除濕輪空調系統之性能分析

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摘　　要

本文將固態除濕輪併入一直接蒸發冷卻系統中，為了探討除濕輪空調系統之性能，吾人建立了除濕輪之熱質傳方程式，且建立整個冷卻系統之數學模式並加以數值模擬，本研究並於再生過程中應用階段再生之觀念以提昇系統之性能係數。

雖然空調系統之性能係數在ARI之外界條件下可高達18.8，然而，本系統在台灣夏季應用時必須加以修正，才能維持舒適之室內空氣條件。本研究亦顯示在室外絕對濕度為0.03kg /kg時，除濕輪空調系統之性能係數通常會低於0.3。

關鍵字：除濕輪，空調，性能係數

The Performance Analysis of a Desiccant Air Conditioning System

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ABSTRACT

A commercial desiccant wheel is incorporated into the DINC (direct-indirect evaprative cooling) system. In order to examine the performance of this desiccant cooling system, the heat and mass transfers of desiccant wheel and the whole cooling system are mathematically modelled and numerically calculated. In this study, the concept of "staged regeneration" is employed for the regeneration process to promote the system's COP(coefficient of performance).

Although th COP of the cooling system can be as high as 1.88 under the ARI(American Refrigeration Institution) conditions. Under the condition of mid-summer weather in Taiwan, the same cooling system has to be modified such that the desired comfortable indoor condition can be maintaned. The study also reveals that the COP of desiccant cooling system is usually less than 0.3 when the outdoor humidity ratio is as high as 0.03 kg/kg.

Key word: desiccant, air conditioning, coefficient of performance

INTRODUCTION

As a result of expanding markets for air-conditioning products, there is a dramatic increase in the demand for electricity. According to Taipower's report, the national reserve electricity is going to dry out anytime in the near future unless proper actions (such as energy conservation policy, the construction permit for new power generation etc.) have to be enforced immediately.

One way to relief the tension of electricity shortage is to suppress the peak demanding for electricity which usually occur during the mid-summer time when more and more people are enjoying their air conditioning facilities.

Like absorption cooling system, desiccant cooling system is also able to achieve the thermal comfort conditions by consuming less electricity than the traditional (compressor operated) cooling system.

Steady progress in the development of high-performance solid desiccant dehumidifiers has resulted in desiccant-based air conditioning systems that are becoming competitive with conventional vapor compression machines. In desiccant air conditioning systems, air is dried by passing it over the desiccant and the heat of sorption is removed by sensible cooling. The air is further cooled by adiabatic cooler and is directed into the residence as cool dry air.

Analytical models for the flow of moist air through desiccant wheel have been developed based on the governing equations for conservation of heat and mass. Maclaine-cross and Banks[1], Banks[2] developed an analogy method to predict the exit fluid temperature and humidity of a desiccant wheel. The analogy theory relates the performance of a rotary heat and mass exchanger to a superposition of two analogous regenerators. In this method, Le is limited to unity. Maclaine-cross[3] and Holmberg[4,5] use finite difference method to solve the heat and mass equation for modeling rotary dehumidifiers. This method is superior in accuracy to the analogy-theory based models, but it takes more CPU time. Barlow (1982) has proposed a pseudo-steady-state model which considers discrete sections of the dehumidifier to act as simple steady-state heat and mass exchanger . Schultz has made a comparison of pseudo-steady-state model with a finite difference solutions, the comparison indicates that the pseudo-steady-state model can produce accurate results when used carefully, but at greater computational expense.

About desiccant cooling systems, an advanced desiccant cooling system DINC (direct-indirect evaporative cooling) is proposed by Waugaman[6] in 1987. It replaces one direct evaporative cooler at traditional Pennington cycle with an indirect evaporative cooler and redirects the flows. Collier[7] has been working on a idea of "staged regeneration" for low-heat capacity desiccant wheel to improve system efficiency. At constant regeneration temperature and rotation speed for desiccant wheel, Waugaman progressed a simulation for DINC system in a building in Texas. The results show that the average thermal COP can reach to 1.44, overall COP was 1.1 .

In those mentioned literatures for system simulation, there were no discussion about optimization for desiccant parameters (such as regeneration temperature, rotation speed and so on). This study thus focus on system efficiency. A finite difference method will be used to solve the governing equations for rotary dehumidifier. A simulation represents the high-humid weather in Taiwan will be shown for the modified DINC system.

MODELING OF ADIABATIC DESICCANT WHEEL

The governing equations for the combined heat and mass transfer in the rotary desiccant wheel can be expressed as follows[3][5]:

$$\frac{\partial y}{\partial x} = NTU_{t,j}\left[\frac{1}{Le}\right](y_w - y) \tag{1}$$

$$\frac{\partial T}{\partial x} = NTU_{t,j}(T_w - T) \tag{2}$$

$$(C_m + C_w W)\frac{\partial T_w}{\partial \tau} = \left[\frac{NTU_{t,j}}{\Gamma_j}\right]C_p\left[(T - T_w) + \frac{Q}{LeC_p}(y - y_w)\right] \tag{3}$$

$$\frac{\partial W}{\partial \tau} = \left[\frac{NTU_{t,j}}{\Gamma_j}\right]\left[\frac{1}{Le}\right](y - y_w) \tag{4}$$

definition of the dimensionless parameters:

$x = Z / L \qquad 0 \leq x \leq 1$

$\tau = \theta / \theta_j \qquad 0 \leq \tau \leq 1$

assume constant rotation speed, the desiccant rotation angle,θ,can thus be considered as the time scale, see Fig. 1

$NTU_{w,j} = K_m A_j / \dot{m}_j$, $NTU_{t,j} = K_h A_j / C_p \dot{m}_j$

$Le = \dfrac{NTU_{t,j}}{NTU_{w,j}} = \dfrac{K_h}{K_m C_p}$, $\Gamma_j = \dfrac{M_{dj}}{\theta_j m_j} = \dfrac{M_d}{\theta_T \dot{m}_j}$

when the subscript j=p stands for process period, and j=r is the regeneration period.

Eq.(1) represents the conservation of mass of the vapor in the moist air, and Eq.(2) is the conservation of energy of the moist air. Eq.(3) represents the energy balance of the desiccant matrix and Eq.(4) is the mass balance of the water vapor in the desiccant matrix. In the above equations, Q is the heat of adsorption and it can be obtained from Clausius-Clapeyron equation, Le is the Lewis number and assumed to be unity in this paper. Eq.(1)-(4) are coupled through the thermodynamics property relationships for the desiccant-air-water vapor mixture.

The initial boundary conditions for this system of equations are

$y(x = 0, \tau) = y_{ji}$

$T(x = 0, \tau) = T_{ji} \qquad 0 \leq \tau \leq 1 \;;\quad j = p$ or r

The periodic equilibrium boundary conditions for the matrix state properties are

for $0 \leq x \leq 1$

$$\lim_{\tau_p \to 1^-} W(x, \tau_p) = \lim_{\tau_r \to 0^+} W(1 - x, \tau_r)$$

$$\lim_{\tau_p \to 1^-} T_w(x, \tau_p) = \lim_{\tau_r \to 0^+} T_w(1 - x, \tau_r)$$

$$\lim_{\tau_p \to 0^+} W(x, \tau_p) = \lim_{\tau_r \to 1^-} W(1 - x, \tau_r)$$

$$\lim_{\tau_p \to 0^+} T_w(x, \tau_p) = \lim_{\tau_r \to 1^-} T_w(1 - x, \tau_r)$$

The above equations and initial/boundary conditions are based on the following assumptions for the analyzed desiccant wheel:

(1) The substrate of desiccant wheel is homogenous.
(2) The air flows through the wheel is a fully-developed laminar flow (air channel is small), and one-dimensional.
(3) The heat conduction and water vapor diffusion in axial flow direction are negligible.
(4) The rotation speed of desiccant wheel is so small that the carryover (heat or mass) between the process side and regeneration side is negligible.
(5) There is no heat and mass losses through the outer-body of desiccant wheel.
(6) The rotation speed of desiccant wheel is constant, and the mass/heat transfer between the air flow and desiccant body reaches to a periodic steady-state performance.

(A) DESICCANT WHEEL WITH STAGED REGENERATION

A technique[7] so-called "staged regeneration" is used in balanced flow designs for improving the COP of desiccant cooling systems. The basic idea of this concept (shown in Fig.2) is to heat only a fraction of the regeneration air stream to the maximum regeneration temperature. The regeneration process therefore consists of two stages:

- The first stage is regeneration by air exiting the sensible heat exchanger with no additional heating.
- The second stage is regeneration by the remainder of the air exiting the sensible heat exchanger with enough additional heating to bring this air to the desired regeneration temperature that will sustain the dehumidification side of the cycle.

When modeling the cooling system, the concept "staged regeneration" will be used. In reference[7], there wasn't discussion about how to choose proper values for T_{ri1} 、 T_{ri2} and re. Here we will find the proper range for these valuables by system modeling.

(B) OPTIMUM ROTATION SPEED OF A DESICCANT WHEEL

The optimum rotation speed discussed hereby means the speed when outlet humidity ratio becomes lowest on the process side with no consideration about outlet temperature. It takes more heat to make process outlet humidity ratio lower and it results in higher outlet temperature. However it is not a big concern, since the air will be cooled later when it passes through the heat exchanger, indirect evaporator and direct evaporator. Therefore, the humidity control is more important than temperature control.

We may make a list of all the variables which affect the optimum rotation speed (1) basic physical characteristics of desiccant wheel (such as specific heat, specific weight ... etc.) (2) geography shape of desiccant wheel (3) inlet temperature, humidity, flow rate on process side (4) inlet temperature, humidity, flow rate on regeneration side.

The first group of the above variables related to the physical properties of desiccant material, which are shown in the literature[8]. The second group related to the geometry of desiccant wheel here we take diameter=1.0 meter, thickness=0.4 meter. This represents the cooling capacity of generating 5-8 tons when the desiccant wheel is put into a cooling system. With the change of the remaining variables, the influence of outlet humidity ratio on process side was found by computer modeling. After several trials, the inlet temperature and humidity ratio on regeneration side, and the flow rate most influence the outlet humidity ratio on process side. Since the desiccant wheel in this text is a balanced flow, therefore, it's not necessary to separate the inlet flow rate of process side from that of regeneration side.

Picking the regeneration temperature at 70, 80, 90, 100, 110℃ and flow rate at 0.8, 0.9, 1.0, 1.1, 1.2(kg /sec), the corresponding optimum rotation speed was found by computer modeling, as in Fig.3. The inlet humidity ratio on both process and regeneration sides are 0.014kg/kg and the inlet temperature on process side is 35°C.

In order to make Fig.3 applicable on the desiccant wheel with staged regeneration, we can find the optimum rotation speed by taking T_{av} from

$$T_{av} = (1-re)(T_{H1}) + re(T_{H2}) \tag{5}$$

As the regeneration temperature obtained from Eq.(5), the optimum rotation speed of desiccant wheel with staged regeneration is very close to the result numerical simulation.

When the inlet humidity ratio is larger than 0.016kg/kg, the predicted value from Fig.3 is not so accurate. After many tests, the relationship shown in Eq.(6) is used to modify the rotation speed when high humidity ratio is considered, in which $(rph)_c$ is the optimum rotation speed by numerical simulation, $(rph)_o$ is the optimum rotation speed obtained from Fig.3, y_{pi} is the inlet humidity ratio on process side.

$$\frac{(rph)c}{(rph)o} = 1.0 - 0.25 \times \frac{(rph)_c}{(rph)_o} = 1.0 - 0.25 \times \frac{y_{pi} - 0.014}{0.014} \tag{6}$$

This equation can make an estimate for optimum rotation speed under the condition of higher inlet humidity ratio. According to flow rate and regeneration temperature, $(rph)_o$ can be obtained from Fig.3 and then substitute it into Eq(6) along with y_{pi}, then we can get the optimum rotation speed for high inlet humidity ratio $(rph)_c$.

COMPUTER MODELING OF THE DINC SYSTEM

To analyze the performance of the DINC system, it is necessary to know about the relation between every system component. The relation will be found by the definition of efficiency and the conservation heat and mass for system components. The efficiency definitions for sensible heat exchanger E_h, indirect evaporative cooler E_{ide} and direct evaporative cooler E_{de} are expressed as follows, see Fig.4

$$E_h = \frac{T_7 - T_1}{T_2 - T_1} \tag{7}$$

$$E_{ide} = \frac{T_{3m} - T_{44}}{T_{3m} - T^*_5} \tag{8}$$

$$E_{de} = \frac{y_4 - y_{44}}{y_{s2} - y_{44}} \tag{9}$$

In the condition space, the humidity ratio and enthalpy can be obtained by the heat mass conservation equations.

$$y_5 = \frac{M_{\inf} y_1 + M_4 y_4 + y_{gen}}{M_{\inf} + M_4} \tag{10}$$

$$h_5 = \frac{Q_{load} + M_{\inf} h_1 + M_4 h_4}{M_{\inf} + M_4} \tag{11}$$

The COP of the system with staged regeneration is defined as

$$COP = \frac{Q_{load}}{M_1 \times re \times (h_8 - h_7)} \tag{12}$$

Combining the desiccant wheel modeling program with the efficiency definition and heat and mass conservation equations for system component, the whole system simulation can be obtained. The optimum rotation speed mentioned before will be applied to the system simulation. The calculation will be continue until the results to meet the indoor requirement are satisfied by changing the flow rate. At the same time, the value of "re" and temperature difference between two stages are varied in the similar way, we will see how these variables affect the COP value in the next section.

The input condition is listed in table 1. The desired indoor condition is at dry-bulb temperature of 24-27°C and the humidity ratio of 0.006-0.012kg/kg.

RESULTS

The results for COP value larger than 1.2 are listed in Table 2 . From Table 2, we can summarize that

1. The COP values are smaller than 1.2 at re=0.8 to 0.9, so desiccant wheel with staged regeneration can indeed improve the system performance. As the re value becomes smaller the system performance becomes higher, but the simulation results can not meet indoor desired condition when re is small than 0.3. The proper value of re should be 0.4-0.7.
2. The simulation results show that the efficiency of sensible heat exchanger needed in the system is quite high. The highest value of COP is 1.7, but the corresponding E_h is 0.98. The reasonable result is at COP=1.41 and the corresponding E_h is 0.9. The psychometric chart is demonstrated in Fig.5.
3. The results also show that the temperature difference between T_7 and T_8 are around 25 and 30, especially the temperature difference values for most of the results are equal to 30 when re is smaller than 0.6. For high efficiency system, the temperature difference value between T_7 and T_8 should be greater than 25. Further simulation indicates that the COP value becomes smaller when the temperature difference value between T_7 and T_8 is large than 35 and the corresponding E_h is still high. The E_h value can be low as 0.7 and its corresponding COP is only 0.81.

MODELING OF THE MODIFIED DINC SYSTEM

In Taiwan, humidity is in the high bound and it is also very hot in summer. For example, the dry-bulb temperature can be 36.5°C and the monthly average relative humidity is as high as 83% . Obviously, the system should be modified in order to apply the DINC system in Taiwan. As depicted in Fig.6, the modified DINC system has two stages dehumidification process.

CS is condition space, DE is direct evaporative cooler, IDE is indirect evaporative cooler, HE is sensible heat exchanger, HS is heat source, and DH is desiccant wheel. Comparing the modified DINC with the original DINC system, new DINC has extra HE,HS and DH. Which means there is a two-stage dehumidification process. The COP of modified DINC system can be expressed as

$$COP = \frac{Q_{load}}{M_1 \times re_1 \times (h_{11} - h_{10}) + M_1 \times re_2 \times (h_8 - h_7)} \tag{13}$$

For the simulation of modified DINC system, the indoor condition and component efficiency are the same as that in Table 1, and the new input conditions are listed in Table 3. To deal with high humidity air, we increase the flow rate. The temperature difference values between T_8, T_7 and T_{11}, T_{10} are taken according to the experience obtained from the previous results.

Because it requires a higher regeneration temperature to deal with high humidity air. As a result, the efficiency of corresponding sensible heat exchanger will exceed 1.0 . To solve this problem, the regeneration air is heated at two stages as shown in Fig.7.

The simulation is progressed again after this two-stage modification. In the meantime, E_{de} changes to 0.5, E_{ide} is 0.8, the input conditions for desiccant wheels are shown in Table 4.The COP value can be calculated from below

$$COP = \frac{Q_{load}}{M_1 \times \left[re_1 \times (h_{11} - h_{10a}) + re_2 \times (h_8 - h_{7a}) + h_{10a} - h_{10} + h_{7a} - h_7\right]} \tag{14}$$

The results which can meet the indoor desired are listed in Table 5.
In all of the test cases, the highest COP value is 0.3 and the associated psychometric chart is demonstrated in Fig.8.

CONCLUSION

The purpose of this study tries to investigate the thermal performance of a desiccant cooling system for the typical summer weather in Taiwan. A standard SSCR (Seibu Giken) silica dehumidifier is incorporated in DINC cooling system. Under the ARI indoor and outdoor conditions, the COP as high as 1.88 can be reached. However, if the outdoor condition is replaced by the mid-summer outdoor weather in Taiwan (at 35 ℃ and y=0.030kg/kg), the same ARI indoor condition (considered as thermal comfort) can not be achieved unless the modification to the DINC system is made. In this study, a two-stage dehumidification is employed to obtained the desired indoor condition. The corresponding COP of this two-stage dehumidification system is however less than 0.3.

Currently, the high initial cost of desiccant cooling system withholds its advancement and popularity. The lower-cost dehumidifier with higher performance (such as super SSCR, a new product) matches with high efficient heat exchanger will make the desiccant cooling system more attractive especially when the energy for regeneration coming from waste heat.

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Indoor Condition
Q_{load}=25kW, y_{gen}=0.001kg/sec, M_{inf}=0.2kg/sec, f=0.6
Outdoor Condition
T_1=35.0 C, y_1=0.014kg/kg (ARI outdoor condition)
Desiccant Wheel Parameters
M_1=0.8 to 1.2kg/sec, interval 0.5; re=0.3 to 0.9, interval 0.1;T_8=90 to 120 C, interval 5 C; T_7=(T_8-30) to (T_8-10) C, interval 5 C
Component Efficiency
E_h is calculated from Eq.(8). When the valus of E_h over 1.0, the results will not be used. E_{de}=0.7, E_{ide}=0.7

Table 1 Input Conditions

re=0.7					
T_7	T_8	M_1	E_h	opt.rph	COP
70	95	1.05	0.97	19.2	1.33
65	90	1.10	0.93	15.9	1.33
70	95	1.10	0.98	18.0	1.29
65	90	1.15	0.95	19.1	1.23
re=0.6					
70	100	1.05	0.95	19.9	1.32
75	105	1.05	0.97	21.4	1.31
70	100	1.10	0.95	20.7	1.29
75	105	1.10	0.97	22.5	1.27
65	90	1.20	0.95	13.6	1.31
re=0.5					
65	95	1.10	0.96	16.7	1.49
70	100	1.10	0.98	19.2	1.47
65	95	1.15	0.97	17.3	1.41
60	90	1.20	0.91	15.4	1.35

Table2 COP value larger than 1.2

Outdoor Condition	
T_1=35.0°C y_1=0.030kg/kg	
Desiccant Wheel Parameters	
M_1=1.0 to 1.4 kg/sec, interval 0.5	
re_1=0.6, re_2=0.6	re_1=0.8, re_2=0.7
T_8=90 to 120°C, interval 5°C	
T_7=T_8-30°C	T_7=T_8-20°C
T_{11}=90 to 120°C, interval 5°C	
T_{10}=T_8-30°C	T_{10}=T_8-25°C

Table 3 Input conditions of modified DINC system

Desiccant Wheel Parameters
M_1=1.0 to 1.3 kg/sec, interval 0.5
re_1=0.6, re_2=0.6
T_8=100 to 130°C, interval 5°C
T_{7a}=T_8-30°C
T_7=T_{7a}-15°C
T_{11}=100 to 130°C, interval 5°C
T_{10a}=T_{11}-30°C
T_{10}=T_{10a}-8°C

Table 4 Input conditions of desiccant wheels

T_{11}	T_{10a}	rph_1	E_{h1}	T_8	T_{7a}	rph_2	E_{h2}	M_1	COP
125	95	26.8	0.87	130	100	40.5	0.87	1.25	0.30
130	100	27.6	0.88	130	100	40.5	0.88	1.2	0.31
130	100	28.9	0.88	125	95	37.7	0.88	1.25	0.29
125	95	28.0	0.87	130	100	42.2	0.87	1.30	0.29

Table 5 COP value of the test cases