THEORETICAL ANALYSIS OF A LIQUID DESICCANT AIR DEHUMIDIFIER

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ABSTRACT

An efficient design configuration of a liquid desiccant air dehumidifier, to be used in liquid desiccant air conditioning systems powered by solar or any low thermal energy sources, has been investigated in this paper. The proposed dehumidifier consists of a non-adiabatic contact tower, packed with finned tube heat exchangers and cooled with available water.

A simplified theoretical model is used to investigate the complex phenomena of simultaneous heat and mass transfer, as well as the interaction between the air dehumidification and the used desiccant regeneration processes. This theoretical model is based on the logarithmic mean differences of the inlet and outlet thermodynamic properties of the three working fluids: the water, the moist air and the desiccant solution.

The system performance is predicted through parametric studies using a computer simulation. The results show that the cooling of the air dehumidification process enhances the mass transfer, makes the operation more efficient and compact and reduces the desiccant strong solution requirements. Obvious consequences are the lower regenerating temperature requirements and the subsequent higher performance of the solar powered liquid desiccant air conditioning systems.

Keywords: *absorption, liquid desiccant, air-dehumidification, heat and mass transfer, solar energy.*

الملخص

يعرض في هذا البحث تصميم لمجفف للهواء الرطب، يعتمد على المجففات السائلة ويمكن إستعماله في منظومات تكييف الهواء التي تعمل بمثل هذه السوائل وبالطاقة الشمسية أوأي مصدر طاقة حرارية منخفضة. هذا المجفف يتكون من برج غير أديباتيكي معبأ بملفات تبريد ذات زعانف ويبرد بالماء.

تمت دراسة عملية إنتقال الحرارة والمادة المتزامن والمعقد وتداخل عمليتي تجفيف الهواء وتنشيط المحلول المجفف عبر نموذج نظري مبسط، يعتمد على متوسط فوارق الخواص الحرارية اللوغرتمية للموائع المستعملة الثلاثة: الماء، الهواء الرطب والمحلول المجفف.

التنبؤ بأداء المجفف وقع عبردراسة مدى تأثير مختلف العوامل واستعمال الحاسب الآلي. النتائج تؤكد على أن تبريد عملية التجفيف يحسن من كفاءتها ويقاص من أبعاد المجفف، كما أنه يخفض من متطلبات تركيز محلول التجفيف,وبالتالي تكون درجات حرارة تنشيط السائل المجفف أصغر وأداء المجمعات الشمسية أحسن في حالة إستعمالها.

1. INTRODUCTION

Solar air-conditioning based on liquid-desiccant dehumidification and subsequent evaporative-cooling appear to be promising, yet not fully explored. Liquid desiccant systems have been commercially available; they have primarily been used to provide dehumidification, heat recovery and dust and bacteria control. Lof [1955] was the first to suggest a solar powered liquid desiccant air-conditioning system. Further investigations are reported by Kapur [1960], , Mullick and Gupta [1976], Olsen [1976], Lodwig et al. [1977], Robison [1978], Johanson [1978], Peng [1981], Queiroz et al. [1984], Chebbah [1987], Spears and Judge [1997], Chebbah [2000] and others.

One of the key components of a liquid desiccant air-conditioner is the dehumidifier in which moist air is dried on physical contact with a strong desiccant solution such as tri-ethylene glycol or lithium bromide. Several design configurations have been suggested for the dehumidifier: adiabatic spray chamber [Grossman et al., 1977], adiabatic packed-bed column [Leboeuf et al., 1980 and Factor et al., 1980] and the water-cooled chamber packed with finned tube heat exchangers [Turner,1979, Peng, 1981 and Chebbah, 1987]. The last design configuration permits to reject the latent heat of the absorbed moisture to the cooling water, and thus maintains the desiccant cool during the dehumidification process. This leads to a lower water vapor pressure in the desiccant solution and therefore larger mass transfer driving potential and lower desiccant concentration requirements

In this paper, a water-cooled liquid desiccant dehumidifier, packed with finned-tube heat exchangers, is investigated. A simplified theoretical model is used for the analysis of the complex phenomena of simultaneous heat and mass transfer between the three working fluids: moist air, liquid desiccant and cooling water.

2. THEORETICAL MODEL

Consider the general case of a non-adiabatic absorber, which consists of a column packed with finned tube heat exchangers, as shown in figure 1. A gas mixture, of a solvent mass flow rate Gs, a solute to solvent mass ratio Y'_1, and at a temperature T_{G1}, is blown upwards through the contactor. At the same time, a liquid solution of a solvent mass flow rate Ls and a solute mass ratio X'_2, enters at the top at a temperature $T_{L,in}$. The solution is sprayed on the outer surface of the finned tube heat exchangers, and trickles down in the form of a thin falling film, countercurrent to the rising gas. Note that in liquid desiccant air-dehumidifiers, the solute corresponds to the moisture, while the liquid and gas solvents represent the desiccant and dry air respectively.

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To control the temperature inside the tower, process water at a temperature T_{w2} and a flow rate M_w is simultaneously pumped through the finned tubes to exchange thermal energy with both liquid and gas phases. At any section of the contactor, boundary layers are developed, and simultaneous heat and mass transfers takes place. As a result, the gas leaves at a mass ratio Y'_2 and a temperature T_{G2}, while the liquid drains to the bottom of the contactor at a composition X'_1 and a temperature T_{L1}, and the process water flows out at a temperature T_{w1}.

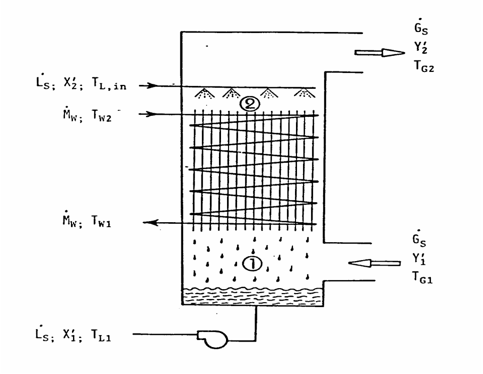

Figure 1: Non- adiabatic column packed with finned-tube heat exchangers.

The simplified theoretical model, for the analysis of non-adiabatic and nearly isothermal contactors is developed and experimentally checked by Chebbah [1987]. This model assumes linear equilibrium properties X'and Y' in terms of the liquid/gas interface temperature. It is used in this investigation to simulate the air dehumidification process. The simultaneous heat and mass transfer are hence controlled by the following set of equations:

- Integral equation of simultaneous heat and mass transfer:

$$Q_L = M_L C_{L,W} (T_{L2} - T_{L1}) = - K_Y \; E \; A_O \; \Delta H^*_{lm} \qquad (2.1)$$

where H^* and ΔH^*_{lm} are defined by:

$$H* = Cp_G T + \frac{F}{Le^{2/3}} Y'(\lambda' = \Delta H'_m) \qquad (2.1.a)$$

$$\Delta H^*_{lm} = \frac{(H^*_{G2} - H^*_{i2}) - (H^*_{G1} - H^*_{i1})}{\ln[(H^*_{G2} - H^*_{i2}) / (H^*_{G1} - H^*_{i1})]} \qquad (2.1.b)$$

- Integral equation of mass transfer:

$$M_A = G_S (Y'_2 - Y'_1) = - K_Y \; A_o F \, Y'_{lm} \tag{2.2}$$

where:

$$\Delta Y'_{lm} = \frac{(Y'_2 - Y'_1) - (Y'_2 - Y'_{i1})}{\ln[(Y'_2 - Y'_1)/(Y'_2 - Y'_{i1})]} \tag{2.2.a}$$

- Solute (moisture) mass balance:

$$L_s (X'_2 - X'_1) = G_s (Y'_2 - Y'_1) \tag{2.3}$$

- Energy balance:

$$M_W \, Cp_W (T_{W2} - T_{W1}) + L_S \, C_{S,L} (T_{L,in} - T_{L,1}) = G_S[C_G(T_{G2} - T_{G1}) + (Y'_2 - Y'_1)(\lambda^{'} + \Delta H_m)] \tag{2.4}$$

- Energy exchange with the process water:

$$M_W \, Cp_W \; (T_{W2} - T_{W1}) = h_W \, A_W \, \Delta T_{L,W} \tag{2.5}$$

where:

$$\Delta T_{L,W} = (T_{L2} - T_{W2}) = (T_{L1} - T_{W1}) \tag{2.5.a}$$

Equations (2.1) through (2.5) are sufficient to completely analyze the overall behavior of the simultaneous heat and mass transfer in a nearly isothermal absorber. By successive substitutions, the above system of equations can ultimately be reduced to the following system of two nonlinear equations:

$$f(T_{G2}, \; Y'_2) = Q_L + K_Y \; E \, A_O \; \Delta H^*_{lm} \tag{2.6}$$

$$g(T_{G2}, Y'_2) = G_S (Y'_2 - Y'_1) + K_Y \; A_o F \, Y'_{lm} \tag{2.7}$$

For the small range of working concentration, an average heat of mixing $\Delta H'_m = 24.3$ Btu/lb (2.36 Kj/Kg) of absorbed water was deduced for the tri-ethylene glycol used. The interface equilibrium properties as well as the latent heat $\lambda^{'}$ of evaporation are calculated in terms of temperature and/or desiccant concentration using appropriate empirical correlations. The heat and mass transfer coefficients of the wetted finned-tube heat exchanger are computed using the correlations developed by McQuiston [1978] and which account for the geometric and hydrodynamic variables.

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The above system of nonlinear equations may then be solved using Newton-Raphson successive iterations technique. Because of the large amount of computation involved, a computer program was written to solve this system and to investigate the dehumidification process.

3. RESULTS AND DISCUSSION

Considering the complex interaction between the numerous controlling parameters, the analysis was conducted on a one square foot cross-section basis. The inlet air conditions of the dehumidifier were taken identical to those of the return air, namely: 78 ºF (25.5 ºC) dry bulb temperature and 50% relative humidity. The cooling water was assumed to enter the contactor at 70 ºF (21ºC), which is an estimation of available ground water temperature. The tri-ethylene glycol solution was considered regenerated to a 96% concentration by weight, which is feasible with hot water at 150 ºF (65.5 ºC) that can be supplied by today's flat plate collectors. The hot and concentrated desiccant, coming from the regenerator, was assumed pre-cooled to 90 ºF (32 ºC) by a liquid-to-liquid intercooler. The other parameters were selected or allowed to vary within specified ranges, based on both practical and economical considerations.

An air face velocity of 365 ft/mn (1,85 m/sec) was chosen to maintain good heat and mass transfer properties at reasonable blower power costs. A minimum desiccant cross-section flow rate of about 200 Lb/hr.ft^2 (976 Kg/hr.m^2) is judged necessary for an effective and even spray. The maximum flow rate is, however limited by the flooding conditions, which are also controlled by the coil geometry and air flow rates.

In order to maintain the absorber as cool as possible for better mass transfer, the heat capacity of the process water should be higher than that of both air and desiccant, combined. A minimum water flow rate of 1600 Lb/hr.ft^2 (2.17 Kg/sec.m^2) cross section was therefore selected, while practical (noise) and economical considerations were used to determine the maximum.

Numerous computer runs were performed for the analysis of the dehumidifier, allowing the controlling parameters to vary, one at the time. The results are summarized and discussed below.

3.1 Effect Of Desiccant Flow Rate And Packing Depth

Figures 2 through 5 illustrate the effect of the desiccant flow rate (MD) and the packing depth (NR) on the different output parameters of interest. Figure 2 gives the leaving air humidity (W2A). It shows that, for a given desiccant flow rate, more dehumidification occurs with deeper packing, due to the larger moist-air/desiccant contact area for mass transfer. It also

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shows that larger desiccant flow rates permit more dehumidification as well because of the resulting higher solution concentration throughout the contactor, and therefore higher capacity of absorbing moisture.

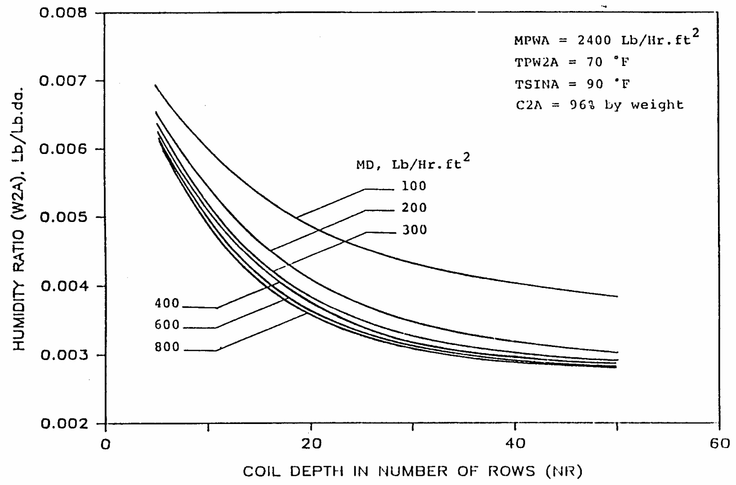

Figure 2: Humidity ratio of air leaving the absorber, versus packing depth, for different desiccant flow rates

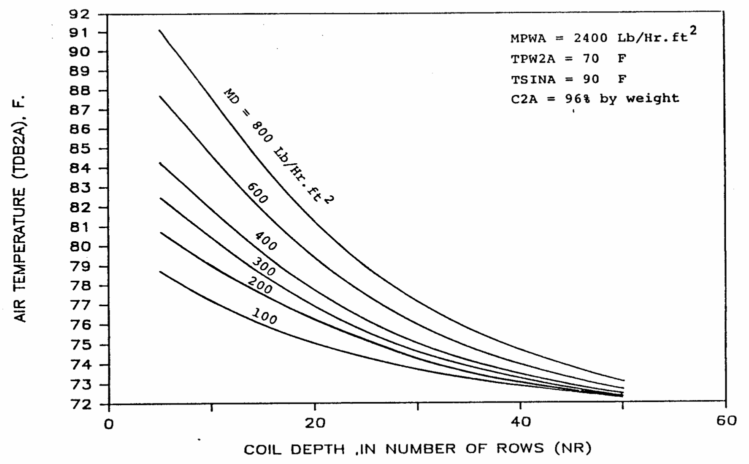

Figure 3: Temperature of air leaving the absorber versus packing depth, for different desiccant flow rates.

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Figure 3 illustrates the decrease in the leaving air dry bulb temperature (TDB2A) with larger packing depth since more heat is then rejected to the cooling process water. It also shows that higher desiccant flow rates result in higher leaving air temperatures. This can be attributed to the fact that, with more desiccant introduced at 90 ºF (32 ºC), more sensible heat is added to the process in addition to the larger latent heat rejected by the resulting higher dehumidification

The combination of the temperature and humidity results gives the leaving air enthalpy (H2A) as illustrated by Figure 4. The enthalpy curves confirm the complex interaction between the heat and mass transfer mechanisms inherent in the process. As expected, they show that for a given desiccant flow rate, the leaving air enthalpy decreases with larger packing number of rows because of the larger contact area for both heat and mass transfer.

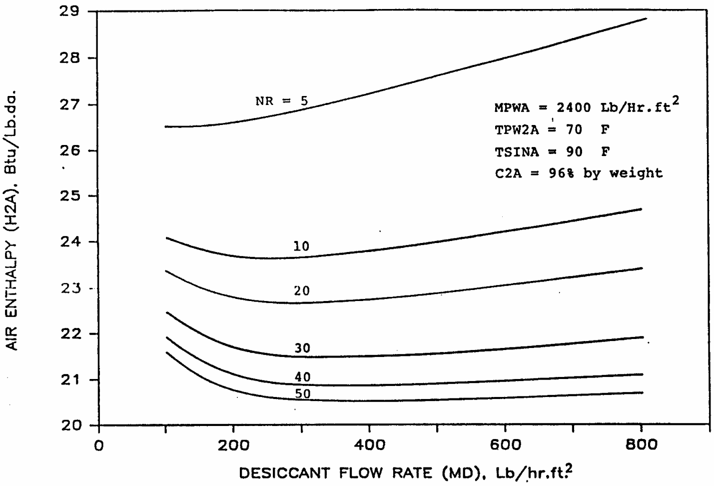

Figure 4: Enthalpy of air leaving the absorber, versus desiccant flow rates, for different packing depths

However, for a specified packing depth (NR), the enthalpy (H2) first decreases to a minimum then starts increasing with larger desiccant flow rates. This behavior is explained by the fact that the equilibrium water partial pressure is function of both concentration and temperature, and increasing the desiccant flow rate beyond a certain limit does not cause further noticeable dehumidification (Figure 2), but introduces enough heat to make the process warmer as illustrated by the higher air temperatures (Figure 3). The increase of air temperature offsets the small gain in dehumidification and results in the higher air enthalpies.

Figure 5 pictures the variation of the outlet process water temperature (TPWlA). It confirms that larger amounts of heat are rejected to the cooling water with larger desiccant flow and/or deeper packing coils.

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Figure 6 illustrates by the dependence of the concentration (C_{IA}) of the weak solution leaving the absorber, on both packing depth and desiccant flow rate. As the desiccant flow rate increases, more solution is available to absorb the moisture from the air stream. As a result, the solution leaves the contactor at a higher concentration, in spite of the larger dehumidification due to the consequent higher mass transfer driving potential. It is also obvious that, with deeper packing, more contact between the falling desiccant film and the countercurrent air stream is available for mass transfer, and therefore results in lower weak solution concentrations leaving the absorber.

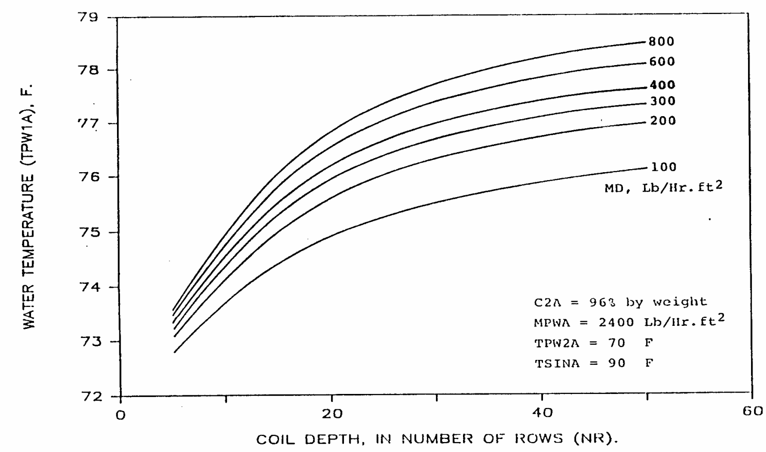

Figure 5: Outlet temperature of cooling water, versus packing depth, for different desiccant flow rates.

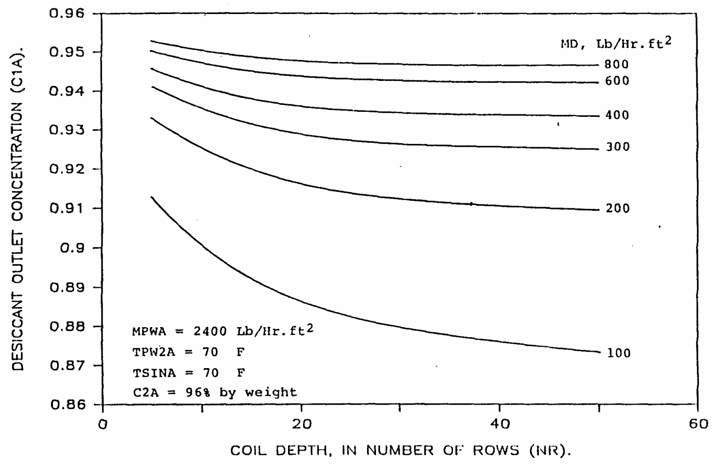

Figure 6: Concentration by weight of the desiccant solution leaving the absorber versus packing depth, for different desiccant flow rates

3.2 Effect of Cooling Water Flow Rate and Inlet Temperature

Figures 7 and 8 represent the air conditions leaving the absorber versus the coil depth and for different cooling water flow rates and inlet temperatures. The cross examination of the curves shows that larger cooling water flow rates result in dryer and cooler air leaving the absorber. More cooling water means larger heat capacity, which permits to maintain the desiccant cooler throughout the packing, and thus enhances the mass transfer by lowering the equilibrium water partial pressure in the desiccant solution.

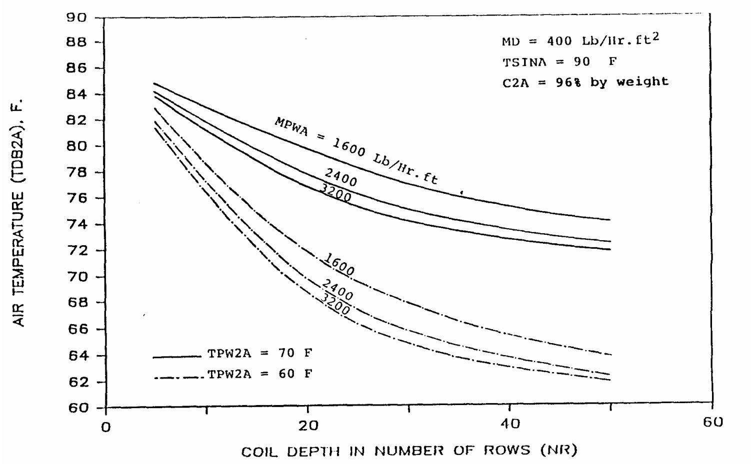

Figure 7: Temperature of air leaving the absorber versus packing depth , for different cooling water flow rates and inlet

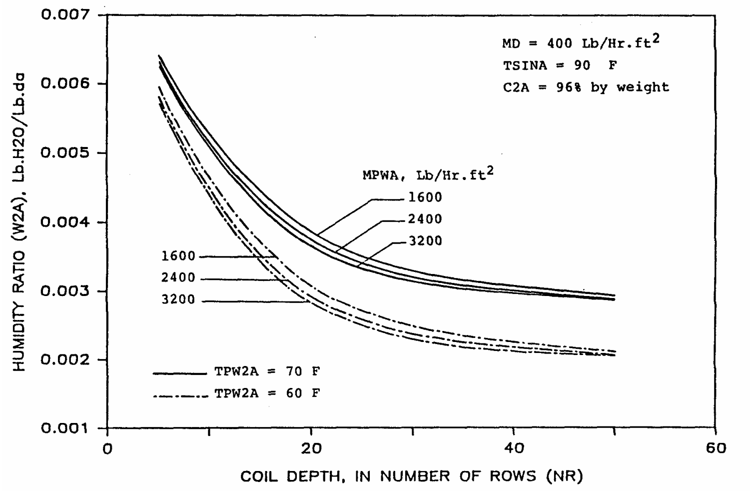

Figure 8: Humidity ratio of air leaving the absorber, versus packing depth, for different cooling water flow rates and inlet temperatures

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In addition, the resulting larger air-to-water temperature differential and higher liquid side heat transfer-coefficient lead to a better cooling of the air stream. At large flow rates and large coil depths, the curves approach certain asymptotic limits. Those limits correspond to the inlet process water temperature (TPW2A) and to the vapor pressure of water in the desiccant solution at the inlet temperature and concentration.

The comparison of the curves corresponding to different water inlet temperatures, shows that cooler cooling water results in the same effects as those of larger water flow rates. That is as expected since both cases lead to lower operating temperatures.

The results, discussed above, emphasize the advantage of non-adiabatic contactors, namely: more efficient mass transfer. The equilibrium water partial pressure is lowered, by cooling the desiccant instead of using higher concentrations, which would require higher regenerating temperatures.

3.3 Effect of the Solution Inlet Concentration and Temperature

The effects of the solution inlet conditions on the absorber performance are illustrated in Figures 9 through 11. For a given inlet concentration, warmer desiccant introduces more sensible heat to the process. As a result, the air leaves the absorber at a higher temperature, as shown in Figure 9; and since the desiccant is then slightly warmer throughout the contactor, less dehumidification takes place, as shown in Figure 10, due to the resulting higher equilibrium water partial pressure in the aqueous solution. This confirms the importance of pre-cooling the desiccant, which can easily be done by a solution-to-solution intercooler.

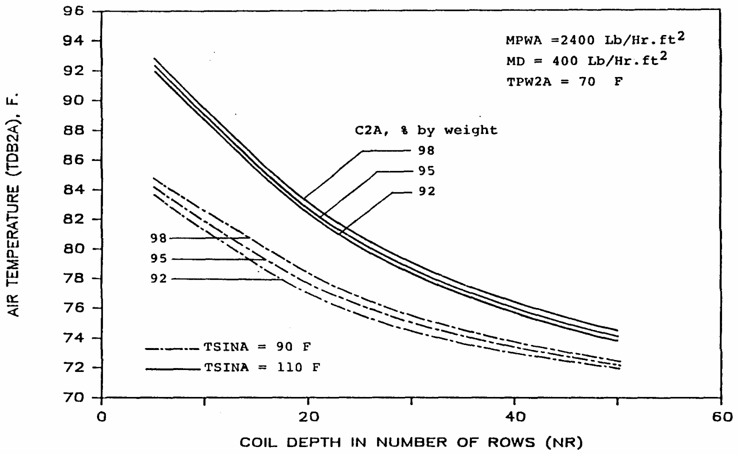

Figure 9: Temperature of air leaving the absorber, versus packing depth, for different desiccant solution inlet concentrations and temperatures.

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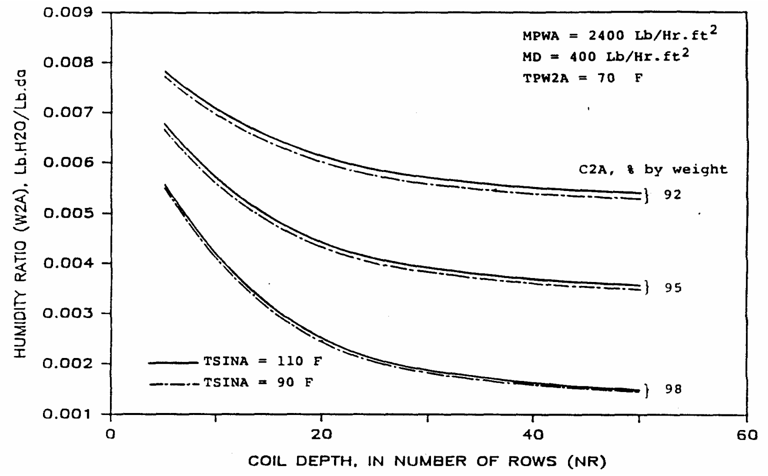

Figure 10: Humidity ratio of air leaving the absorber, versus packing depth, for different desiccant solution inlet concentrations and temperatures

As the inlet concentration is decreased, the desiccant solution becomes less capable of absorbing moisture (Figure 11). A minimum concentration is however required if any dehumidification is to occur. At that minimum, which is 84% by weight for the set of conditions of Figure 10, the equilibrium water partial pressure in the desiccant solution equals that of inlet moist air and the mass transfer driving potential becomes zero. If the inlet concentration was further reduced, the process would be reversed and water would be removed from the dilute desiccant solution.

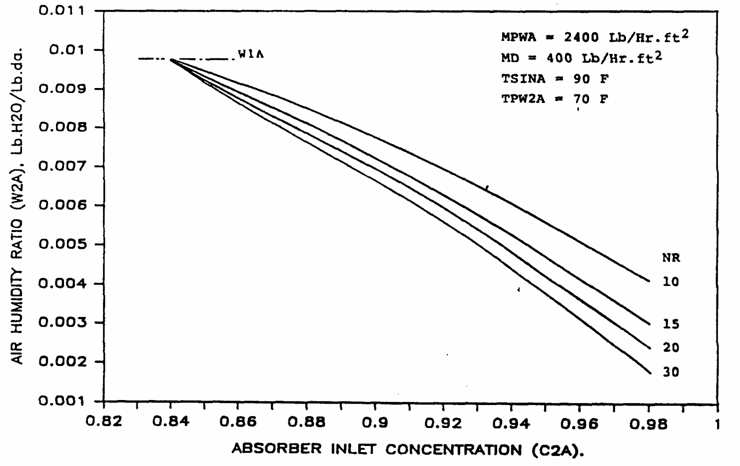

Figure 11: Humidity ratio of air leaving the absorber, versus desiccant solution inlet concentrations, for different packing depths

4. CONCLUSIONS

Based on the findings of this work the following conclusions may be drawn:

a- Non-adiabatic dehumidification permits to maintain the desiccant solution cooler throughout the process for a more efficient mass transfer operation and results in lower desiccant solution concentration requirements. Obvious consequences are the lower regenerating temperatures and the subsequent higher efficiency of the solar collectors if used.

b- The finned tube dehumidification chamber has a good performance, is compact in size, is flexible for multipurpose solar applications and suitable for other energy sources for back up.

c- In addition to transferring moisture, from the dehumidifier to the regenerator, desiccant solutions introduce undesired heat to the dehumidifier of a continuous process and should therefore be pre-cooled for a better performance.

d- The theoretical model used in this work and already experimentally checked, represents a simple tool to analyze the overall performance of the complex simultaneous heat and mass transfer that takes place in liquid desiccant dehumidifiers.

LIST OF SYMBOLS

A	*Area,* ft^2 *(*m^2*)*
C_p	*Specific heat, Btu/Lb.ºF (Kj/Kg.ºC)*
C	*Concentration of the desiccant solution, dimensionless*
E	*Surface effectiveness, dimensionless*
F	*Wetting factor, dimensionless*
G	*Gas mass flow rate per unit cross- section area, Lb/hr.*ft^2 *(Kg/sec.*m^2*)*
h	*Convective heat transfer coefficient, Btu/sec.*ft^2*.ºF(Kw/*m^2*.ºC)*
H	*Specific enthalpy, Btu/Lb (Kj/Kg)*
H^*	*Modified enthalpy expression, Btu/Lb (Kj/Kg)*
K_y	*Overall mass transfer coefficient,* ft^2*/hr (*m^2*/sec)*
L_s	*Liquid mass flow rate per unit cross- section area, Lb/hr.*ft^2 *(Kg/sec.*m^2*)*
Le	*Lewis number, dimensionless*
M	*Mass flow rate per unit cross- section area, Lb/hr.*ft^2 *(Kg/sec.*m^2*)*
MA	*Air mass flow rate per unit cross section area, Lb/hr.*ft^2 *(Kg/sec.*m^2*)*
MD	*Desiccant mass flow rate per unit cross- section area, Lb/hr.*ft^2 *(Kg/sec.*m^2*)*

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MPW	*Process water mass flow rate per unit cross- section area, Lb/hr.ft^2 (Kg/sec.m^2)*
NR	*Number of rows*
Q	*Rate of heat exchanged, Btu/hr (Kw)*
T	*Temperature, ºF (ºC)*
TPW	*Process water temperature, ºF (ºC)*
TSIN	*Solution inlet temperature, ºF (ºC)*
Y'	*Humidity ratio, dimensionless*
X'	*Mass ratio in liquid phase, dimensionless*
ΔHm'	*Heat of mixing per unit mass of solvent, Btu/Lb (Kj/Kg)*
λ'	*Specific latent heat of evaporation, Btu/Lb (Kj/Kg)*

Subscripts:

A	*Absorber*
D	*Desiccant*
G	*Gas*
i	*interface*
in	*Inlet*
Lm	*Logarithmic mean value*
L	*Liquid*
m	*Mass transfer*
o	*Outside*
S	*Solvent*
w	*water*
1	*Bottom of the contactor*
2	*Top of the contactor*

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