ADVANCED OPEN-CYCLE DESICCANT COOLING SYSTEM

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ABSTRACT

The concept of staged regeneration as means of improving the desiccant cooling system performance is the subject of investigation in this study. In the staged regeneration, the regeneration section of desiccants dehumidifier is divided into two parts and only the latter fraction is subjected to the desorption air stream which has been heated to the desired regeneration temperature.

In the present work, the mathematical model describing the heat and mass transfer processes that occur during sorption of moisture in the desiccant dehumidifier includes both the gas-side (film) and solid-side resistances for heat and mass transports. The moisture diffusion in the desiccant material is expressed by gas-phase diffusion and surface diffusion. Effects of several parameters on the performance of desiccant cooling system with staged regeneration are investigated and the results of present model are compared with those of the lumped-resistance model.

Results of this study show that coefficient of performance of the desiccant cooling system can be substantially improved by using the staged regeneration concept. There is an optimum stage fraction and optimum cycle time for given system parameters and operating conditions. The results also indicate that the cooling system performance is higher than that predicted by the lumped-resistance model.

NOMENCLATURE

\(a_d\) thickness of desiccant, m
\(a_w\) thickness of supporting wall, m
\(A\) mass transfer area, m²
\(B_h\) heat transfer Biot number defined by Equation (16)
\(B_{num}\) overall mass transfer Biot number defined by Equation (20)
\(C_d\) specific heat of dry air, kJ/kg air °C
\(C_w\) specific heat of supporting wall, kJ/kg °C
\(COP\) thermal coefficient of performance
\(D_{eff}\) effective gas-phase diffusivity, m²/s
\(D_s\) effective surface diffusivity, m²/s
\(Q\) latent heat of vaporization of water, kJ/kg H₂O
\(h_p\) heat transfer coefficient, W/m²K
\(H_p\) enthalpy of process steam air, kJ/kg air
\(k_d\) apparent thermal conductivity of desiccant, kW/m°C
\(m_p\) mass transfer coefficient of process steam, kg/m²s
\(m_{th}\) mass flowrate of process steam air, kg/s
\(m_{tr}\) number of transfer unit
\(NTU\) overall mass transfer NTU based on film resistance defined by Equation (22)
\(NTU_{fm}\) overall mass transfer NTU based on Equation (16)
\(\rho\) heat of sorption, kJ/kg H₂O
\(r\) isotherm shape factor
\(RH\) relative humidity
\(SF\) stage fraction
\(t\) time, s
\(T\) temperature, °C
\(T_d\) desiccant temperature, °C
\(T_{do}\) initial desiccant temperature, °C
\(T_p\) temperature of process stream air, °C
\(T_{p,i}\) inlet temperature of process stream air, °C
\(T_w\) wall temperature, °C
\(U_m\) overall mass transfer coefficient, kJ/m²s
\(W_d\) moisture content in desiccant, kg H₂O/kg desiccant
\(x\) space variable in the flow direction, m
\(x_i\) channel length, m
\(X\) non-dimensional space variable defined by Equation (14)
\(Y\) non-dimensional channel length
\(y\) space variable in channel width direction, m
\(y_i\) channel width, m
\(\eta\) humidity ratio, kg H₂O/kg dry air
\(\eta_p\) humidity ratio in the gas-phase of desiccant, kg H₂O/kg dry air
\(\eta_{i,p}\) inlet humidity ratio in the gas-phase of desiccant, kg H₂O/kg dry air
\(\eta_{do}\) initial humidity ratio in the gas-phase of desiccant, kg H₂O/kg dry air
\(\eta_{p,i}\) humidity ratio of process stream air, kg H₂O/kg dry air
\(\eta_{p,i}\) relative humidity ratio of process stream air, kg H₂O/kg dry air
\(z\) space variable in channel height direction, m
\(z_{i}\) nondimensional space variable defined by Equation (15)
\(\rho_d\) density of desiccant
\(\rho_{d,a}\) air density, kg/m³
\(\rho_d\) density of supporting wall, kg/m³
\(\rho_s\) density of supporting wall, kg/m³
\(\rho_{s,a}\) air density, kg/m³
\(\phi\) nondimensional time defined by Equation (33)

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INTRODUCTION

An extension of desiccant dehumidification to cooling system was done in an emly development by Pennington (1). Destke (2) introduced a method of solar air-conditioning using an open-cycle desiccant system. Earlier researchers (3, 4) investigated the system with adiabatic dehumidifier operating with equal adsorption and desorption period.

Huske et al (5) and Collier and Cohen (6) showed that the system performance can be improved by adding inert material to increase the heat capacity of the desiccant matrices. Worek and Lavan (7) and Mathiprnkasan~ (8) studied the desiccant cooling system using cross-cooled dehumidifiers where the heat of sorption is removed by cooling air stream. The results indicated that the cooling capacity of the system is improved, since the desiccant is at lower temperature during adsorption process and therefore has higher sorption capacity.

In 1966, Glav (9) introduced a technique called staged regeneration for the rotary type air-conditioning apparatus. In this concept, the regeneration section of desiccant dehumidifier is divided into two parts and only the latter fraction is subjected to the desorption air stream which has been heated to the desired regeneration temperature.

The dynamic sorption analysis for the desiccant dehumidifier is important for the proper design of desiccant cooling system. Banks et al(10) introduced a linearized solutions using analogy method for the adiabatic dehumidifier. Roy and Gidaspow (11) used Green's function to get the nonlinear solution for the cross-cooled dehumidifier. A solution was also obtained by Mathiprnkasan and Lavan (12) using Laplace Transforms.

Many investigators (13, 14, 15, 16) used lumped-resistance models to describe the heat and mass transports occur during the sorption processes in the dehumidifier. These models do not include the solid-side resistances, and if necessary the solid-side resistances will be compensated by artificially increasing of film resistances.

In the present work, both film and solid-side resistances to heat and mass transports are considered to analyze the dynamic sorption in the dehumidifier, and the staged regeneration concept as means to improve the system performance is investigated. The effects of solid-side mass transfer resistance are also investigated and are compared to those obtained by the lumped-resistance model (13).

COOLING SYSTEM CONFIGURATION

Various cooling systems utilizing solid desiccants are discussed in (3). In this study, adiabatic dehumidifier desiccant cooling systems operating in the ventilation mode are considered.

Figure 1 shows the schematic diagram of the system and the corresponding psychrometric diagram is shown in Figure 2. In this system, the process air stream for the adsorption process is drawn from outside and is passed through the dehumidifier. The warm dry air leaving the dehumidifier enters the heat exchanger where it is sensibly cooled down to near room temperature. This air stream is then passed through an evaporative cooler where it is cooled and humidified before entering the conditioned room. During the desorption process, air is drawn from the conditioned room, humidified and cooled in an evaporative cooler and enters the heat exchanger in which it recovers the sensible heat from the adsorption air stream. This warm air stream in then heated to the desired regeneration temperature by a heat source, is passed through the dehumidifier to regenerate the desiccant material and is discarded to the outside.
processes are equal, the stage fraction can be defined by the process time fraction. Stage fraction equals to unity means that all the desorption air stream passes through the heat source, i.e. no staged regeneration.

**Figure 3.** Schematic diagram of desiccant cooling system with staged regeneration

**Figure 4.** Psychrometric diagram of desiccant cooling system with staged regeneration

**MATHEMATICAL MODEL**

The analysis of sorption processes in the desiccant dehumidifier requires a detailed description of the diffusion phenomena in porous desiccant materials which can be considered as a multi-phase system consisting of solid desiccant, sorbate and inert gas (air). Figure 5 shows the physical model for moisture transfer in the pores. When water molecules enter desiccant pores, some molecules will be attached to the surface of the pore, while others diffuse into the pore by gas-phase diffusion which can be either an ordinary diffusion or Knudsen diffusion depending on the pore diameter. The water molecules which have been attached in the pore surface then can diffuse into the pore along the wall of the pore and is called surface diffusion.

**Figure 5.** Physical model of mass transfer in desiccant pore

**Figure 6.** shows an element of a desiccant dehumidifier. The governing equations for mass and heat transfer can be described with the following assumptions:

1. The mass and heat diffusions in the channels are small compared to the convective transfer, thus can be neglected.
2. The temperature and concentration gradients across the channel are negligible.
3. The mass and heat transfer rates between the airstream and the solid desiccant material surface can be calculated from Nusselt and Sherwood numbers which are constant.
4. The nonhygroscopic supporting material is considered to be a good conducting material and the thermal contact resistance between supporting wall and the desiccant material can be neglected. Therefore its temperature is the same as the contacting desiccant material and is uniform in y and z directions.
5. The mass and heat diffusion in the y and z directions are neglected in the solid desiccant material.
6. Equilibrium always exists between the sorbate in the gas phase and the sorbent.

Using these assumptions, the governing equations describing the heat and mass transports in the dehumidifier can be derived (17) and are given as follows:

**Figure 6.** Schematic of desiccant dehumidifier element
The mass diffusion equation with the initial and boundary conditions,

\[ \frac{\partial C}{\partial t} + \frac{\partial (D \frac{\partial C}{\partial x})}{\partial x} = 0 \]

(1)

The thermal diffusion equation with the initial and boundary conditions,

\[ \frac{\partial T}{\partial t} + \frac{\partial (D \frac{\partial T}{\partial x})}{\partial x} = 0 \]

(2)

The mass balance equation and the inlet condition for process stream,

\[ \frac{\partial Y_d}{\partial t} + \frac{\partial (D \frac{\partial Y_d}{\partial x})}{\partial x} = \frac{\partial Y_d}{\partial x} \]

(3)

The heat balance equation and the inlet condition for process stream,

\[ \frac{\partial Y_q}{\partial t} + \frac{\partial (D \frac{\partial Y_q}{\partial x})}{\partial x} = \frac{\partial Y_q}{\partial x} \]

(4)

The governing Equations (I), (5) and (11) are nonlinear and coupled. These equations are rearranged using the following nondimensional independent variables:

\[ \tau = \frac{D t}{a^2} \]

(5)

\[ \chi = \frac{2K \theta y x}{m^2} \]

(6)

\[ Z = \frac{a}{d} \]

(7)

A finite difference technique is used with backward differences for time and central differences for space for the mass and thermal diffusion Equations (1) and (5) in the desiccant material. For Equations (9) and (11), a forward difference scheme is used.

RESULTS AND DISCUSSIONS

Numerical solutions to the governing equations were obtained using the operating conditions listed in Table 1. All of the results are presented for the cooling system and dehumidifier under periodic steady-state which is indicated by identical performance of successive cycles.

The cooling system performance is characterized by the thermal coefficient of performance (COP) and the cooling capacity. The thermal coefficient of performance is defined as the ratio of the cooling capacity to the thermal energy input. The cooling capacity of the system is defined as the enthalpy difference of the process air stream between the inlet and outlet of the conditioned space. Since adiabatic humidification evaporative coolers are used, the cooling capacity can be calculated from the evaporative cooler inlet conditions of the adsorption process and the evaporative cooler outlet conditions of the desorption process. The thermal energy input is calculated from the enthalpy difference of the desorption air stream across the heat source.

\[ Y_{d}(x, 0) = Y_{d}(x, 0) \]  

(8)

\[ Y_{q}(x, 0) = Y_{q}(x, 0) \]  

(9)

\[ \frac{\partial Y_{d}(x, 0)}{\partial t} = \frac{\partial Y_{d}(x, 0)}{\partial t} \]  

(10)

\[ \frac{\partial Y_{q}(x, 0)}{\partial t} = \frac{\partial Y_{q}(x, 0)}{\partial t} \]  

(11)

\[ T_{d}(x, 0) = T_{d}(x, 0) \]  

(12)
Table 1. Summary of system operation conditions

<table>
<thead>
<tr>
<th>Dehumidifier operation mode</th>
<th>Process stream cycle</th>
<th>Ambient temperature</th>
<th>Ambient humidity ratio</th>
<th>Regeneration temperature</th>
<th>Heat exchanger effectiveness</th>
<th>Evaporative cooler effectiveness</th>
<th>Conditioned room temperature</th>
<th>Conditioned room humidity ratio</th>
<th>Equilibrium relation*</th>
</tr>
</thead>
<tbody>
<tr>
<td>adiabatic</td>
<td>ventilation</td>
<td>35°C</td>
<td>0.0142 kg H2O/kg air</td>
<td>95°C</td>
<td>0.91</td>
<td>0.95</td>
<td>26.7°C</td>
<td>0.0111 kg H2O/kg air</td>
<td>r = 0.1</td>
</tr>
</tbody>
</table>

Maximum loading 0.4 kg H2O/kg desiccant
Heat of sorption $h_{so}$

*Equilibrium relation:

$$
\frac{W}{W_{max}} = \frac{1}{r + (1 - r)RH}
$$

Effect of Stage Fraction on the Performance:

Figures 7 and 8 show the effect of nondimensional cycle time on the COP and the cooling capacity for various stage fractions. There is an optimum cycle time for a given stage fraction. However, there is only a small difference in the system performance from the optimum cycle time.

The optimum COP with corresponding cooling capacity are shown as function of stage fraction in Figures 9 and 10 respectively. The COP increases with increasing SF up to SF=0.3 - 0.4 which is the optimum SF, then it decreases. But the cooling capacity continuously increases with increasing SF. The increasing rate of cooling capacity is large at small SF and it

Figure 7. Effect of nondimensional cycle time on COP with variable stage fraction

Figure 8. Effect of nondimensional cycle time on cooling capacity with variable stage fraction

Figure 9. Effect of stage fraction on optimum COP

Figure 10. Effect of stage fraction on cooling capacity at the cycle time of optimum COP
decreases with increasing SF. The cooling capacity increases continuously with increasing SF because the desiccant is drier with higher SF (more heating) after desorption process, therefore it is capable of absorbing more moisture which results in higher cooling capacity. At SF=1 (continuous heating), the COP is very low, about 0.63, and the cooling capacity is maximum, about 21. But at SF=0.4, the COP has nearly the maximum value, about 1.0 while the cooling capacity decreases to 17. Consequently, a COP increase by 37% can be expected in this case with a corresponding cooling capacity defect of only 20%.

Effect of Mass Transfer Biot Number

The diffusion of moisture in the desiccant material can be described by gas phase diffusion and surface diffusion. The mass transfer Biot number (\(Bi_{m}\)) which is based only on the gas-phase diffusion resistance is defined as

\[
Bi_{m} = \frac{K_{ad}}{\rho DG} 
\]

(16)

In order to investigate the effect of this parameter, two different cases are considered. One is for \(D_{s}=0\) and the other is for variable \(D_{s}\) proposed by Sladek et al (18) and is given by

\[
D_{s} = 1.6 \times 10^{-6} \exp (-0.97 D_{G})
\]

(17)

Figures 11 and 12 show the effect of mass transfer Biot number on the optimum COP and the cooling capacity at the cycle time of optimum COP for these two cases. The results indicate that the optimum COPs for both cases decrease with increasing mass transfer Biot number. The COPs of both cases are very close at the small \(Bi_{m}\) (less than 5). In this region, the surface diffusivity \(D_{s}\), which is given by Equation (17), is negligible as compared to \(DG\). In the region of large \(Bi_{m}\), where the surface diffusion is significant, the COP decreases very slowly with increasing \(Bi_{m}\), when \(D_{s}\) is considered and the COP decreases very rapidly when only the gas-phase diffusion is considered \((D_{s}=0)\). The cooling capacity curves of Figure 12 have similar trends as the COP curves.

Effect of Overall Mass Transfer NTU

The overall mass transfer NTU can be defined as

\[
NTU_{om} = \frac{\frac{D_{G}}{\rho DG}}{\frac{\omega}{\rho DG}}
\]

(18)

where the overall mass transfer coefficient \(U_{om}\) is given by

\[
\frac{1}{U_{om}} = \frac{1}{K_{y}} + \frac{\omega}{\rho DG} + \frac{D_{s}}{\rho DG} \frac{\omega}{\rho DG}
\]

(19)

Also, the overall mass transfer Biot number can be expressed as

\[
Bi_{om} = \frac{K_{y} \omega}{\rho DG + D_{s} \frac{\omega}{\rho DG}}
\]

(20)

Therefore the overall mass transfer NTU is related to the overall mass transfer Biot number by

\[
NTU_{om} = \frac{U_{om}}{1 + Bi_{om}}
\]

(21)

where \(NTU_{om}\) = mass transfer NTU based on film resistance only and is given by

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The overall mass transfer Biot number and the overall mass transfer NTU are calculated using the following parameters:

\[ \frac{K_f A}{\Delta p} = 2NTU_{fm} = 28 \]

\[ \frac{\Delta p}{\rho_d} \approx 3.15 \]

\[ \frac{K_f D_s}{\rho_d} = 1.684 \times 10^{-4} \]

The results are listed in Table 2 for variable \( D_s \) given by Equation (17) and in Table 3 for \( D_s = 0 \).

**Table 2. Relationship between mass transfer Biot number and overall mass transfer NTU for a given value of \( D_s \) and variable \( D_G \)**

<table>
<thead>
<tr>
<th>( B_{in} )</th>
<th>( D_G )</th>
<th>( \frac{\Delta x}{\rho_d} \frac{\Delta p}{\rho_d} )</th>
<th>( B_{imp} )</th>
<th>NTU_{fm}</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>( \infty )</td>
<td>0.61 \times 10^{-5}</td>
<td>0.00</td>
<td>14.00</td>
</tr>
<tr>
<td>1</td>
<td>0.165x10^{-3}</td>
<td>0.61 \times 10^{-5}</td>
<td>0.97</td>
<td>7.12</td>
</tr>
<tr>
<td>4</td>
<td>0.421x10^{-4}</td>
<td>0.61 \times 10^{-5}</td>
<td>3.50</td>
<td>3.11</td>
</tr>
<tr>
<td>8</td>
<td>0.210x10^{-4}</td>
<td>0.61 \times 10^{-5}</td>
<td>6.20</td>
<td>1.95</td>
</tr>
<tr>
<td>16</td>
<td>0.103x10^{-4}</td>
<td>0.61 \times 10^{-5}</td>
<td>10.13</td>
<td>1.36</td>
</tr>
<tr>
<td>32</td>
<td>0.526x10^{-5}</td>
<td>0.61 \times 10^{-5}</td>
<td>14.80</td>
<td>0.89</td>
</tr>
<tr>
<td>64</td>
<td>0.263x10^{-5}</td>
<td>0.61 \times 10^{-5}</td>
<td>19.30</td>
<td>0.69</td>
</tr>
</tbody>
</table>

**Table 3. Relationship between mass transfer Biot number and overall mass transfer NTU for \( D_s = 0 \) and variable \( D_G \)**

<table>
<thead>
<tr>
<th>( B_{in} )</th>
<th>( D_G )</th>
<th>( B_{imp} )</th>
<th>NTU_{fm}</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>( \infty )</td>
<td>0.00</td>
<td>14.00</td>
</tr>
<tr>
<td>0.5</td>
<td>0.336x10^{-3}</td>
<td>0.5</td>
<td>9.33</td>
</tr>
<tr>
<td>1</td>
<td>0.168x10^{-3}</td>
<td>1.0</td>
<td>7.02</td>
</tr>
<tr>
<td>2</td>
<td>0.842x10^{-3}</td>
<td>2.0</td>
<td>4.07</td>
</tr>
<tr>
<td>4</td>
<td>0.421x10^{-4}</td>
<td>4.0</td>
<td>2.80</td>
</tr>
<tr>
<td>8</td>
<td>0.210x10^{-4}</td>
<td>8.0</td>
<td>1.56</td>
</tr>
<tr>
<td>16</td>
<td>0.105x10^{-4}</td>
<td>16.0</td>
<td>0.82</td>
</tr>
<tr>
<td>32</td>
<td>0.526x10^{-5}</td>
<td>32.0</td>
<td>0.42</td>
</tr>
<tr>
<td>64</td>
<td>0.263x10^{-5}</td>
<td>64.0</td>
<td>0.22</td>
</tr>
</tbody>
</table>

Figure 13 shows the effect of overall mass transfer NTU on the thermal COP as compared to those of Collier (13) where the solid-side resistance is represented by an artificially increased film resistance. As can be seen, the lumped-resistance model used by Collier predicts a faster decrease in COP with decreasing NTU as compared to the results of present model which properly handles the internal resistances. In Collier's case, a COP decrease of 6% can be expected if the overall mass transfer NTU is reduced from 14 to 7. In the present simulation, the COP decreases by only 2% for the same NTU reduction.

Figure 14 shows the effect of overall mass transfer NTU on the cooling capacity. The cooling capacity decrease with overall mass transfer NTU is slightly less than the COP decrease.
It can be concluded that the cooling system performance degradation due to solid-side mass transfer resistance is much less than due to increased film resistance that yields the same overall resistance. The reason being that the effect of solid side resistance kicks in gradually in time program whereas the effect of film resistance is immediate.

CONCLUSIONS

A mathematical model, which considers both gas-side and solid-side resistances for heat and mass transports, is used to investigate the performance of an open-cycle desiccant cooling system with staged regeneration. The results show that the cooling system performance can be substantially increased using the staged regeneration concept. Under the same operating conditions, the COP is approximately 1.0 as compared to about 0.6 for the conventional system without staged regeneration.

The results indicate that the optimum cycle time is almost unaffected by the change in stage fraction. Also, the internal mass transfer resistances affect the changes of COP and cooling capacity in the same trend. Finally, there is a minimal effect of the solid-side resistance on the performance of the cooling system.

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