air,in}(t_0) = Q_{lb}. \tag{39}$$

To illustrate the practical applications and robustness of the MPC strategy, the optimization of the desiccant wheel in this commercial building during the 5 days is studied. In this case study, assume that α_{k+1} in Step (3) of MPC strategy is replaced by $\alpha_{k+1} + w[k]$, where $w[k]$ is a noise vector and each of its components is generated randomly by the Matlab uniform distribution function rand (1)/20. The simulation results are shown in Figs. 2 and 3. The indoor environment is varied due to the variation of outdoor environment and the added noise in the optimal solution. The variation of indoor humidity ratio during working hours of the 5 days is shown in Fig. 2. In Fig. 2, the solid line is the profile of indoor humidity ratio. The red dash line is the profile of the humidity ratio set point. As shown in Fig. 2, the maximal indoor humidity ratio is 0.0084846. The minimal indoor humidity ratio is 0.0080854. The maximal difference between the indoor humidity ratio and its set point is 4.13% of the set humidity. Fig. 3 shows the variation of temperature of supply air during working hours of the 5 days. The solid line shows the variation of temperature and the dash line shows the profile of indoor temperature set point. As shown in Fig. 3, the temperature of

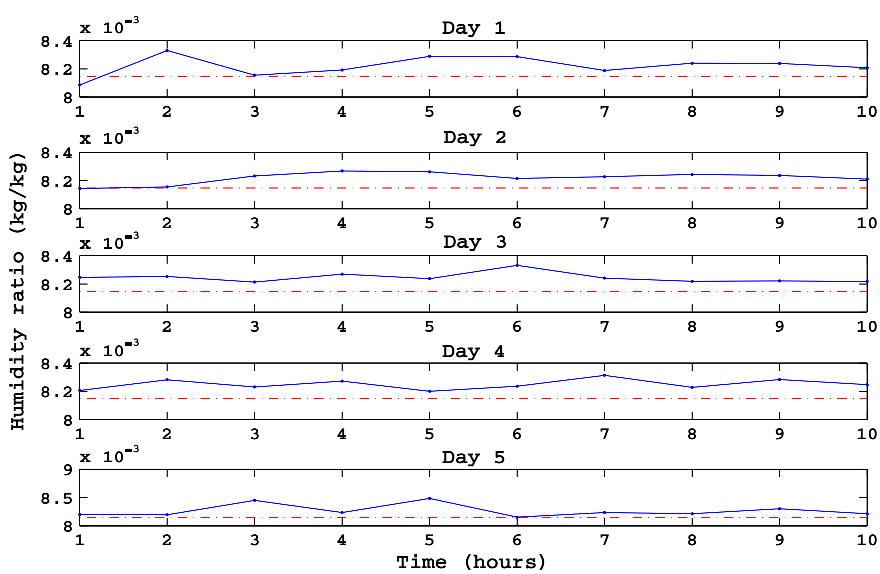

Fig. 2. Variation of indoor humidity ratio during working hours of the 5 days.

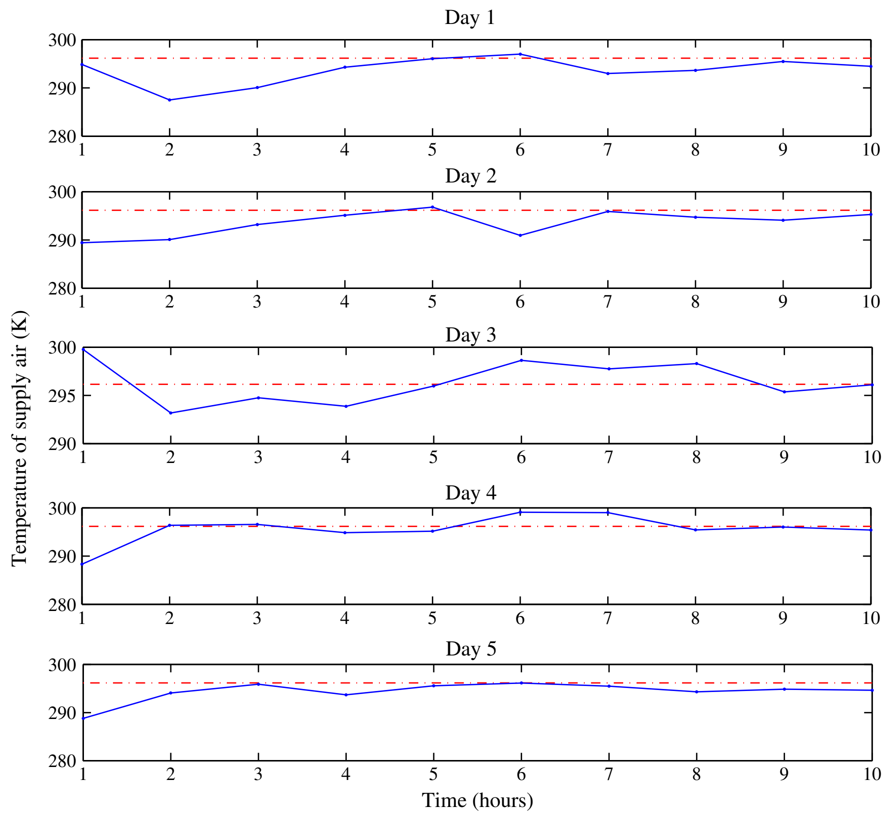

Fig. 3. Variation of temperature of supply air.

supply air is lower than the indoor temperature set point during most of the working hours, which implies that the obtained optimal solution can help the desiccant wheel to dehumidify and cool the indoor environment, and eventually reduce the indoor cooling load and save energy.

In additional, the energy consumption of the cooling system can be saved if the cooling system uses the desiccant wheel as a pre-cooling component. Note that there are four control variables in the humidity model: the velocity u_{ar} and temperature T_{ar} of inlet air at the regeneration section, the wheel speed u_d, and the flow rate Q_{sa} of the supply air. The velocity u_{ar} is determined by the speed of the fans at the regeneration section; the temperature T_{ar} is controlled by the electric heater; the wheel speed u_d is decided by the wheel's motor speed; and the flow rate Q_{sa} is adjusted by the supply fan. The proposed optimal humidity control model will give the optimal combination of the four control variables, and thus the electric power consumption of the regeneration fan, supply air fan, electric heater, and the wheel motor will be optimized to achieve the desired indoor humidity and temperature.

5. Conclusion

This paper presents an optimal humidity control model and provides the corresponding MPC strategy to implement the optimal operation solution of the desiccant wheel during working hours for commercial buildings. The optimization of the desiccant wheel control in a commercial building of South Africa is studied to illustrate the practical applications of the MPC strategy. The results show that the maximal difference between the indoor humidity ratio and its set point is 4.13% of the set humidity. The OHCM established in this paper considers only humidity and temperature differences in the objective function. A more complete model including the load caused by condensation and energy consumption of desiccant wheel will be considered in the future work.

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