

**ANALYSIS OF LIQUID-DESICCANT SYSTEMS
AND COMPONENT MODELING**

by

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ABSTRACT

A disadvantage of vapor-compression air conditioning is that the air must be cooled below its dewpoint in order to provide dehumidification. Hybrid liquid-desiccant systems can reduce overall energy use and/or cost by separating an air conditioning load into its sensible and latent components. Currently, liquid-desiccant air-conditioning equipment is not widely used. Computer simulations of different system configurations are required in order to study the performance of liquid-desiccant systems relative to traditional systems.

A computationally simple effectiveness model has been developed for packed-bed liquid-desiccant heat and mass exchangers. It compares well with a more detailed finite-difference model as well as with experimental data.

The effectiveness model is used in the evaluation of two systems using the modular simulation program TRNSYS. The results are compared with a traditional vapor-compression air conditioning system. Evaporative cooling can enhance liquid-desiccant system performance because of the desiccant system's production of very dry air.

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NOMENCLATURE

A_v	transfer area/unit volume [1/m]
c_p	specific heat [kJ/kg-°C]
C_{sat}	saturation specific heat [kJ/kg-°C]
COP	coefficient of performance [dimensionless]
COP_c	cooling coefficient of performance [dimensionless]
COP_h	heating coefficient of performance [dimensionless]
h	enthalpy [kJ/kg]
h_D	mass transfer coefficient [kg/s-m ² -Δω]
h_C	heat transfer coefficient [kW/m ² -°C]
Le	Lewis number [dimensionless]
\dot{m}	mass flow rate [kg/s]
m^*	capacitance ratio [dimensionless]
NTU	number of transfer units [dimensionless]
q	heat flux per unit transfer area [kW/m ²]
T	temperature [°C]
V_T	packing material volume [m ³]
Z_{max}	height of packing material [m]
η	efficiency
ω	humidity ratio [kg/kg dry air]
ξ	solution concentration [weight fraction]

Nomenclature, cont'd

Subscripts

a	air
des	desired value
eff	effective value
i	inlet
m	moist air (used as subscript to specific heat)
o	outlet
s	solution
$T_{s,sat}$	air saturated at solution temperature
T_s	at solution temperature
v	water vapor
0	at 0 °C.
1	inlet
2	outlet

Chapter 1

INTRODUCTION

In the United States, energy used to cool and dehumidify the air in buildings accounts for about 3% of all energy used nationwide or 8% of the energy used in the residential and commercial sector [1]. Traditional vapor-compression air-conditioning systems are completely powered by electricity, a relatively expensive energy source that is often accompanied by peak load charges as well as its base cost.

The use of liquid-desiccant air conditioning has been proposed as an alternative to the vapor-compression systems. Liquid-desiccant systems can reduce the overall use of energy, as well as shift energy use away from electricity and toward cheaper fuels.

1.1 Background

Currently, the use of liquid-desiccant air-conditioning systems is limited to specialized applications where precise humidity control is required. Whether they are practical for general air-conditioning use remains to be determined by a combination of experimental and modelling studies.

1.1.1 Use of Liquid Desiccants

One of the main disadvantages of vapor-compression air conditioning is that providing dehumidification requires cooling the air below its dewpoint. In most cases,

this results in an air temperature that is colder than the temperature necessary to meet the sensible load. Reheating the air is then necessary, resulting in a thermodynamically inefficient process. Figure 1.1 shows the process schematically on a psychrometric chart. The air is cooled to the dewpoint at the desired humidity level, and then reheated to the temperature necessary to meet the sensible load. A hybrid liquid-desiccant air-conditioning system separates the load into its sensible and latent components, and then meets each individually. This is shown in Figure 1.2. The desiccant conditioner dehumidifies the air, which is then cooled in a conventional chiller. Sensible cooling can also be provided by other means, such as evaporative coolers.

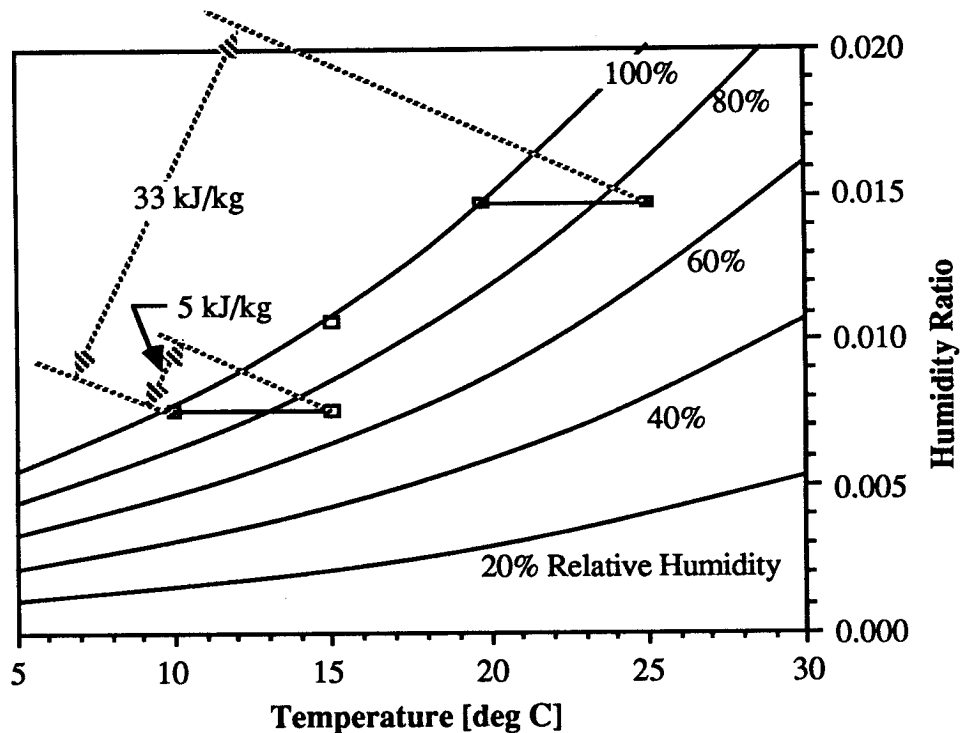


Fig. 1.1 Vapor-Compression Air Conditioning

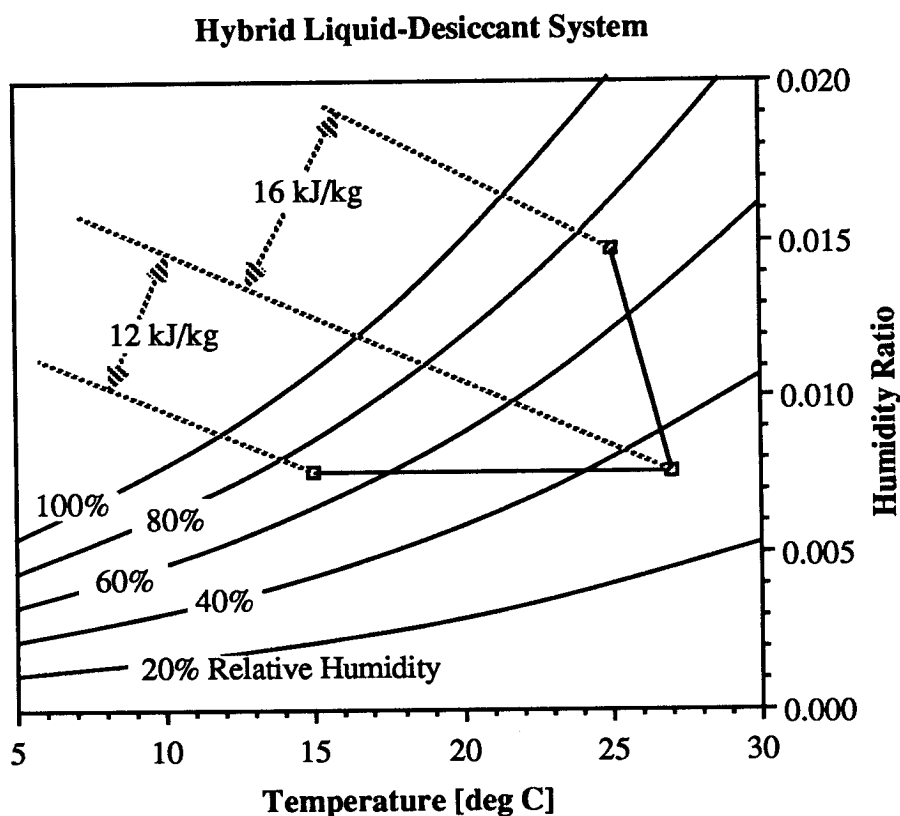


Fig. 1.2 Hybrid Liquid-Desiccant Air Conditioning

Inside the liquid-desiccant heat and mass exchanger, the desiccant solution flows over packing material and comes into contact with an air stream. The mass transfer is driven by the difference between the partial pressure of water vapor in the air and the vapor pressure associated with the solution. The vapor pressure above the solution decreases with increasing solution concentration and increases with increasing solution temperature. Therefore, cold, highly concentrated solution provides the best dehumidification while warm, dilute solution provides the best humidification. The air state will always change toward being in equilibrium with the solution, as explained by equilibrium curves (Figure 1.3). These curves show the state of air in equilibrium with

different solution conditions. For example, if LiCl solution has a concentration of 40% and a temperature of 30 °C, the air in equilibrium with it is also at 30° C and has a humidity ratio of 0.0053.

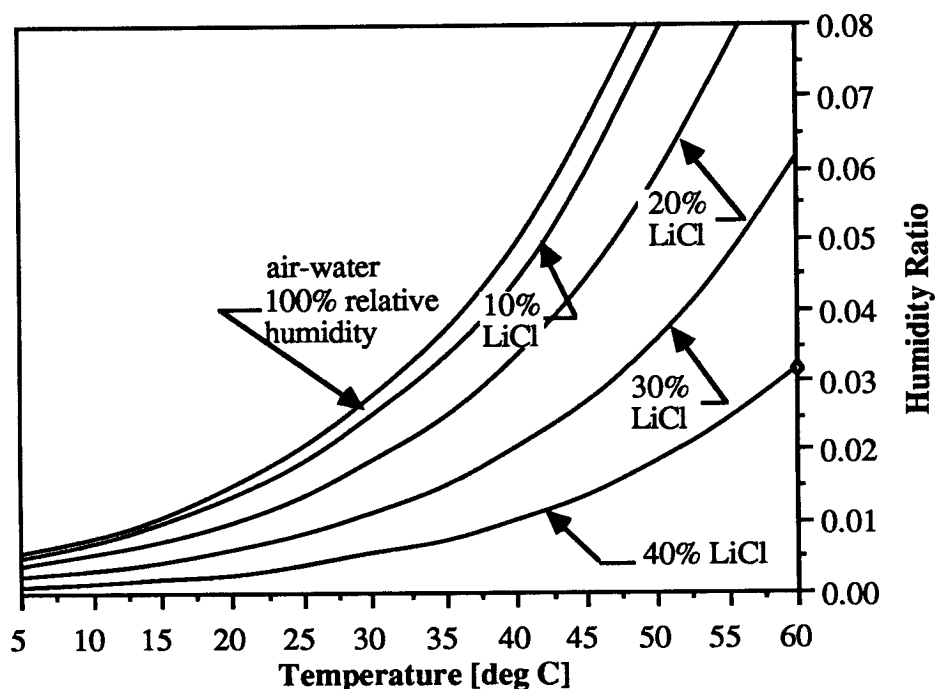


Fig. 1.3 Lithium Chloride Equilibrium Curves

A system that makes use of liquid desiccants consists of several major components. The liquid-desiccant conditioner and regenerator (Figure 1.4) transfer heat and mass between the air and the desiccant solution. The conditioner dehumidifies the air and may, at the same time, increase or decrease the air's temperature. The solution leaves the conditioner slightly diluted from its initial state, and must be reconcentrated in the regenerator. The regenerator is physically similar to the conditioner, but humidifies an exhaust air stream or ambient air to reconcentrate the solution. Figure 1.5 shows a basic configuration for a system that uses only liquid

desiccants for conditioning, while Figure 1.6 shows one of many possible configurations for a hybrid system. In addition to providing a more thermodynamically-efficient method of air conditioning, liquid-desiccant systems can also provide very precise humidity control, very low humidity ratios (useful for applications such as pharmaceutical manufacturing) and, depending on the desiccant solution used, other benefits, such as air purification [2].

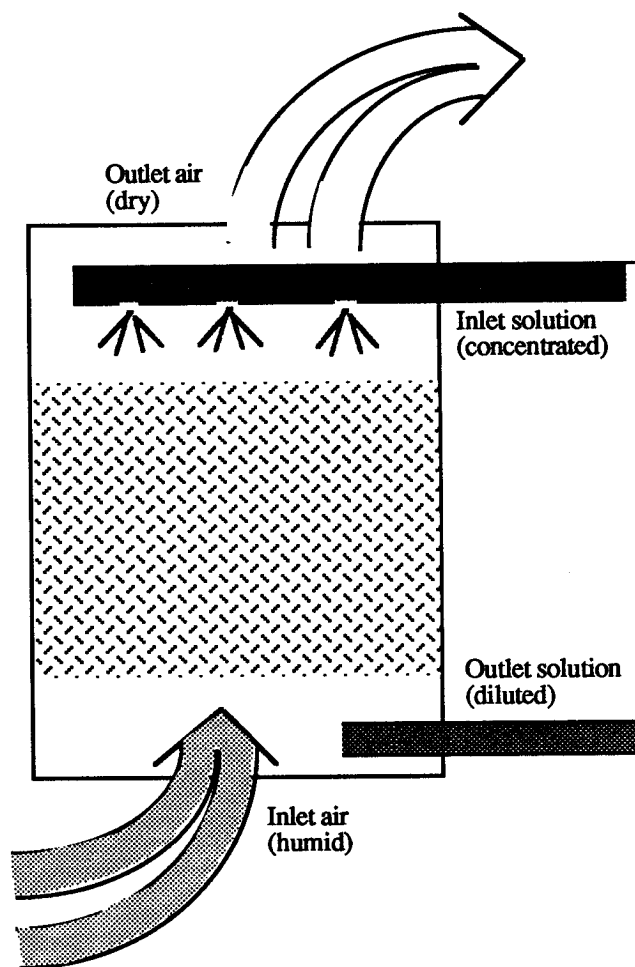


Fig. 1.4 Liquid-Desiccant Conditioner

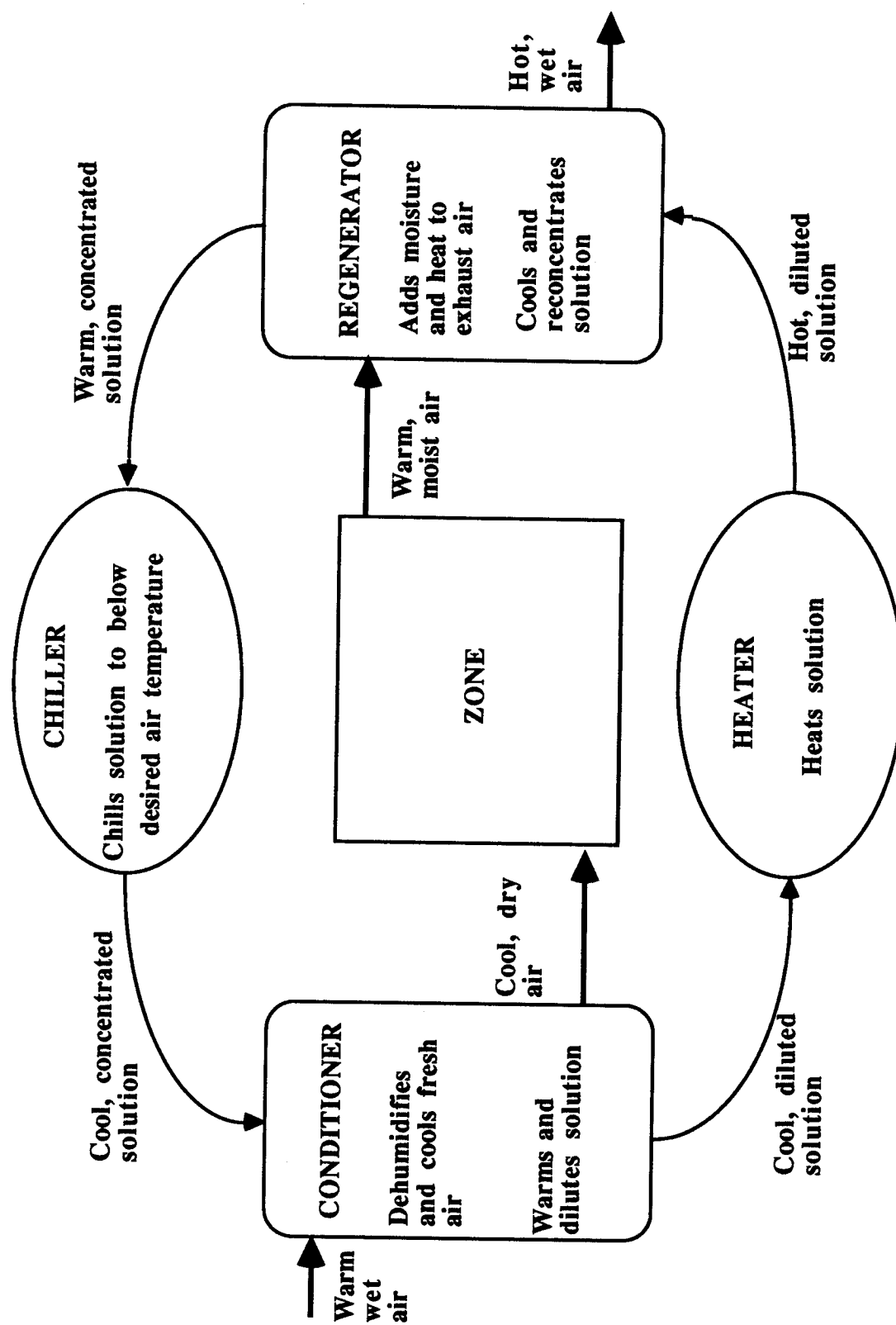


Fig. 1.5 Basic System Configuration -- Liquid-Desiccant Cooling and Dehumidification

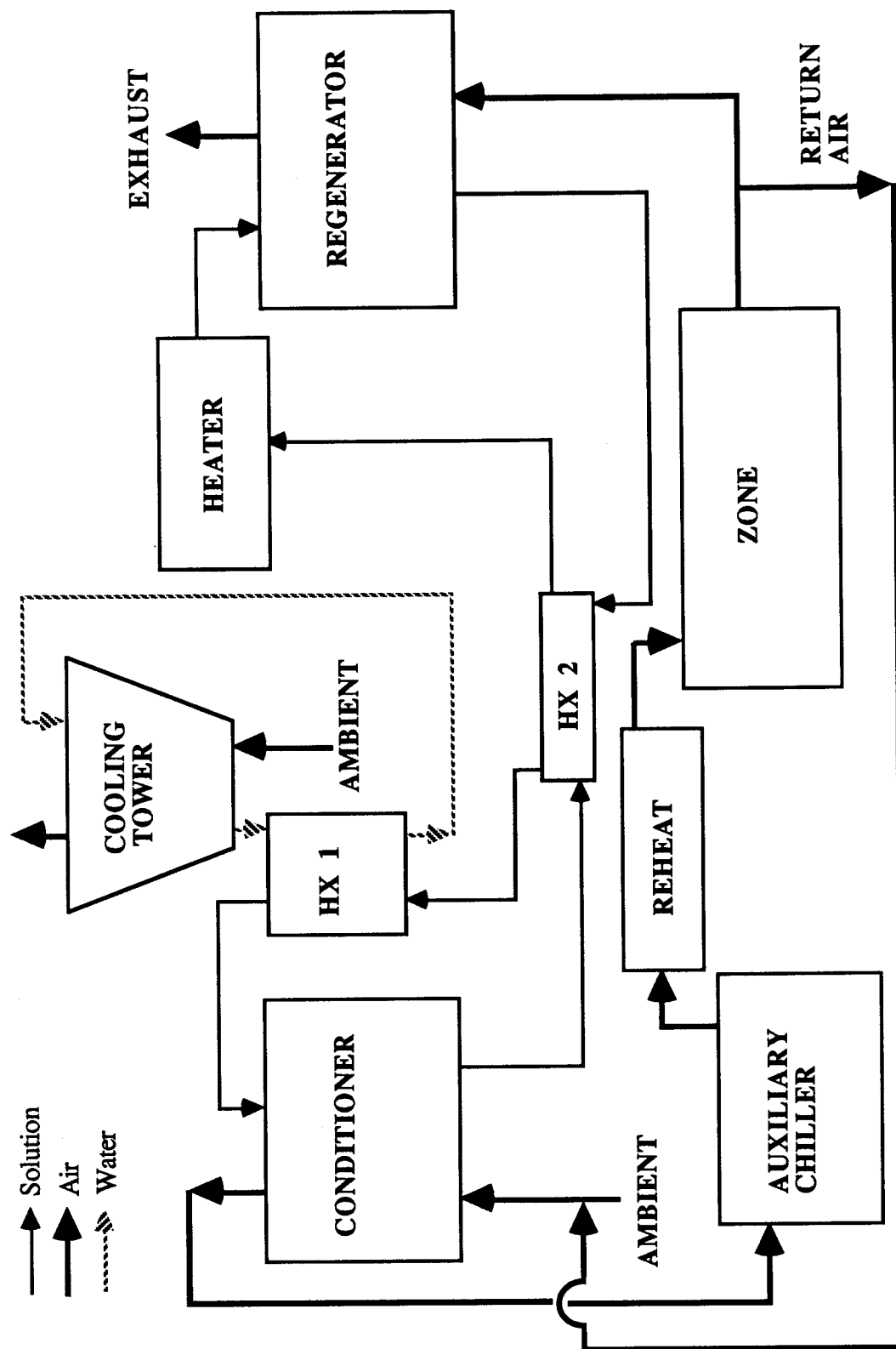


Fig. 1.6 Schematic of Hybrid Liquid-Desiccant Air Conditioning System

1.1.2 Previous Studies at the University of Wisconsin

Previous liquid-desiccant studies at the University of Wisconsin have focused on several items. Buschulte [3] compiled property relations for air-water mixtures and LiBr-water and LiCl-water solutions. He also developed finite-difference and effectiveness models of the heat and mass transfer in a liquid-desiccant component. He began a simulation of the liquid-desiccant system at the Science Museum of Virginia (SMVA). Sick [4] continued to look at a system configured similarly to the SMVA system. He modelled the entire system in several different operation modes, and drew conclusions based on yearly simulations. A main conclusion was that using a chiller to cool the conditioner solution is not economically advantageous. Between Buschulte and Sick, several types of liquid-desiccant models were developed and used.

1.1.3 Current Liquid-Desiccant Models

Finite-difference models have been developed by Factor and Grossman [5] and Buschulte [3]. Buschulte used his as a base with which to compare simpler models. However, finite-difference models are computationally slow. In yearly simulations of liquid-desiccant systems, the individual component models might each be referenced thousands of times. In order to study many configurations at different geographic locations it is preferable to have a model that is more computationally efficient. Chapter 2 describes a detailed finite-difference model used to help validate the simpler model found in Chapter 3.

The K-Factor model [4] was suggested by Kathabar Engineering Service (a manufacturer of liquid-desiccant equipment). It is based on two relationships: first, the air outlet humidity ratio is equal to the humidity ratio in equilibrium with the solution at

its inlet concentration and the air inlet temperature, and second, the temperature difference between the outlet air and the inlet solution is proportional to the enthalpy change of the air stream. The constant of proportionality (the K-Factor) depends on the size of the component and mass flow rates and is determined experimentally. This model was shown to be useful [6] for simulating a specific system at specific conditions. It also has the advantage of computational efficiency. However, the relationship between the K-Factor and any tangible parameter is unknown. For example, if a modelling study is desired that would show the effect of changing mass flow rates on system performance, it is not known how the K-Factor should be changed. This model then, is good for specific system simulations at constant conditions, but not very useful for general studies.

Several studies have been performed which resulted in empirical liquid-desiccant models. For example, CSU [7] correlated a relationship for water evaporation rate in the regenerator as a function of air temperature and humidity ratio, and solution concentration and mass flow rate. These models are limited to the specific equipment for which they were developed.

Buschulte's effectiveness model used heat and mass transfer equilibrium-effectiveness relationships to determine the states of the outlets from the desiccant chamber. This model was shown to give poor results at operating conditions away from its design point [4].

1.1.4 Current Liquid-Desiccant Systems

Desiccant air-conditioning systems have been proven to be practical for certain specialized applications such as grocery stores and some manufacturing processes. Of

these, most of the systems in use are solid-desiccant systems. The Science Museum of Virginia has a liquid-desiccant system that has been analyzed and found not to offer decreased operating expense because of the method of cooling the solution.

1.2 Objective

The goal of this study is twofold: to develop a liquid-desiccant model suitable for use in system simulations, and to use the model to show the relative advantages of different system configurations.

1.2.1 Model Development

The existing models for liquid-desiccant components have not been found to be adequate for general system comparisons and yearly simulations of different desiccant system configurations. A new model should have the characteristics of speed and accuracy. It should also be based upon tangible parameters, such as component size and mass flow rates. Braun [8] has developed a cooling tower model with these characteristics that can be modified to model liquid-desiccant conditioner and regenerator performance.

1.2.2 System Analysis

Because the optimal system configuration of a liquid-desiccant air conditioning system remains to be found, it would be useful to show some improvements over the most basic system. This study will show several different variations and their effects on overall system performance.

Chapter 2

FINITE-DIFFERENCE MODEL

A finite-difference solution to the governing equations of liquid-desiccant components is the most fundamental model available. In this study, a finite-difference model is used to verify a simpler model described in Chapter 3. In this chapter, the finite-difference model is derived. Since there is an assumption of a unity value of Lewis number in the simpler model, the effects of changing the Lewis number are examined, as well as the effects of changing system size and operating conditions.

2.1 Derivation

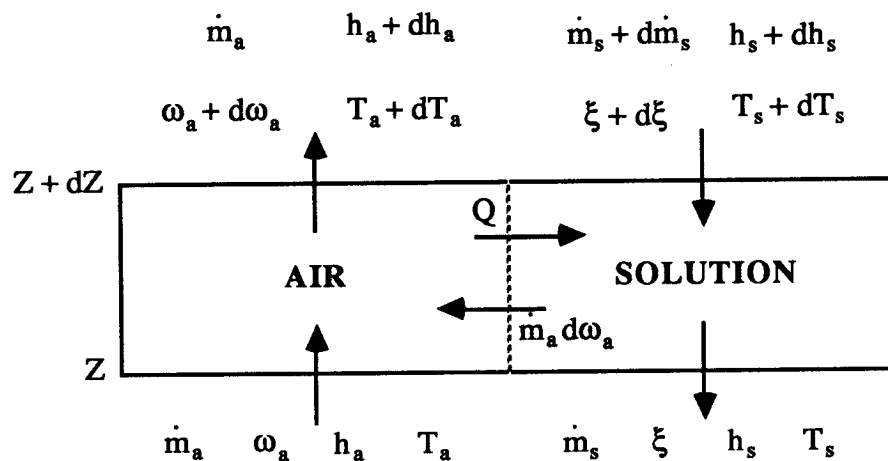


Fig. 2.1 Differential Heat and Mass Transfer Element Used to Derive Finite-Difference Equations

Figure 2.1 shows a schematic of a differential slice of a liquid-desiccant heat and mass exchanger as used in the derivation. The chamber is filled with packing material having a surface area to volume ratio of AV , a total volume of V_T , a total height of Z_{\max} , and a mass transfer coefficient of h_D .

A mass balance on the water vapor gives:

$$\frac{d\dot{m}_s}{dZ} = \dot{m}_a \frac{d\omega_a}{dZ} \quad (2.1)$$

The mass transfer is driven by the difference in partial vapor pressures between the air and the solution. The vapor pressure above the solution is determined by the solution's temperature and concentration.

$$\frac{d\omega_a}{dZ} = \frac{h_D}{S} (\omega_{T_s, \text{sat}} - \omega_a) \quad (2.2)$$

where

$$S = \frac{\dot{m}_a Z_{\max}}{V_T A_V} \quad (2.3)$$

The heat flux per unit transfer area, q , is defined by the following equation:

$$q = h_C (T_a - T_s) \quad (2.4)$$

The enthalpy of the water vapor is calculated from air enthalpies:

$$h_v = \frac{h_a - h_{a, \omega=0}}{\omega_a} \quad (2.5)$$

Energy balances on the air and solution give the following equations when simplified:

$$\frac{dh_a}{dZ} = \frac{d\omega_a}{dZ} h_v - \frac{q}{S} \quad (2.6)$$

$$\frac{dh_s}{dZ} = \frac{\dot{m}_a}{\dot{m}_s + d\dot{m}_s} \left(h_v \frac{d\omega_a}{dZ} - h_s \frac{d\omega_a}{dZ} - \frac{q}{S} \right) \quad (2.7)$$

In the following analysis, two other terms, Le and NTU are used. They are defined as follows:

$$Le = \frac{h_C}{h_D c_{p,m}} \quad (2.8)$$

$$NTU = \frac{h_D A_V V_T}{\dot{m}_a} \quad (2.9)$$

Since the differential equations cannot be solved analytically, the most basic solution is a numerical integration along the height of the transfer chamber. One end of the chamber must be chosen as a starting point. This requires the outlet conditions for one of the streams to be guessed at the start of the integration. Either stream can be chosen, but arbitrarily assume that integration begins at the air inlet.

The steps in solving the finite-difference model are:

- 1) Calculate the constant mass flow rate of salt. (2.10)
- 2) Calculate the value of S. (eq. 2.3)
- 3) Guess the outlet temperature of the solution.
- 4) Guess the outlet concentration of the solution (inlet concentration is close).
- 5) Repeat the following for each node:
 - 5a) Calculate enthalpy of air and enthalpy of solution.

- 5b) Calculate enthalpy of water vapor (eq 2.5).
- 5c) Calculate the mass transfer (eq. 2.2).
- 5d) Calculate the heat transfer (eq. 2.4).
- 5e) Calculate the change in solution mass flow rate (eq. 2.1).
- 5f) Calculate the change in solution enthalpy (eq. 2.7).
- 5g) Calculate the change in air enthalpy (eq. 2.6).
- 5h) Calculate the change in solution concentration (eq. 2.8).
- 5i) Use known property relations to calculate air and solution temperatures.
- 6) Iterate until the calculated inlet solution conditions match the known conditions.

The iteration process in the finite-difference model is critical. Several different methods were attempted. The MINPACK program LMDIF1 was used successfully, but required many iterations and therefore used excessive CPU time. To reduce the number of iterations, a different convergence technique was used. In order to guess a new value for the solution outlet conditions, the difference between the calculated inlet and the actual inlet conditions was used as follows (using temperature as an example):

$$T_{\text{out,guess}}^{i+1} = T_{\text{out,guess}}^i + (T_{\text{in}} - T_{\text{in,calc}}^i) \quad (2.11)$$

The use of this equation resulted in convergence trouble. Some cases required many iterations, and some never converged at all. To solve this problem, a relaxation factor, R , was introduced as follows:

$$T_{\text{out,guess}}^{i+1} = T_{\text{out,guess}}^i + R(T_{\text{in}} - T_{\text{in,calc}}^i) \quad (2.12)$$

A value of 0.5 for R worked well to solve the cases of interest in considerably fewer iterations than the MINPACK program.

2.2 Effect of Lewis Number

The effect of the Lewis number on the outlet states of the the liquid-desiccant chamber can be observed easily with the finite-difference model. Sixteen different cases were examined with several parameters varied between low and high values. Table 2.1 shows the cases tested. All 16 cases had a air mass flow rate of 1 kg/s, a solution mass flow rate of 2 kg/s, A_V , V_T , and Z_{\max} all equal to 1.0. Each case was evaluated at 10 values of h_D (varied between 0.01 and 10.0), 3 values of Le (0.8, 1.0, and 1.2) and used 25 nodes.

TABLE 2.1

Case Numbers for Finite-Difference Model Evaluations

		$T_{a1} = 15\text{ C}$		$T_{a1} = 35\text{ C}$	
		$\omega_{a1} = 0.001$	$\omega_{a1} = 0.01$	$\omega_{a1} = 0.003$	$\omega_{a1} = 0.03$
$T_{s1} = 15\text{ C}$	$\xi_1 = 0.1$	1	5	9	13
	$\xi_1 = 0.4$	2	6	10	14
$T_{s1} = 35\text{ C}$	$\xi_1 = 0.1$	3	7	11	15
	$\xi_1 = 0.4$	4	8	12	16

Figures 2.2 thru 2.4 show finite-difference model results for Case 4. From these figures, it can be seen that the effect of the Lewis number is most important between NTUs of 0 and 6. The Lewis number has no effect on the equilibrium states reached at

higher values of NTU. The effect of the Lewis number on the air outlet humidity ratio is negligible in comparison to its effect on the air and solution outlet temperatures. In Case 4 at $NTU = 2$, the change in air temperature increased from 14.8 to 17.2 (16% difference) when Lewis number was increased from 0.8 to 1.2. The change in solution temperature increased from 5.4 to 5.9 C (8% difference). There was, therefore, an overall increase in heat transfer, but mass transfer remained constant. This is exactly what would be expected because of the way the finite-difference model is programmed. The mass transfer coefficient (h_D) is held constant, while the heat transfer coefficient is calculated as $h_C = h_D Le c_{pm}$ where c_{pm} is the specific heat of the moist air.

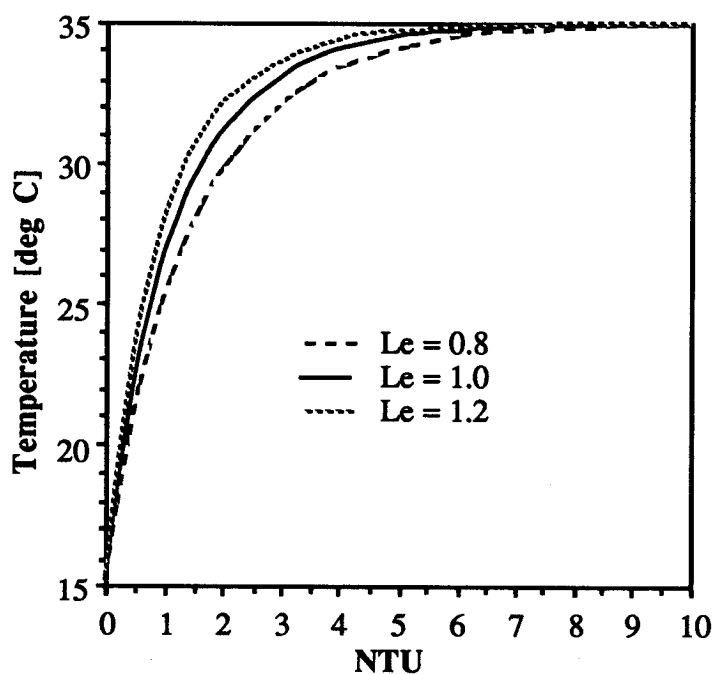


Fig. 2.2 Air Outlet Temperature vs. Component Size for Case 4

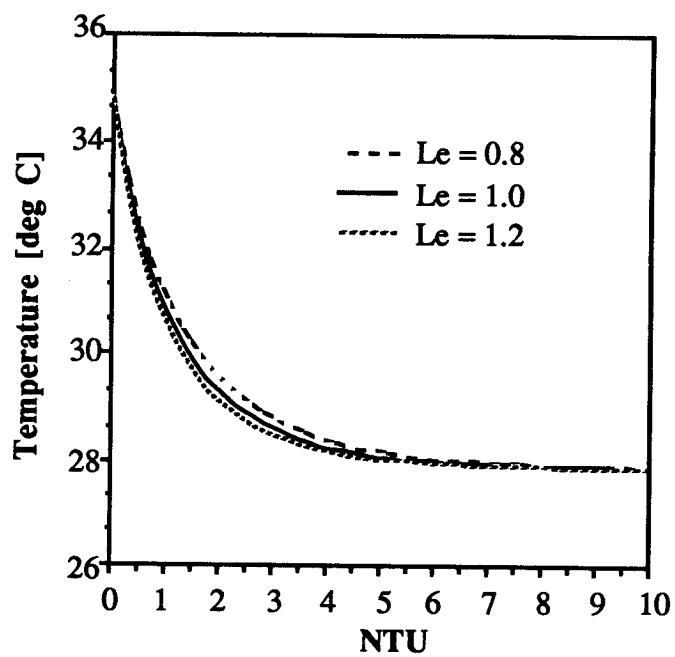


Fig. 2.3 Solution Outlet Temperature vs. Component Size for Case 4

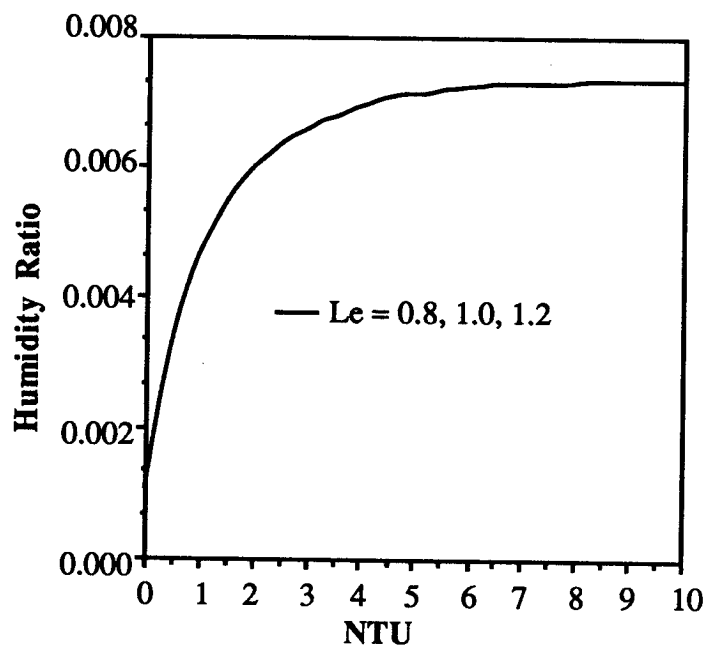


Fig. 2.4 Air Outlet Humidity Ratio vs. Component Size for Case 4

2.3 Finite-Difference Model Results

Appendix A shows the tables of data resulting from the Lewis number comparison. These tables show the outlet conditions at different values of NTU and Le for each of the 16 cases examined. In several of the cases, anomalies in the results showed up when the outlet conditions were plotted. The most common was in the air outlet temperature in all of the cases where the inlet air and inlet solution temperatures were equal (cases 1, 2, 5, 6, 11, 12, 15 and 16). Figure 2.5 shows an example of this behavior, from case 16.

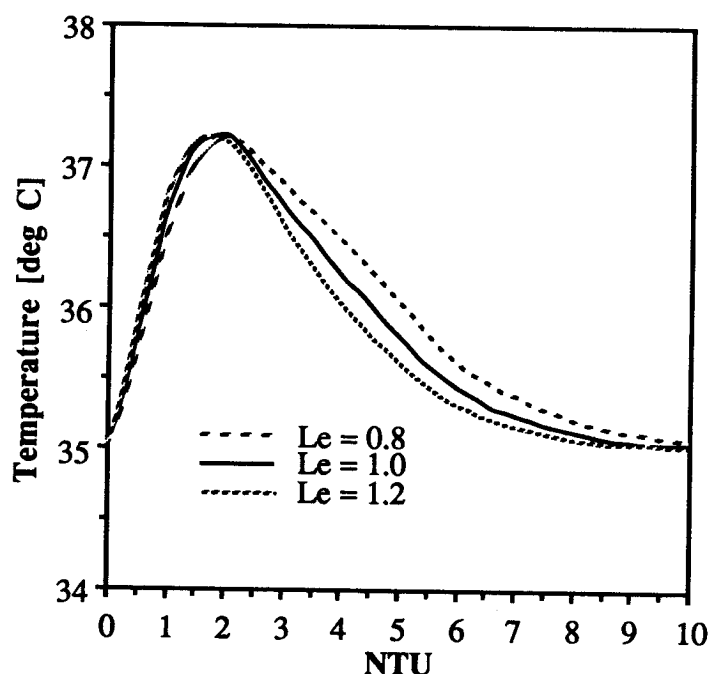


Fig. 2.5 Air Outlet Temperature vs. Component Size for Case 16

The temperature change of the air stream in these cases ranges from 0.015 C for Case 5 to more than 2 C for Case 16. The behavior can be explained by looking 'inside' the finite-difference model. Figures 2.6 and 2.7 show the solution temperature

and the air temperature and humidity ratio for each node inside the finite-difference model. The figures shown are taken from Case 16, $NTU = 6$, and $Le = 1$.

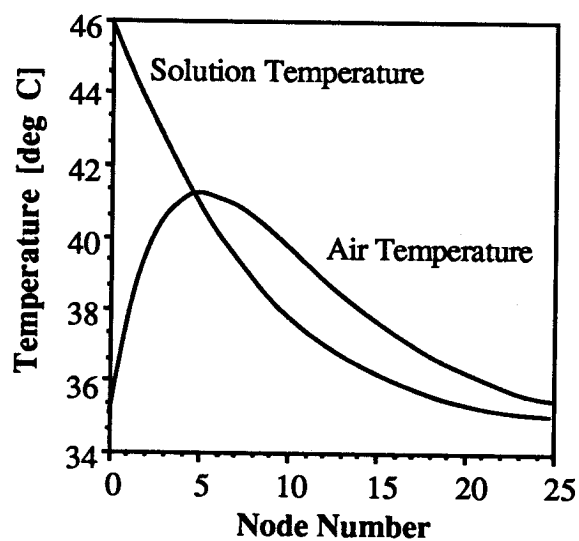


Fig. 2.6 Air and Solution Temperatures vs. Node Number for Case 16

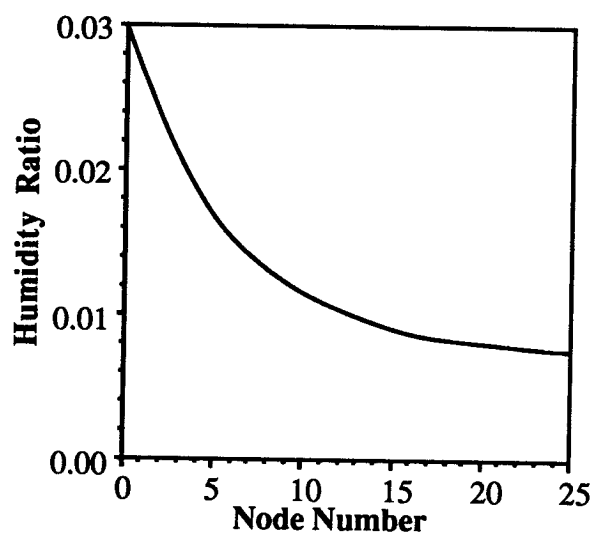


Fig. 2.7 Air Humidity Ratio vs. Node Number for Case 16

In Case 16, since the air is becoming *less* humid, either the air or the solution must be absorbing the heat of vaporization from the condensing vapor. Remembering that the solution inlet is at Node 25, we see that the solution temperature increases most steeply when the air humidity is decreasing most steeply. In this same range, the solution temperature is changing *away* from the air temperature. These observations indicate that the solution is absorbing the heat of vaporization from the condensing water vapor. The temperature change of the air seems to be predominately driven by the solution temperature, which the air sees at first as being higher than the air temperature, then being lower.

Another unexpected result occurred in Case 8. In this case, the air humidity ratio decreased with increasing NTU, and then increased slightly. This is shown in Figure 2.8.

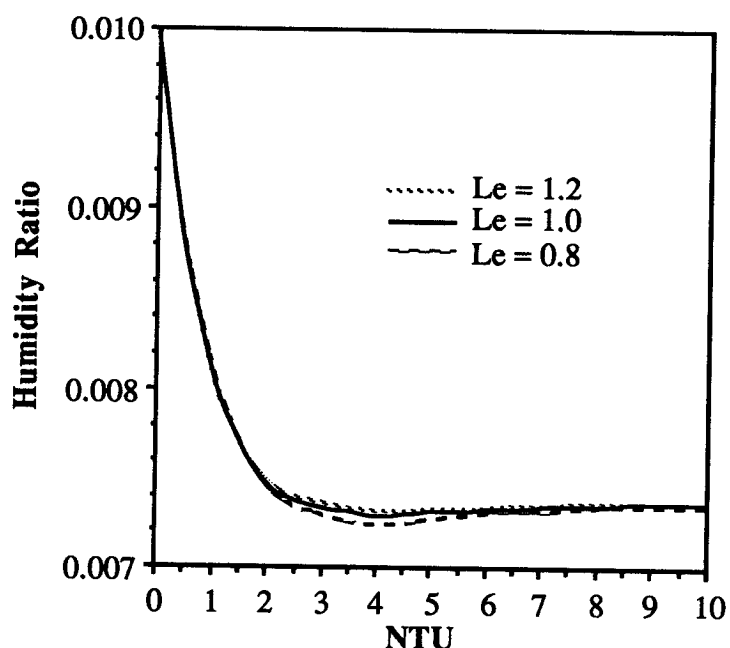


Fig. 2.8 Air Outlet Humidity Ratio vs. Component Size for Case 8

Since the dip in the humidity ratio plot is most extreme for the case where Lewis number is equal to 0.8, the explanation of the dip will use that case, at an NTU of 8 for an example. Figures 2.9 and 2.10 show the states of the air and the solution inside the finite difference model.

In Case 8, the heat of vaporization from the condensing water vapor seems to be completely converted to a temperature rise in the air stream, while the solution's temperature change is due to direct heat transfer with the air. When the solution temperature decreases, the associated air humidity ratio that would be in equilibrium with the solution also decreases. At approximately node 7, the air and solution-equilibrium humidity ratios cross. At this point, the mass transfer changes direction, from dehumidifying the air to humidifying it.

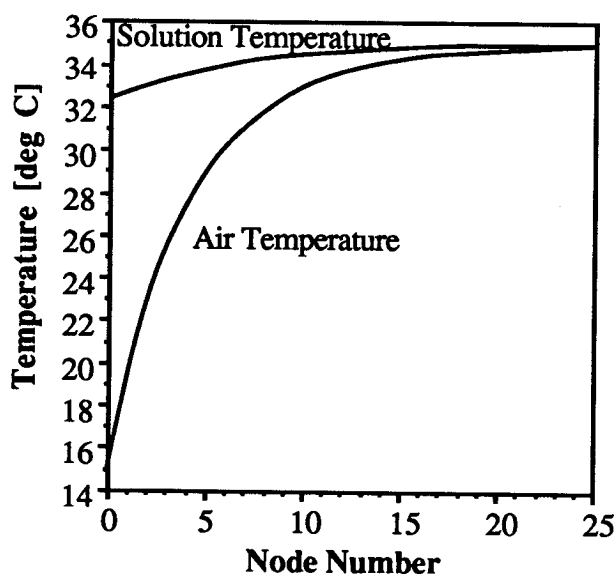


Fig. 2.9 Air and Solution Temperatures vs. Node Number for Case 8

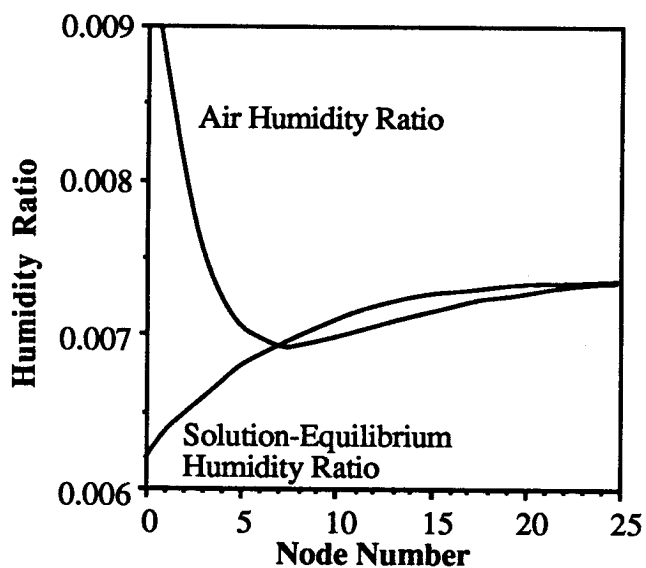


Fig. 2.10 Air and Solution Humidity Ratios vs. Node Numbers for Case 8

One other set of unexpected results showed up in the evaluation of the 16 cases. In case 9, the solution outlet temperature showed behavior that was more dependent on Lewis number than any of the other cases.

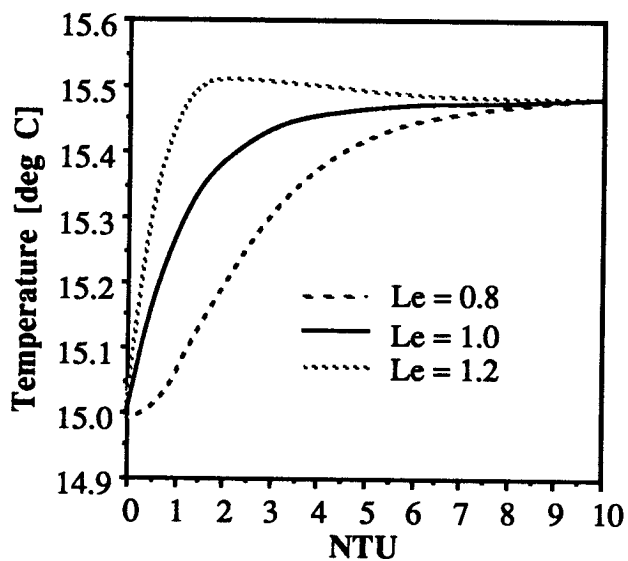


Fig. 2.11 Solution Outlet Temperature vs. Component Size for Case 9

This behavior has been examined more closely for an NTU of 6 at each of the three Lewis numbers. It was thought that the behavior might be the result of insufficient nodes in the finite-difference model. Case 9 was repeated for 200 nodes (instead of 25), but the results did not change.

The results at each of the three Lewis numbers were nearly identical. Figures 2.12 and 2.13 show the states of the air and solution at each node for $Le=1.2$.

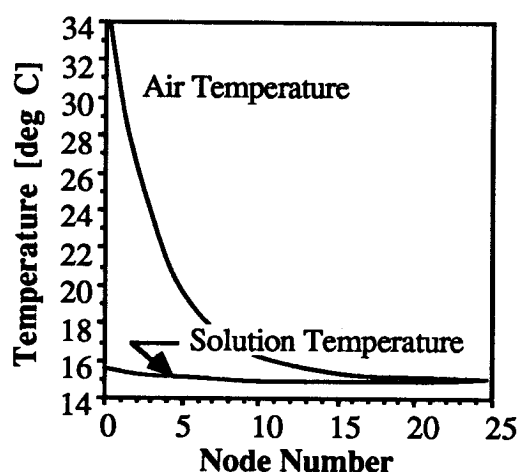


Fig. 2.12 Air and Solution Temperatures vs. Node Number for Case 9

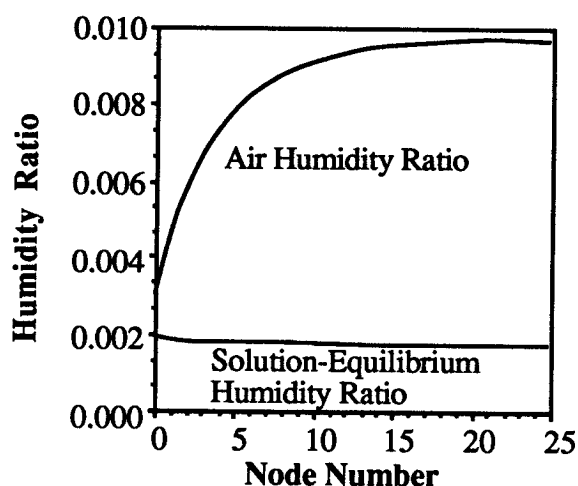


Fig. 2.13 Air and Solution Humidity Ratios vs. Node Numbers for Case 9

Case 9 is an example where the air is humidified. The solution concentration is very low, in the range where the solution acts almost like pure water. The inlet air humidity ratio is also very low, so it is humidified easily. The solution inlet temperature is low and the air inlet temperature is high. Under these conditions, the air is cooled because it gives up heat to vaporize the transferred mass of water. The solution temperature remains virtually constant throughout the process. Perhaps the explanation for the behavior in Figure 2.11 is that the solution is losing energy to vaporize the water at the same time that it gains a similar amount of energy from the warmer air and that the relative magnitudes of these two transfers change slightly with a changing Lewis number.

While the finite-difference model of a packed-bed liquid-desiccant heat and mass exchanger is useful for observing specific details about the chamber's performance, its use of computer time is prohibitive. Each analysis of outlet conditions from given inlet conditions involves many calculations. (See appendix F for FORTRAN code.) At each node, the air's specific heat, the water vapor enthalpy, the mass transfer, the heat transfer, the incremented air enthalpy, solution mass flow rate, solution enthalpy, and air humidity, the new solution concentration and temperature, and the new air temperature must all be calculated. For a small number of nodes (25), this results in at least 275 calculations. This must be multiplied by the number of iterations (4 or more). The total number of calculations can easily exceed 1000, just to calculate the conditions at one operating point.

Chapter 3

EFFECTIVENESS MODEL

Because of the prohibitive calculations involved in the finite-difference model of a packed-bed liquid-desiccant heat and mass exchanger, and the difficulties inherent in adapting other existing models to simulations of liquid-desiccant systems, it was necessary to develop a new model. The desired characteristics of the new model are speed, accuracy and versatility. It should be based upon fundamental equations, like the finite-difference model, so that it is easy to understand the parameters, and modify them as necessary.

A model with these characteristics, an effectiveness model, has been developed for cooling towers by Braun [8]. Fundamentally, there are very few differences between cooling towers and liquid-desiccant components. Both are characterized by a combination of heat and mass transfer where the mass transfer is driven by a difference between the partial pressure of the water vapor in the air and the vapor pressure of a liquid. The difference is that while a cooling tower (with water as the working liquid) usually only humidifies air, a liquid-desiccant chamber (using a salt-water solution) can humidify or dehumidify, depending on the operating conditions.

Braun's effectiveness model has been modified to model the performance of a liquid-desiccant heat and mass exchanger. Some modifications were necessary, such as including a salt mass balance, but the equations are basically the same.

3.1 Derivation

Like the finite-difference model of a packed-bed liquid-desiccant heat and mass exchanger, the effectiveness model is derived from the governing differential equations, with certain simplifying assumptions. These equations differ slightly from those of the finite-difference model derivation because this derivation follows Braun [8] while the finite-difference model derivation was based upon the derivation given by Factor and Grossman [5]. The differences result from changing the control volume over which the mass and energy balances are taken.

Figure 3.1 is a schematic of the liquid-desiccant chamber with a differential slice shown. An energy balance on the differential slice gives:

$$\dot{m}_s dh_s + h_s d\dot{m}_s = \dot{m}_a dh_a \quad (3.1)$$

A mass balance on the salt in the slice shows that the amount of salt is constant:

$$0 = d(\dot{m}_s \xi) \quad (3.2)$$

A mass balance on the solution side of the slice shows:

$$d\dot{m}_s = \dot{m}_a d\omega_a \quad (3.3)$$

Integrating Equation 3.3 from the bottom of the slice to the top of the chamber shows:

$$\dot{m}_s = \dot{m}_{s,i} - \dot{m}_a (\omega_{a,o} - \omega_a) \quad (3.4)$$

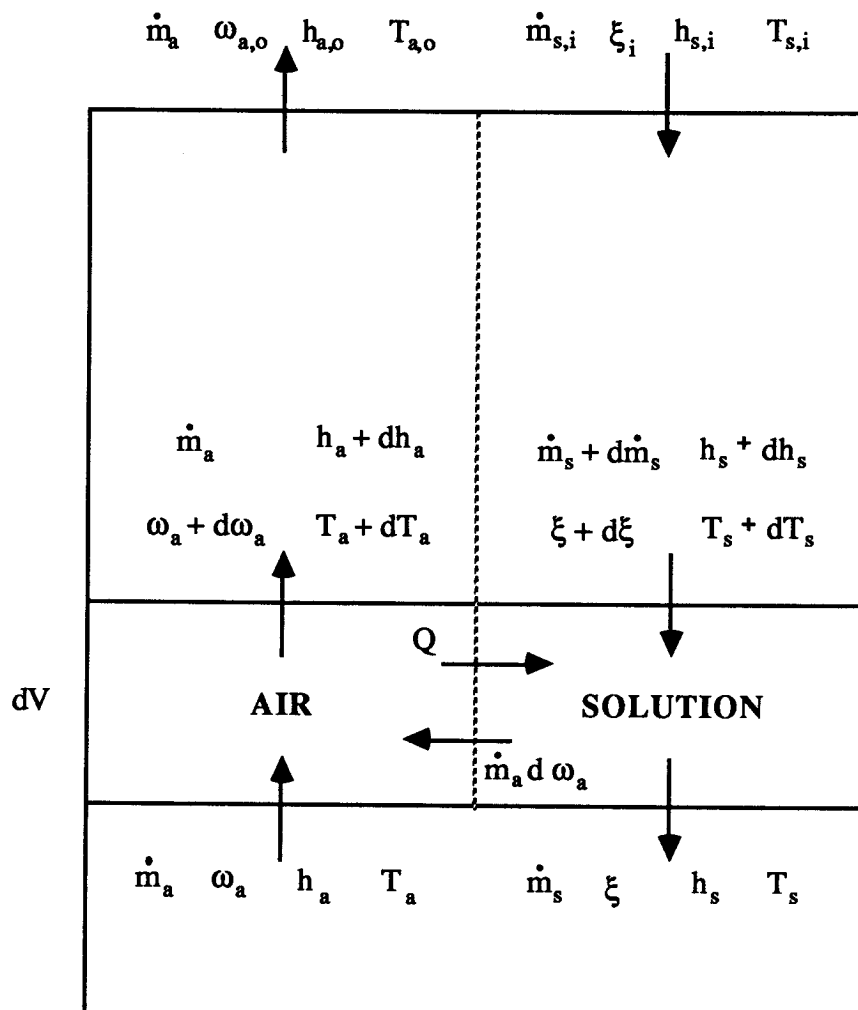


Fig. 3.1 Schematic of Liquid-Desiccant Chamber used to Derive Effectiveness Model

Define the heat transfer coefficient, h_C , with an air-side heat transfer equation,

$$\dot{m}_a dh_a = h_C A_V dV (T_s - T_a) + h_{v, T_s} \dot{m}_a d\omega_a \quad (3.5)$$

where the enthalpy of water vapor at solution temperature is defined as the sum of the vapor enthalpy at 0 °C and the product of the vapor specific heat and the solution temperature [°C].

$$h_{v, T_s} = h_{v, 0} + c_{p, v} T_s \quad (3.6)$$

The mass transfer coefficient, h_D , is defined by an air-side mass transfer equation:

$$\dot{m}_a d\omega_a = h_D A_V dV (\omega_{T_s, \text{sat}} - \omega_a) \quad (3.7)$$

The reference states where enthalpy is equal to zero are taken to be dry air at 0 °C and liquid water at 0 °C. Define the enthalpy of the desiccant solution as the sum of the solution enthalpy at concentration ξ and 0 °C and the product of the solution specific heat and temperature.

$$h_s = c_{p, s} T_s + h_{0, \xi} \quad dh_s = c_{p, s} dT_s \quad (3.8, 3.9)$$

Define the enthalpy of moist air as the sum of two products: the moist air specific heat and air temperature, and the humidity ratio and the enthalpy of water vapor at 0 °C.

$$h_a = c_{p, m} T_a + \omega_a h_{v, 0} \quad (3.10)$$

The Lewis number and NTU are defined here as in Chapter 2:

$$Le = \frac{h_C}{h_D c_{p, m}} \quad (3.11)$$

$$NTU = \frac{h_D A_V V_T}{\dot{m}_a} \quad (3.12)$$

The effectiveness model results form the following algebraic manipulations.

- 1) Substitute Equation 3.3 into 3.1:

$$dh_s = \frac{1}{\dot{m}_s} (\dot{m}_a dh_a - h_s \dot{m}_a d\omega_a) \quad (3.13)$$

- 2) Combine Equations 3.4 and 3.13:

$$dh_s = \frac{\dot{m}_a dh_a - h_s \dot{m}_a d\omega_a}{\dot{m}_{s,i} - \dot{m}_a (\omega_{a,o} - \omega_a)} \quad (3.14)$$

- 3) Divide the numerator and denominator of Equation 3.14 by the air mass flow rate:

$$dh_s = \frac{dh_a - h_s d\omega_a}{\dot{m}_{s,i} / \dot{m}_a - (\omega_{a,o} - \omega_a)} \quad (3.15)$$

- 4) Combining Equations 3.5, 3.7, 3.11 and simplifying gives:

$$\dot{m}_a dh_a = h_D A_V dV \left[c_{p,m} Le (T_s - T_a) + h_{v,T_s} (\omega_{T_s,sat} - \omega_a) \right] \quad (3.16)$$

- 5) Substituting Equation 3.9 into 3.15 gives:

$$dT_s = \frac{1}{c_{p,s}} \left(\frac{dh_a - c_{p,s} T_s d\omega_a}{\dot{m}_{s,i} / \dot{m}_a - (\omega_{a,o} - \omega_a)} \right) \quad (3.17)$$

- 6) Rewrite Equation 3.16 with bracketed portion algebraically manipulated (See Equations 3.16a thru 3.16g in Appendix B1):

$$\dot{m}_a dh_a = h_D A_V dV Le \left[(h_{T_s,sat} - h_a) + \left(\frac{1}{Le} - 1 \right) h_{v,s} (\omega_{T_s,sat} - \omega_a) \right] \quad (3.18)$$

7) Combine both Equations 3.18 and 3.7 with definition of NTU:

$$\frac{dh_a}{dV} = \frac{NTU}{V_T} Le \left[(h_{T_{s,sat}} - h_a) + \left(\frac{1}{Le} - 1 \right) h_{v,s} (\omega_{T_{s,sat}} - \omega_a) \right] \quad (3.19)$$

$$\frac{d\omega_a}{dV} = \frac{NTU}{V_T} (\omega_{T_{s,sat}} - \omega_a) \quad (3.20)$$

Equations 3.17, 3.19 and 3.20 form a detailed model of the heat and mass exchange. In the analysis of a cooling tower, Merkel [9] suggests these additional assumptions: first, neglect any change in the liquid mass flow rate in the chamber; second, assume the Lewis number to be approximately equal to one.

Equation 3.17 and 3.19 become:

$$\frac{dT_s}{dV} = \frac{1}{c_{p,s}} \left(\frac{(dh_a/dV) \dot{m}_a}{\dot{m}_s} \right) \quad (3.21)$$

$$\frac{dh_a}{dV} = \frac{NTU}{V_T} (h_{T_{s,sat}} - h_a) \quad (3.22)$$

The saturation specific heat is the derivative of the saturated air enthalpy with respect to temperature:

$$C_{sat} = \frac{dh_{T_{s,sat}}}{dT_s} \quad (3.23)$$

The capacitance ratio, m^* , is similar to the capacitance ratio used in sensible heat exchangers:

$$m^* = \frac{\dot{m}_a C_{sat}}{\dot{m}_s c_{p,s}} \quad (3.24)$$

With the chain rule, equation 3.21 becomes:

$$\frac{dh_{T_s, sat}}{dV} = m^* \frac{dh_a}{dV} \quad (3.25)$$

If C_{sat} is constant over the operating conditions of the chamber, Equations 3.22 and 3.25 are exactly analogous to a sensible heat exchanger, and can now be modeled using a heat exchanger counterflow effectiveness relationship: (See Appendix B2 for a detailed solution of the differential equations)

$$\epsilon = \frac{1 - e^{-NTU(1 - m^*)}}{1 - m^* e^{-NTU(1 - m^*)}} \quad (3.26)$$

The solution of Equations 3.22 and 3.25 for the air outlet enthalpy gives:

$$h_{a, o} = h_{a, i} + \epsilon (h_{T_s, sat, i} - h_{a, i}) \quad (3.27)$$

In order to find the outlet humidity ratio, an 'effective' state is found which results in the correct air outlet enthalpy:

$$h_{T_s, sat, eff} = h_{a, i} + \frac{h_{a, o} - h_{a, i}}{1 - e^{-NTU}} \quad (3.28)$$

Using this enthalpy with the condition of saturation to find the corresponding effective humidity ratio, Equation 3.20 is integrated to find the outlet air humidity ratio:

$$\omega_{a, o} = \omega_{T_s, sat, eff} + (\omega_{a, i} - \omega_{T_s, sat, eff}) e^{-NTU} \quad (3.29)$$

In summary, the steps for solving for the outlet states of a liquid-desiccant chamber using the effectiveness model are:

- 1) Determine the value of NTU for the system and conditions.

- 2) Calculate the saturation specific heat for the range of conditions expected (Equation 3.23).
- 3) Calculate the capacitance ratio, m^* (Equation 3.24).
- 4) Calculate the effectiveness (Equation 3.26).
- 5) Calculate the air outlet enthalpy (Equation 3.27).
- 6) Use an energy balance to calculate the solution outlet enthalpy.
- 7) Find the effective saturation enthalpy (Equation 3.28).
- 8) Use this enthalpy and a saturated condition to find the effective saturation humidity ratio ($\omega_{T_s, \text{sat}, \text{eff}}$).
- 9) Find the air outlet humidity ratio (Equation 3.29).
- 10) Use mass balances and the known states to calculate solution outlet flow rate, concentration and temperature, and air outlet temperature.

The calculations required to solve for one operating condition with the effectiveness model are dramatically less extensive than those of the finite-difference model. A listing of a FORTRAN program for the effectiveness model is provided in Appendix F. For one condition, the following quantities must be calculated: the solution specific heat, the air saturation specific heat, the capacitance ratio, the effectiveness, the air saturation enthalpy, the outlet air enthalpy, the effective saturated air enthalpy, the effective saturated air humidity ratio, the air outlet humidity ratio, the solution outlet mass flow rate, the solution outlet enthalpy, the solution outlet concentration, the solution outlet temperature, and the air outlet temperature. Iteration is necessary, because the calculation of the air saturation specific heat depends upon a guessed value of the outlet temperature. Usually, the calculations converge in 2 or 3 iterations, with the new guessed solution outlet temperature equal to the solution outlet

temperature calculated in the last iteration. This model gives results with only about 40 main calculations. The subroutines that are called, however, prevent the effectiveness model from running 25 (1000 calculations per point in the finite-difference model divided by the 40 calculations per point in the effectiveness model) times faster, and in practice, it usually runs only about 16 times faster than the finite-difference model.

3.2 Adding Lewis Number to Effectiveness Model

In the derivation of the effectiveness model, it was assumed that the Lewis number is approximately equal to one for the conditions and solutions in liquid-desiccant systems. It would be of some use to be able to use the effectiveness model at conditions away from a Lewis number of one. Equation 3.19 immediately precedes the unity Lewis number assumption:

$$\frac{dh_a}{dV} = \frac{NTU \text{ Le}}{V_T} \left[(h_{T_{s, sat}} - h_a) + \left(\frac{1}{Le} - 1 \right) h_{v, s} (\omega_{T_{s, sat}} - \omega_a) \right] \quad (3.19)$$

The second term in the brackets must be dropped for simplification. If the value of the Lewis number is near one, the second term should be at least an order of magnitude less than the first. In the remainder of the analysis, it is not necessary to drop the first Lewis number in this equation. Rather, it can be carried through as part of the NTU, or perhaps by creating a new variable, $NTU^* = Le \text{ NTU}$. A similar adjustment was used by Jeffreson [10] in assuming an effective NTU number that included the effects of the Biot number. Equations 3.20, 3.26, 3.28 and 3.29 become:

$$\frac{d\omega_a}{dV} = \frac{NTU^* / Le}{V_T} (\omega_{T_{s, sat}} - \omega_a) \quad (3.30)$$

$$\epsilon = \frac{1 - e^{-NTU^*(1-m^*)}}{1 - m^* e^{-NTU^*(1-m^*)}} \quad (3.31)$$

$$h_{T,sat,eff} = h_{a,i} + \frac{h_{a,o} - h_{a,i}}{1 - e^{-NTU^*}} \quad (3.32)$$

$$\omega_{a,o} = \omega_{T,sat,eff} + \left(\omega_{a,i} - \omega_{T,sat,eff} \right) e^{-NTU^*/Le} \quad (3.33)$$

The results of including the Lewis number in the effectiveness model will be compared with the finite difference model in the next section, and will be compared with experimental data in Chapter 4.

3.3 Comparison with the Finite Difference Model

Appendix A, which provides the finite-difference model results, also shows the effectiveness model results for the 16 cases tested. For all 16 cases, the basic effectiveness model (assuming $Le = 1$) was compared with the finite difference model. For all of the cases, the two models agreed very closely, with the effectiveness model results usually following the finite-difference model results for the $Le = 1$ case. In the cases where the finite-difference model results showed a peak or a dip, the effectiveness model also showed these, except to a lesser magnitude. Case 3 showed typical results, seen in Figure 3.2 thru 3.4.

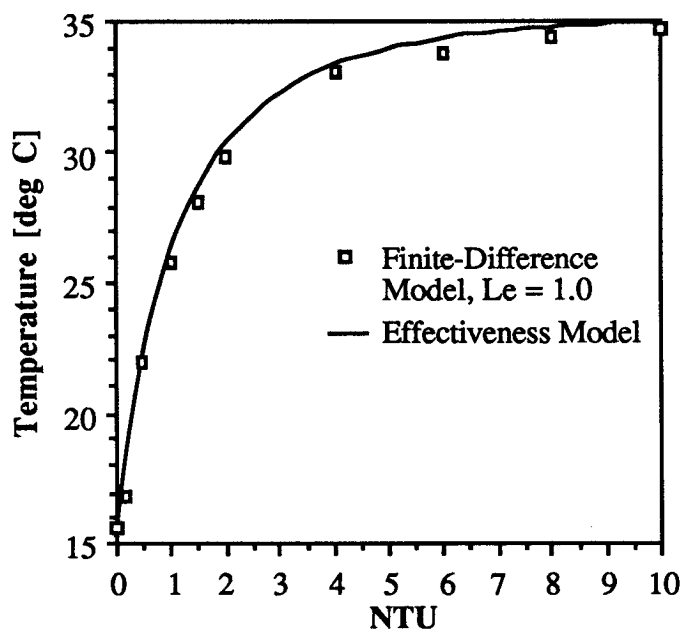


Fig. 3.2 Air Outlet Temperature vs. Component Size for Case 3

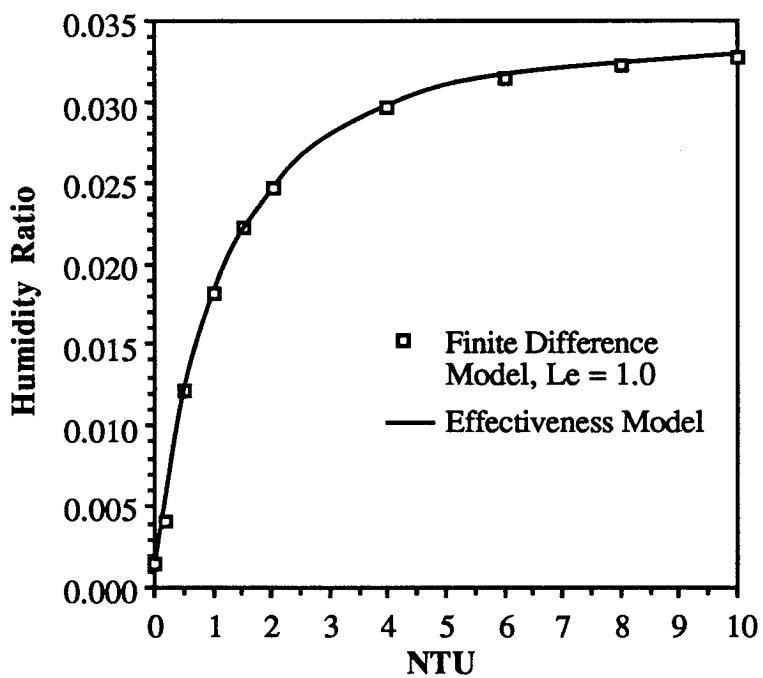


Fig. 3.3 Air Outlet Humidity Ratio vs. Component Size for Case 3

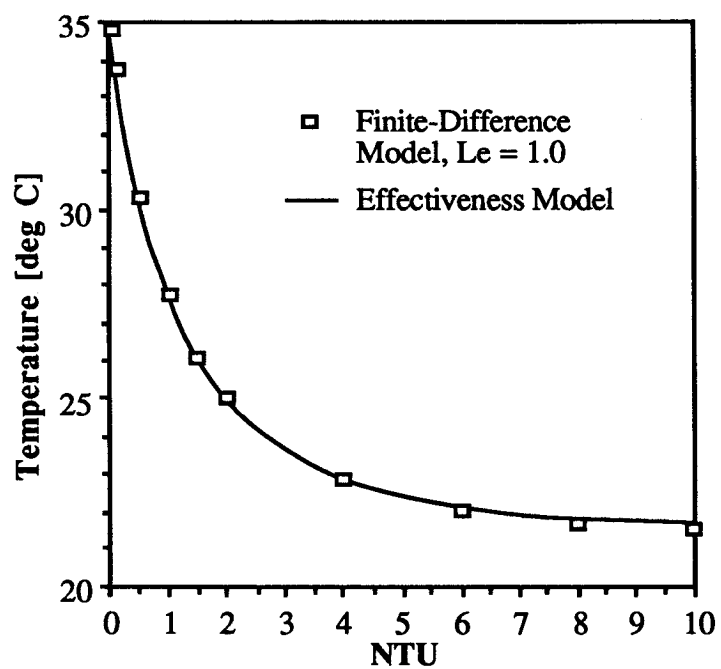


Fig. 3.4 Solution Outlet Temperature vs. Component Size for Case 3

Case 11 showed the dip in the effectiveness model results as having a smaller magnitude than the results of the finite-difference model. This is seen in Figure 3.5.

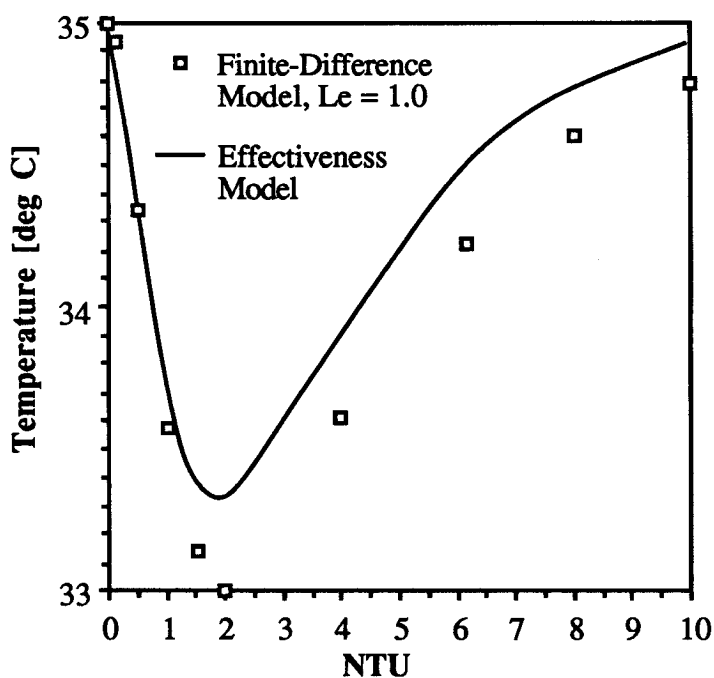


Fig. 3.5 Air Outlet Temperature vs. Component Size for Case 11

The reason for the smaller dip in the effectiveness model is the relatively small number of nodes used in the finite-difference model. The magnitude of the dip decreased when the number of nodes was increased. Perhaps the effectiveness model comes close to representing a very large number of nodes.

When the capability for considering the Lewis number is added to the effectiveness model, the results show a greater dependence on the Lewis number than in the finite-difference model. Figures 3.6 and 3.7 show the results of the effectiveness model at three different Lewis numbers for Case 4. The outlet humidity ratios are not shown because they do not show a dependence on Lewis number.

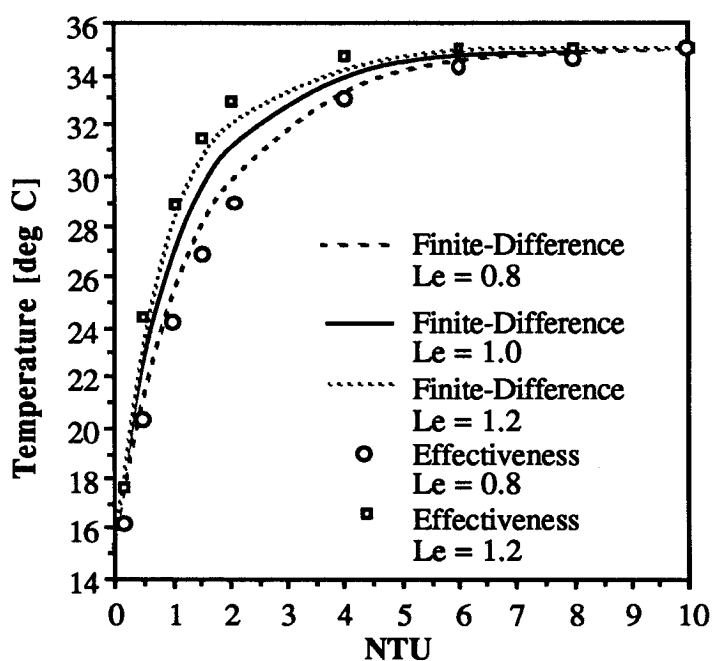


Fig. 3.6 Air Outlet Temperature vs. Component Size for Case 4

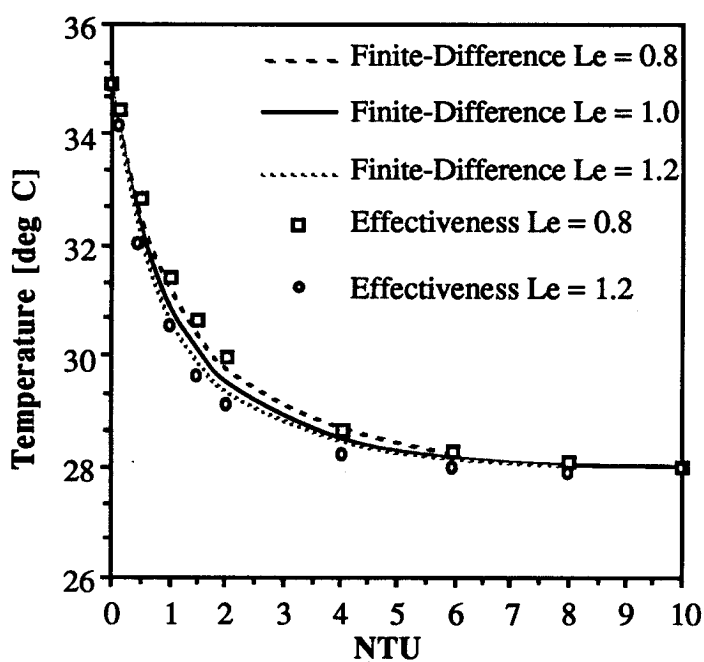


Fig. 3.7 Solution Outlet Temperature vs. Component Size for Case 4

The effectiveness model's results for different Lewis numbers show a slightly exaggerated Lewis number effect. This may be due to ignoring the second term in Equation 3.19. The ramifications of the exaggeration for using the model in simulations are not very great. The effects of different Lewis numbers can be shown with the effectiveness model, but only as trends, not as exact values.

The effectiveness model has been shown to compare well with a more detailed finite-difference model. Its advantage over a finite-difference model is that its use of computer time is considerably smaller. The effectiveness model shows no major disadvantages relative to a finite-difference model.

Chapter 4

EXPERIMENTAL DATA

Unfortunately, there is not a lot of non-proprietary experimental data available on the performance of liquid-desiccant systems. Few commercial systems are in operation, and of those, even fewer are instrumented to record data. In the published literature, there are several liquid-desiccant experimental systems. A majority of these are not applicable to this study. The type of liquid-desiccant component of interest here is a closed (not open to the ambient conditions) adiabatic (no internal heating or cooling coils) chamber filled with packing material. One experimental system and one instrumented commercial system will be examined here, and their operation compared with the predictions of the effectiveness model.

4.1 CSU Experimental Data

Lenz, et.al. at Colorado State University have studied a open-cycle LiCl absorption chiller combined with a packed-bed regenerator. The publications resulting from these studies often included data from the regenerator operation, but the published data often was not in a form useful to this study.

One data set that was useful was in a CSU report to the U.S. DOE in 1981 [11], where seven regenerator data points are given. These points were converted from volumetric to mass flow rates and from English to SI units. In these data, the solution

concentration was given as a mean value. A mass balance was used to calculate the corresponding inlet and outlet values. The data set is shown in Table 4.1. A typical data point, #3, is shown in Figure 4.1.

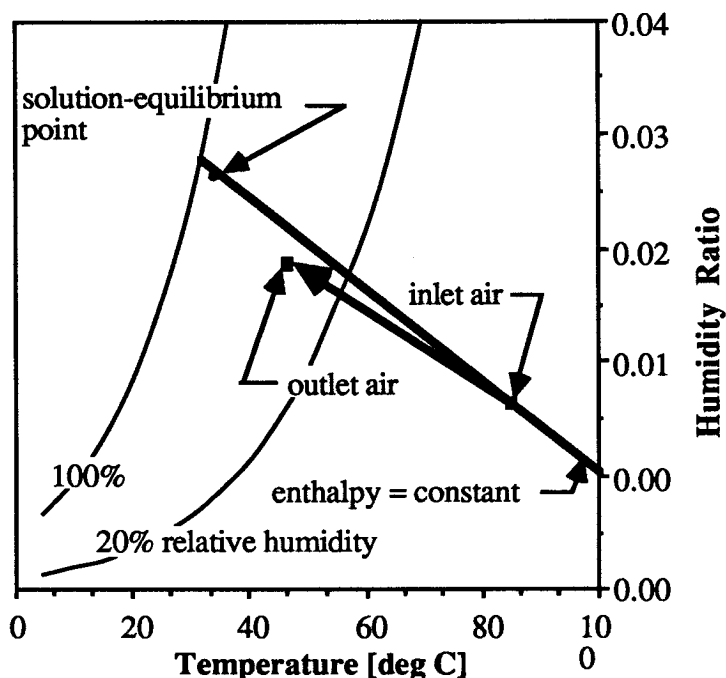


Fig. 4.1 Air and Solution States from 1st CSU Data Set, Point #3

When trying to compare the results from an experiment with the effectiveness model, it is necessary to find an appropriate value of NTU. In most cases, a value for the NTU can be backed out from each data point simply by calculating the effectiveness, which is the ratio of the actual enthalpy change of the air to the maximum possible enthalpy change of the air (the enthalpy change where the air outlet is in equilibrium with the solution inlet). In this set of data from CSU, that technique was ineffective. The reasons for this are obvious when looking at figure 4.1. The 'maximum' air enthalpy change would, in this case, be almost exactly zero!

TABLE 4.1

1981 CSU Liquid-Desiccant Data

air mass	air inlet	air outlet	air inlet	air outlet	solution	solution	solution	solution	solution
flow rate	temp.	temp.	humidity	humidity	mass flow	inlet	outlet	inlet	outlet
[kg/s]	[deg C]	[deg C]	[kg/kg]	[kg/kg]	[kg/s]	[deg C]	[deg C]	[wt. fract]	[wt. fract]
0.16656	108.9	52.2	0.0106	0.0249	0.10577	38.3	38.9	0.2164	0.2116
0.16089	108.9	52.2	0.0175	0.0270	0.10639	37.8	38.1	0.2256	0.2224
0.14134	85.0	46.7	0.0164	0.0288	0.10049	33.9	34.4	0.1210	0.1190
0.14193	83.3	51.1	0.0167	0.0288	0.09852	35.0	35.0	0.0847	0.0833
0.14210	85.6	44.4	0.0120	0.0237	0.10372	35.6	36.1	0.1794	0.1766
0.15239	81.1	40.0	0.0116	0.0203	0.10956	36.1	36.7	0.2757	0.2724
0.15070	82.2	48.9	0.0167	0.0271	0.09917	34.4	34.4	0.0968	0.0953

TABLE 4.2

Comparison of Effectiveness Model with 1981 CSU Liquid-Desiccant Data

Calculated NTU	Change in		Change in		Air Outlet		Air Outlet		Solution Outlet	
	Air Enthalpy	Solution Enthalpy	Air Enthalpy	Solution Enthalpy	Temperature	Humidity Ratio	Temperature	Humidity Ratio	Temperature	Humidity Ratio
	[kW]	[kW]	[kW]	[kW]	[deg C]	[kg/kg]	[deg C]	[kg/kg]	[deg C]	[kg/kg]
	CSU	model	CSU	model	CSU	model	CSU	model	CSU	model
0.78	3.5369	0.2247	0.0853	-0.2247	52.2	70.893	0.0249	0.0249	38.9	39.819
0.65	5.4284	1.4456	0.0676	-1.4456	52.2	75.872	0.027	0.0270	38.1	42.601
0.94	1.0419	0.0315	0.059	-0.0315	46.7	53.623	0.0288	0.0288	34.4	34.659
0.74	0.2595	-0.6608	-0.1217	0.6608	51.1	57.238	0.0288	0.0288	36.0	33.853
0.79	1.6768	-0.2924	0.0436	0.2924	44.4	57.878	0.0237	0.0237	36.1	35.33
1.06	2.9481	0.9524	-0.1043	-0.9524	40.0	52.934	0.0203	0.0203	36.7	39.36
0.65	1.1151	-0.4395	-0.1414	0.4395	48.9	58.767	0.0271	0.0271	35.4	33.782

Therefore, any deviation from a direct line from inlet toward equilibrium would give an actual air enthalpy change greater than the 'maximum' possible, resulting in an effectiveness that could be negative, or have an absolute value greater than one.

In this case, it is necessary to use trial and error to find the closest-fitting value for NTU for each point. The procedure for finding values of NTU for the data points requires running the effectiveness model at various NTUs until one of the outlet values from the effectiveness model matches an experimental value. In this case, the outlet humidity ratios were matched to 4 decimal places. Humidity ratios were chosen because they are more likely to be independent of energy losses than air or solution temperatures. Ideally, for similar operating conditions, the calculated NTUs should be close together. The results are shown in Table 4.2. The changes in enthalpy in Table 4.2 are calculated by taking the product of the air or solution inlet mass flow rate and specific enthalpy minus the product of the outlet mass flow rate and specific enthalpy. Therefore, a positive enthalpy change indicates that the inlet enthalpy is higher than the outlet.

The energy balances on the CSU data clearly do not close. A possibility for this, especially considering the high operating temperatures, is a substantial heat loss to the environment. The effectiveness model assumes adiabatic operation and can not account for the heat loss. The solution outlet temperatures predicted by the effectiveness model are fairly close to the experimental values when the humidity is matched.

A more recent liquid-desiccant system at CSU included a packed bed conditioner and regenerator that use a LiBr-water solution. The first data received from this system was conditioner data included as part of a draft copy of a report [12]. Upon examination of the data, it was found that the energy flows did not come close to

balancing. At one typical point, the solution gained three times as much energy as the air lost. Since the temperatures in the system were in the 30-40 °C range, it seems unlikely that energy could have been gained from the environment.

After further communication with the CSU experimenters, it was found that the solution outlet temperature measurement was suspected. The outlet flow was not isothermal, resulting in inaccurate measurements. Further data were received after this problem was corrected. The new data, consisting of 7 points, was found to have much better energy balances and was compared with the effectiveness model.

The data represents conditioner operation at a relatively small effectiveness. Point 4 had the closest energy balance, and its process is shown in Figure 4.2.

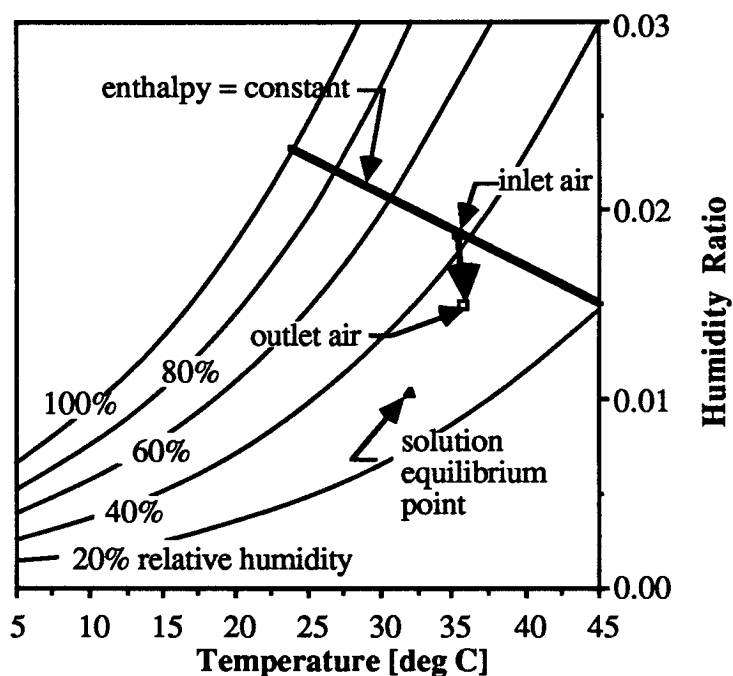


Fig. 4.2 Air and Solution States from 2nd CSU Data Set, Point #4

TABLE 4.3

Comparison of Effectiveness Model with 1987 CSU Liquid-Desiccant Data

Calculated Effectiveness	Air Temperatures		Air Humidity Ratios				Solution Temperatures				[% error]	
	[deg C]		[kg/kg dry air]				[deg C]					
	inlet	outlet	inlet	outlet	inlet	outlet	inlet	outlet	inlet	outlet		energy balance
0.282	33.0	32.3	35.3	0.0185	0.0142	0.0130	27.8	34.3	35.9	19.65		
0.293	31.4	31.0	33.7	0.0195	0.0158	0.0147	28.0	33.0	33.9	16.03		
0.290	34.2	32.9	33.9	0.0194	0.0160	0.0156	28.1	33.1	34.2	17.69		
0.380	35.4	34.8	35.7	0.0187	0.0153	0.0149	32.0	36.6	36.7	1.06		
0.265	35.8	33.7	32.5	0.0152	0.0138	0.0143	28.0	31.5	31.9	9.50		
0.271	36.1	33.8	32.2	0.0148	0.0135	0.0141	27.7	31.3	31.8	11.52		
0.385	36.1	32.7	31.5	0.0162	0.0144	0.0149	27.7	31.4	32.5	23.08		

Since the change in the air state in this data set occurs with a significant change in enthalpy, it was a simple matter to calculate effectivenesses from the data. The effectiveness model was then compared with the results shown in the experimental data. Table 4.3 shows this, including the errors in the experimental energy balances.

The change in air temperature is so low for the first four points that any comparison in results is inconclusive. For the last three data points, the effectiveness model underpredicts air temperature change by about 20%. The effectiveness model's prediction of outlet air humidity ratios is slightly off, with some points low, and some high. Solution outlet temperatures come closest to matching.

It is not possible for the effectiveness model to predict the CSU results exactly. The effectiveness model relies on an energy balance to calculate the outlet states, while the CSU results violate energy balances to some extent. From the first CSU data set, it can be seen that the effectiveness cannot be calculated from experimental enthalpy changes when the state of the air varies close to a line of constant enthalpy. At these conditions it is necessary to determine an NTU based on matching one of the outlet parameters, either air humidity ratio, or air or solution temperature. Since the desired operating condition of a desiccant chamber is adiabatic, the effectiveness model assumes negligible losses.

The effectiveness model shows better agreement with the second data set. This is probably due to the better energy balances. The differences between the second data set and the effectiveness model could be due to several factors, including Lewis number effects and experimental error. The correlation of Lewis number effects with experimental data is not examined in this section because of the small quantity of data. The data analyzed in the next section is more appropriate for a Lewis number analysis.

4.2 SMVA Experimental Data

Data have been received from Tennessee Valley Authority's monitoring of the Science Museum of Virginia (SMVA) liquid desiccant system [13]. Figure 4.3 shows a schematic of the system including the monitoring points. The most recent data available are those from July 10, 1987 to November 31, 1987. The data from September 7, 8, 15, 16, 17 and October 22 - 26 are missing completely, but otherwise the data are reported continuously at 15 minute intervals.

The liquid desiccant system at SMVA is quite complicated. Provisions have been made for various methods of heating and cooling the liquid desiccant solution as necessary for regeneration and conditioning, respectively. We will not be concerned with these different methods. Previous studies [3,4] at the Solar Energy Laboratory have covered them. Rather, this study's purpose is only to determine whether an effectiveness model of a liquid desiccant heat and mass exchanger can accurately predict that component's performance.

Upon review of this data, several problems have been found. The simplest of these is that some of the units were reported incorrectly. Liquid flow rates reported in gallons/scan (a scan is the interval at which data is reported, or 15 minutes) are actually measured in gallons per minute. From here, the problems get more serious. Air flow rates in the conditioner are missing from July 14 to September 14. Energy balances in the conditioner and regenerator fall apart as the weather gets colder. The regenerator consistently violates the second law of thermodynamics. Since the effectiveness model calculations are based on first and second law principles, data which violates these is of little use in validating the model. Following is a more detailed analysis of the experimental data.

Science Museum of Virginia Desiccant Cooling System Instrumentation Schematic

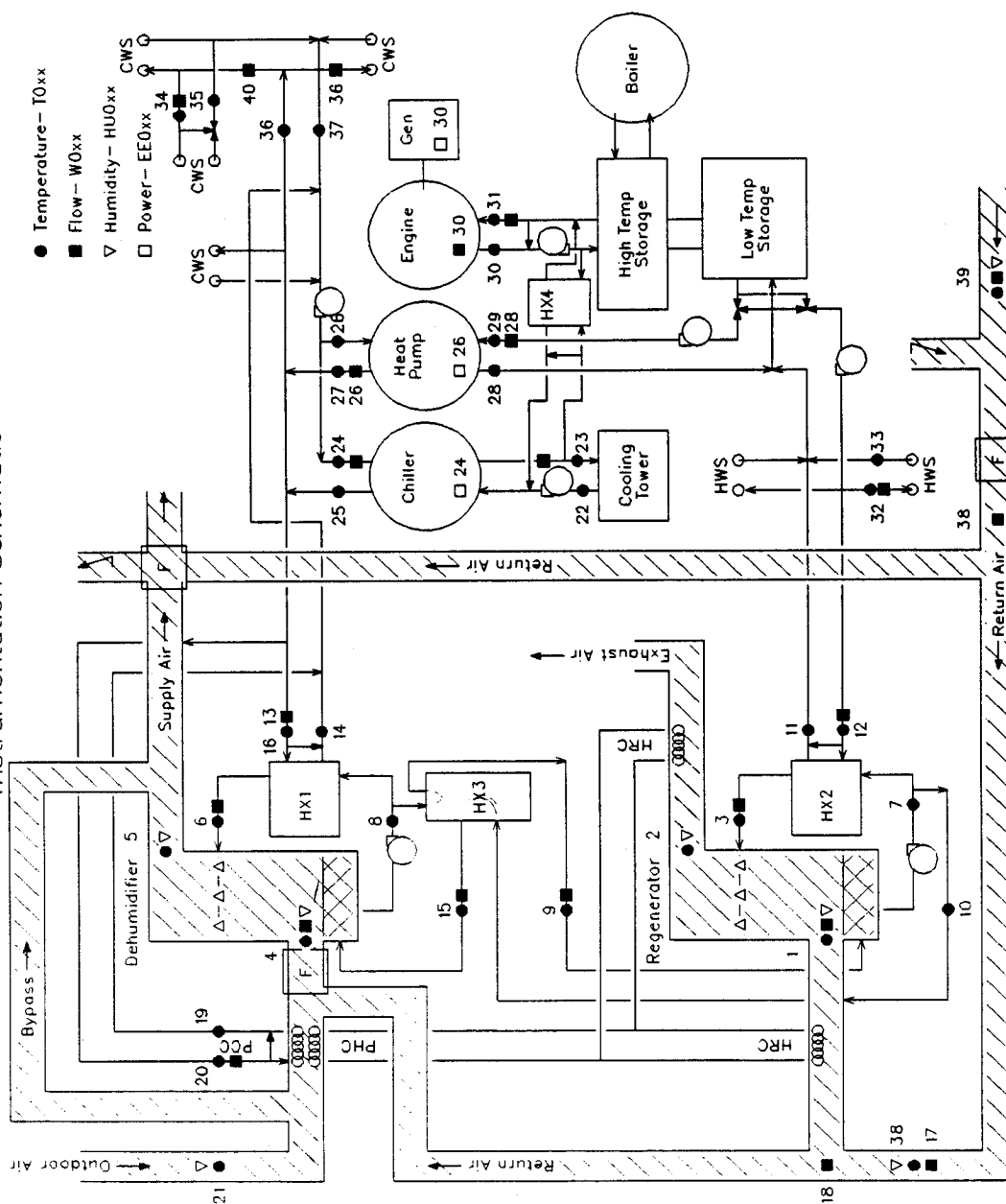


Fig. 4.3 Schematic of Science Museum of Virginia Liquid-Desiccant System

The experimental data from SMVA arrived at the Solar Energy Laboratory in both printed form and on magnetic tape. It was accompanied by a few pages of explanation, including the gain factors, biases and acceptable ranges of each sensor, a schematic of the system and the sensor locations (fig. 4.3), some notes about specific sensor failures, and the units associated with each data point. All 64000 lines of data were in one file. A file this size is too big to handle easily, so it was divided into sections of approximately ten days each. Each of these 14 files was then examined separately.

When a sensor gave a value that fell outside of the acceptable range, or failed altogether, the data point was recorded as ' . '. This character cannot be read into a FORTRAN program which is expecting a numerical value, so each occurrence was either replaced with '99999' or the record was eliminated. Large sections of data from October and November were removed from further analysis because of this; the humidity ratios were often below the accepted range of the sensor.

All of the data were recorded in English units, (degrees Fahrenheit, Btu, pounds-mass, CFM, gallons/min, etc.) and flow rates were on a volumetric basis. The FORTRAN program READ does the following:

- reads in the data from the file;

- ignores data for the part of the system not directly relating to the liquid desiccant chambers;

- skips over records where any mass flow rate is equal to zero;

- converts all units to SI ($^{\circ}\text{C}$, kJ, kg, etc.);

calls solution and air density functions to change volumetric flow rates to mass flow rates;

writes converted, edited data to a file for further use.

Data that are not obviously useless is then passed on to the next step of analysis, in the FORTRAN program BAL. This program does the following:

reads in data;

calculates the mass flow rates at points 7, 8, and 10 (see Fig. 4.3)

calculates the rate of dehumidification in the conditioner (kg water taken from air/s);

calculates the rate of regeneration in the regenerator (kg water taken from solution/s);

performs mass balances on the conditioner and regenerator to find the amount of solution leaving the packed beds (This quantity was not measured, probably because the location was not accessible.);

performs a mass balance on the salt in the system to find the concentration of the solution going into the regenerator (The concentration going into the conditioner was assumed to be 28.5%. This has been confirmed by TVA personnel [13] to be the most likely concentration. Since the concentration doesn't change very much anywhere in the system, one concentration can be chosen to identify the operating concentration of the system which is arbitrarily assigned to the conditioner inlet.);

calculates the enthalpies of all streams -- air, water and LiCl solution;

calculates the energy gained or lost by all streams flowing through heat exchangers and uses this information in heat exchanger energy balances;

performs mass balances on the conditioner and regenerator;

uses energy balances on the packing material section of the regenerator and conditioner to find temperature of the solution leaving that section;

uses energy balances on the sump sections to find the temperature of the solution entering the sump (Result should be same as that calculated in the previous item.);

uses changes in air enthalpy in both conditioner and regenerator to calculate the effectiveness of each chamber (100% effectiveness indicates air leaving in equilibrium with incoming solution.);

Once all of these quantities are calculated, some decisions are made about the value of the data. Data is discarded for the following reasons:

conditioner or regenerator operation is opposite intended direction, for example, conditioner adding water to air;

mass flow of solution out of packing material (either conditioner or regenerator) is equal to zero;

energy balance on any heat exchanger off by more than 15%;

mass balance on conditioner or regenerator off by more than 15%;

energy balance on conditioner or regenerator off by more than 15%;

effectiveness of conditioner or regenerator greater than 1.0;

Unfortunately, after these criteria were applied, very few data were left. And, when a criterion is loosened, the data almost inevitably fail somewhere else. Table 4.4

shows where the data points fail. If a point fails any test, it is not even tested for the remaining criteria, so whether it would pass a later test is not known. 'Wrong way' operation indicates the regenerator acting as a conditioner, or vice versa.

TABLE 4.4
Failure of SMVA Data

<u>Conditioner</u>	<u>Regenerator</u>	<u>Test</u>
22.7%	9.5%	wrong way operation
0.07%		flow equal to zero
0.66%	0.62%	HX #2 energy balance
76.5%	82.4%	Cond. or Reg. Energy bal
	7.6%	Effectiveness > 1.0
3022 / 3259	3204 / 3211	# failed / total

None of the 7 regenerator points that passed all criteria were used because they were scattered randomly throughout the data. Conditioner data from July 10 - 12, 1987 was the only large section where none of the points failed. From these points, an average value of NTU was calculated as 1.86. This value of NTU was then used in the effectiveness model to compare its results with the experimental data.

Figures 4.4 and 4.5 show the results of the comparison using the basic effectiveness model ($Le = 1.0$) and an NTU of 1.86. The effectiveness model and the SMVA results agreed almost exactly on the solution outlet temperature, so this comparison is not shown. In his study of the SMVA system, Buschulte [3] used a

curve fit to find a Lewis number for the conditioner of 1.2. The results from this are shown in Figures 4.6 and 4.7.

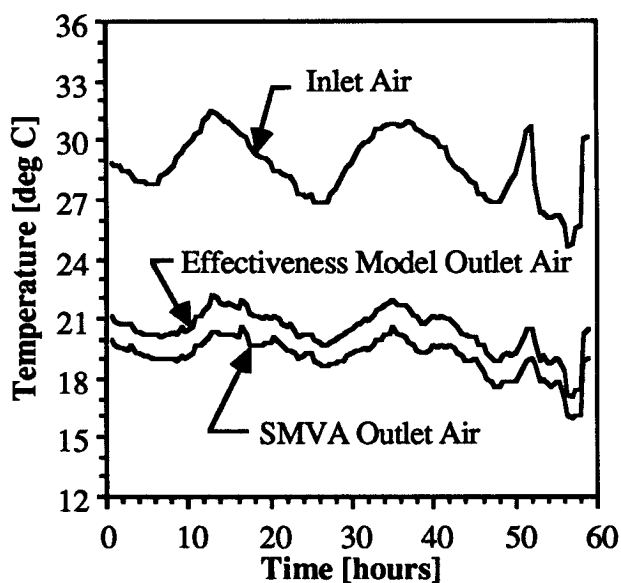


Fig. 4.4 Comparison of Effectiveness Model Outlet Air Temperature with SMVA Data

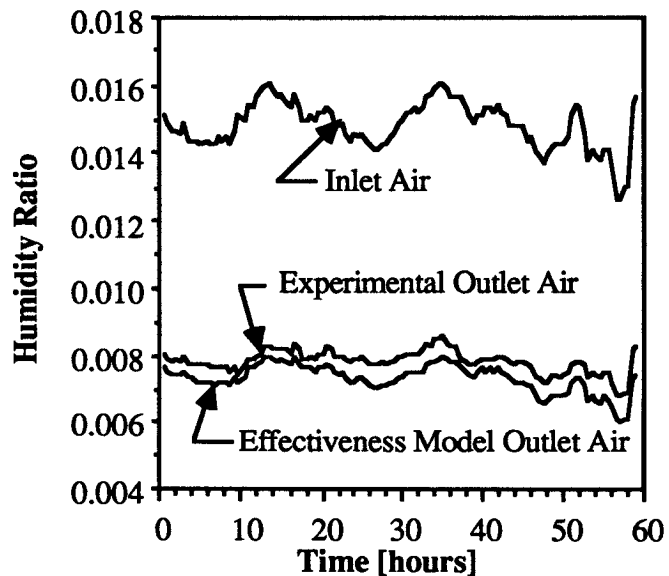


Fig. 4.5 Comparison of Effectiveness Model Outlet Air Humidity Ratio with SMVA Data

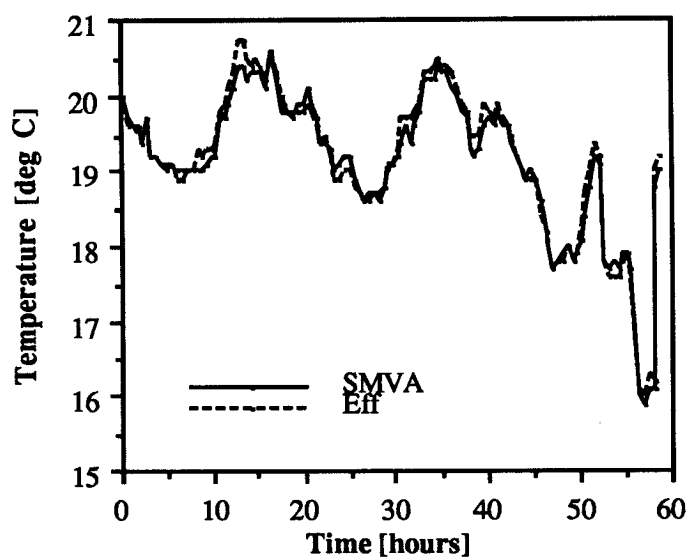


Fig. 4.6 Comparison of Effectiveness Model Air Outlet Temperature with SMVA Data for $Le = 1.2$

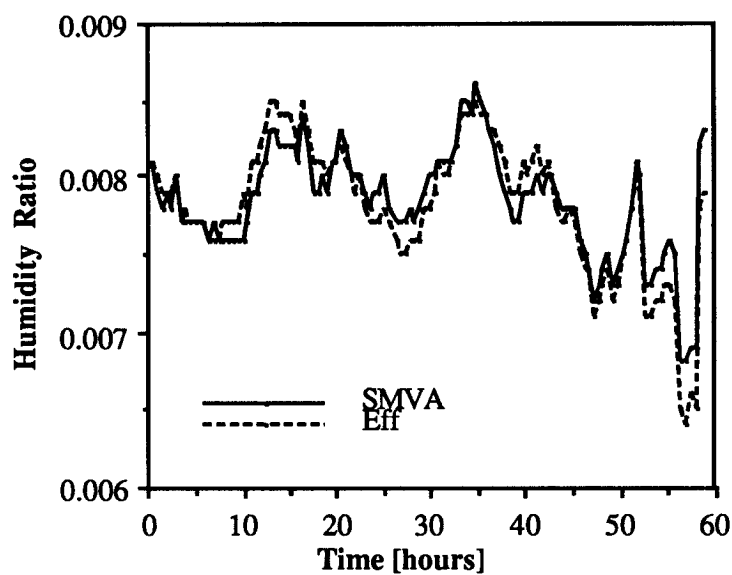


Fig. 4.7 Comparison of Model Air Outlet Humidity Ratio with SMVA Data for $Le = 1.2$

When the effectiveness model is combined with the ability to account for the Lewis number, some experimental results can be predicted fairly accurately. In order to do this, both a value of NTU^* and a value of Le must be chosen. The NTU^* is found from the air-side effectiveness relationship. This alone will result in a correct solution outlet temperature and a correct air outlet enthalpy. A value of Le is chosen that results in the correct balance of air outlet temperature and humidity.

Though agreement between the effectiveness model and the CSU data was not very close, the agreement with the SMVA data was much better. The model is designed for predicting the results of adiabatic systems but could be combined with a heat exchanger model to predict losses.

Chapter 5

LIQUID-DESICCANT SYSTEMS

In this chapter, two liquid-desiccant systems are simulated, and their performances compared. The individual components involved in the simulations are described first, followed by the system configurations.

5.1 Liquid-Desiccant System Components

To simulate the performance of liquid-desiccant systems, models of each of the components involved must be formulated. The models must be combined in such a way that the outputs from one component can be used as inputs to another. For example, the output air temperature and humidity ratio from the conditioner might be used as inputs to an auxiliary air conditioner. TRNSYS [14], a modular transient simulation program, has been designed to simplify the combination of the components. Each component, such as a heat exchanger or a cooling tower, is written as a FORTRAN subroutine. Many standard components, such as heat exchangers, are included in the TRNSYS program and are described in the TRNSYS manual. This chapter describes the non-standard components used in simulations of liquid-desiccant systems. For a TRNSYS simulation, inputs, parameters and outputs must be specified for each component. These are listed in Appendices C1 - C5.

5.1.1 Liquid-Desiccant Conditioner/Regenerator

The liquid-desiccant component model uses the effectiveness relationships developed in Chapter 3. Appendix C1 provides a description of the liquid-desiccant component in a TRNSYS-compatible format and the FORTRAN code.

In the different simulations, the mass transfer - area product of the conditioner is varied to determine the results of changing the 'size' of the conditioner. (See Figures 5.1 and 5.9.) In the TRNSYS component specifications (called a 'deck', see Appendix D) the regenerator inlet LiCl solution concentration and mass flow rate are set equal to constants in order to speed the convergence of the simulations. The concentration at any one point in the system varies little with time, and if an average value is selected as the constant concentration, the actual value differs by less than 0.1%.

5.1.2 Auxiliary Air Conditioner

The air conditioner component used in system simulations was provided by Urban [15]. It is formulated to simulate a generic chiller, as opposed to a specific model. For the rated conditions of a 67 °F evaporator entering wet bulb temperature, a 95 °F condenser entering dry bulb temperature and 37.5 CFM/1000 Btu/hr of cooling capacity, the following values must be specified: COP, by-pass factor (BF, the fraction of air that is not affected by the cooling coils), capacity in Btu/hr, and CFM. These values are modified for off-rated conditions by functions of the entering air state and the condenser temperature [16].

The role of the auxiliary air conditioner in the systems that have been simulated is to ensure that the supply air will meet the zone's latent and sensible loads. Appendix C2 provides a listing of the values that must be provided to simulate this air conditioner with TRNSYS.

5.1.3 Evaporative Cooler

The evaporative cooler model is programmed to show the maximum benefit of evaporative cooling. It does not follow the traditional evaporative cooling effectiveness relationship. It calculates the desired supply air temperature and humidity ratio, based on building loads, and compares the values with its inputs. If the humidity ratio into the evaporative cooler is lower than necessary to meet the latent load, the evaporative cooler will add water equivalent to a specified fraction of the difference between the desired and input humidity ratios. A fraction lower than 100% should be chosen to ensure that the auxiliary air conditioner will not have to do any latent cooling. See Appendix C3 for a listing of the values that must be specified for the simulation of this model and the FORTRAN code.

The desired supply air temperature and humidity ratio are calculated here as in the auxiliary air conditioner model [15]. If the entering humidity ratio is greater than the desired supply humidity ratio, no evaporative cooling is done. Otherwise, the mass flow rate of water added (\dot{m}_w) is calculated by:

$$\dot{m}_w = F \dot{m}_a (\omega_{des} - \omega_{in}) \quad (5.1)$$

where F is a specified fraction.

The exiting air conditions are determined by mass and energy balances:

$$\omega_{\text{out}} = \omega_{\text{in}} + \frac{\dot{m}_w}{\dot{m}_a} \quad (5.2)$$

$$h_{\text{out}} = \frac{\dot{m}_w h_w + \dot{m}_a h_{\text{in}}}{\dot{m}_a} \quad (5.3)$$

The exit air temperature is determined by a call to the psychrometrics subroutine.

5.1.4 Cooling Tower

The cooling tower effectiveness model is described in detail by Braun [8]. In this model, the NTU is calculated from an empirical equation that depends on the ratio of the air and water mass flow rates and two fitted constants: $NTU = c (m_w/m_a)^n$. The values necessary for a TRNSYS simulation appear in Appendix C4. In these simulations, it is operated as a steady state component. It is always operated at 100% fan speed and its sump volume is set equal to zero. This type of operation works well for short, summertime simulations because the lowest temperature possible (limited by the ambient air wet bulb temperature) is always greater than the temperature that would eliminate the need for auxiliary air conditioning. In cooler months when the required supply air temperature is higher and the ambient air wet bulb temperature is lower, the cooling tower would not always have to function at its maximum level to provide full benefits to the cooling system. The cooling tower model has the capability to calculate the required fan power, but, like other parasitics, it is ignored in this set of simulations.

5.1.5 Zone

A separate component has not been formulated for modelling the cooling zone for this group of simulations. Although a detailed zone model is available, a simple one is adequate for demonstrating the general behavior of liquid-desiccant systems. A base load of 100,000 Btu/hr (about 8 tons) was chosen, with the actual value varied diurnally between 82 and 120% of this value. This represents a medium sized commercial building where machinery is operating or people are present most of the 24 hours per day. The latent load is one-third and the sensible load is two-thirds of the total, typical of commercial buildings. The amount of ventilation (fresh) air was set at 15% of the total circulation. The zone temperature is maintained at 24 °C and the humidity ratio at 0.010 kg/kg dry air.

5.1.7 Other Components

One other non-standard TRNSYS component was formulated for use in simulating liquid-desiccant systems. It calculates values of air mass flow rate in lb/hr and kg/s from the inputs of volumetric flow rate, temperature, humidity and atmospheric pressure. The TRNSYS required values are listed in Appendix C5.

The remaining components used are all standard TRNSYS components. For specific details of their use, see the TRNSYS decks in Appendix D and the TRNSYS manual.

5.2 Liquid-Desiccant System Configurations

Two liquid-desiccant system configurations have been examined. The first is a basic system shown in Figure 5.1. In this system, the solution is cooled when passing through a sensible heat exchanger with cooling tower water. The solution is heated in a electric or gas-fired heater. The mixture of fresh and return air is first passed through the conditioner, where it is dehumidified. The auxiliary air conditioner performs any additional latent cooling required, as well as most of the sensible cooling.

In this section, separate COPs are calculated for heating (COP_h) and cooling (COP_c). COP_h and COP_c are determined by dividing the total load on the zone by the heating or cooling energy required, respectively. In calculating an overall COP, the following equation is used:

$$COP = \frac{1}{\frac{1}{\eta_h COP_h} + \frac{1}{\eta_c COP_c}} \quad (5.1)$$

where η_h and η_c are the efficiencies at which heating and cooling energies can be provided. For example, if a gas-fired heater is 88% efficient, and electricity production for cooling is 37% efficient, for $COP_h = 0.385$ and $COP_c = 7.15$, the overall COP is determined by:

$$COP = \frac{1}{\frac{1}{0.88 (0.385)} + \frac{1}{0.37 (7.15)}} = 0.3$$

These values of efficiency ($\eta_h = 0.88$, $\eta_c = 0.37$) are carried through this section to provide an estimate of overall efficiencies of the different systems simulated.

The two systems were both simulated over a six-month period from April 1 to September 30, representing a typical summer season. The weather data used in the

simulations was TMY (Typical Meteorological Year [17]) data representing Miami, Florida. To avoid excessive computer use, some of the example simulations have been performed over a single, typical day. The TMY data for July 1 was chosen to represent a typical summer day in Miami.

The first system was originally simulated with an air flow rate of 3000 CFM and a solution flow rate of 3 kg/s. At these conditions, the required auxiliary heat was excessively high, giving a system COP (total load on zone divided by total heating and cooling energy) of about 0.36. In order to obtain a greater system COP, the solution mass flow rate was reduced to 0.3 kg/s. At the lower mass flow rate, the COP for the first system was approximately 1.3.

The figures in this chapter representing the original mass flow rate, 3 kg/s, are denoted by the letter 'a', as in Fig. 5.2a. Those representing the revised mass flow rate, 0.3 kg/s, are denoted by the letter 'b'. The figures representing both mass flow rates are included to show the effect of the changing mass flow rate.

Figures 5.2a and 5.2b show the effect of changing the conditioner size on the required auxiliary chiller power for a single day.

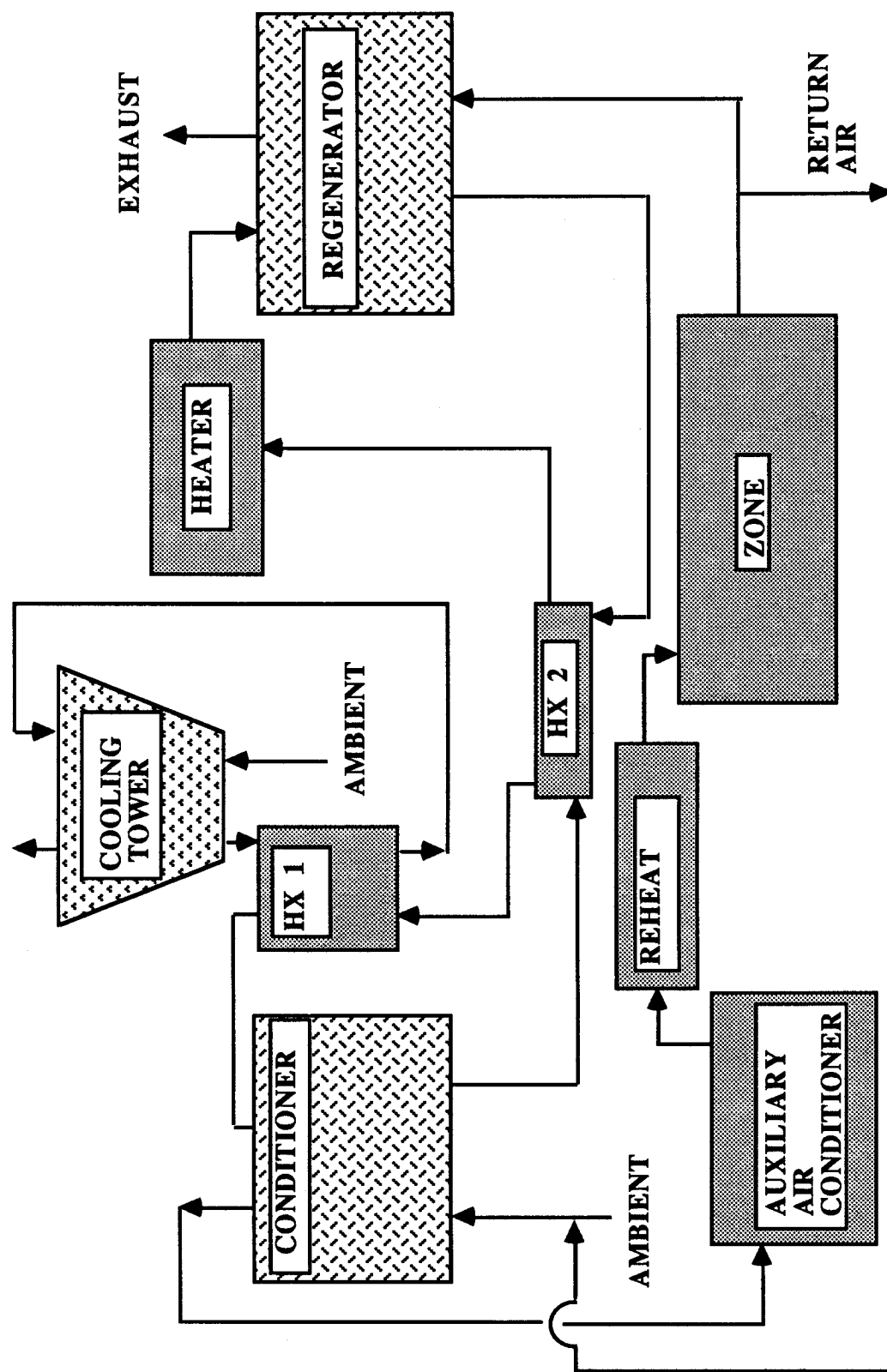


Fig. 5.1 Schematic of First Liquid-Desiccant System Configuration

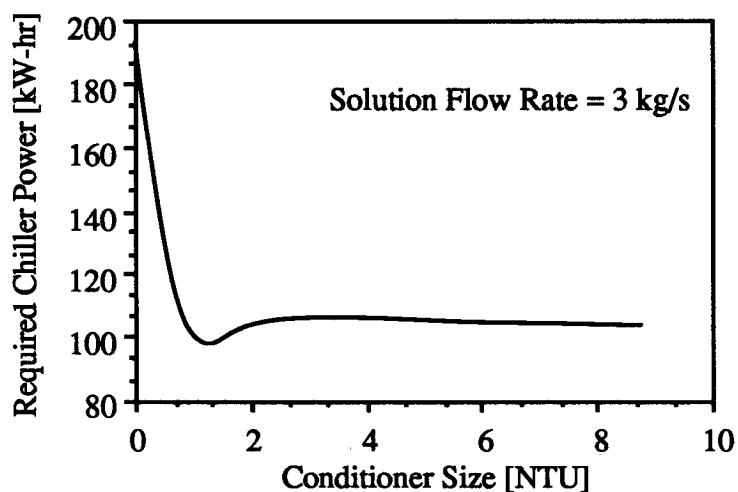


Fig. 5.2a Required Auxiliary Chiller Power vs. Conditioner Size for First System at Solution Flow Rate of 3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

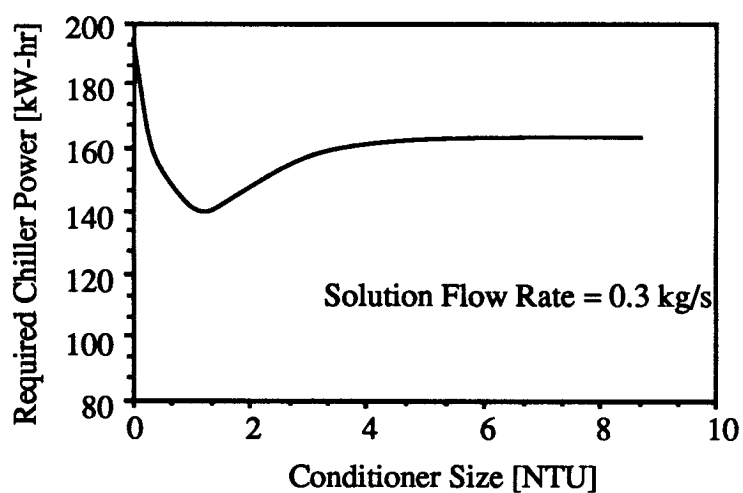


Fig. 5.2b Required Auxiliary Chiller Power vs. Conditioner Size for First System at Solution Flow Rate of 0.3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

As shown in Figures 5.2a and 5.2b, by increasing the conditioner size from zero, the required chiller power is reduced quickly because the conditioner is meeting more and more of the latent load. Once the humidity ratio leaving the conditioner is lower than the required air supply humidity ratio (around an NTU of 1.2 in both cases) increasing the conditioner capacity has no more effect on the latent load. The outlet air temperature at this point, however, is still rising with increasing NTU, which causes the required chiller power to increase slightly, until the air temperature out of the conditioner starts to decrease, near an NTU of 4 in case a, or reaches equilibrium, as in case b.

Figures 5.3a and 5.3b show single day representative air temperatures in the system. Figures 5.4a and 5.4b show solution temperatures, and Figures 5.5a and 5.5b show humidity ratios.

The comparison of Figures 5.3a and 5.3b shows an important result of changing the solution mass flow rate. At the smaller solution mass flow rate, the solution is not able to absorb as much of the heat released by the condensation of the water vapor. The air must, therefore, absorb more energy, and it leaves the conditioner at a higher temperature than it would at a lower solution mass flow rate. In the example shown, heat exchange with ambient air would be helpful in decreasing the required auxiliary air conditioner power.

A similar phenomena is shown in Figures 5.4a and 5.4b. The solution temperature change within the conditioner and regenerator increases when the solution mass flow rate decreases. Heat exchanger #2 becomes a liability to the system when the conditioner solution outlet temperature becomes greater than the regenerator solution outlet temperature, as in Figure 5.4b.

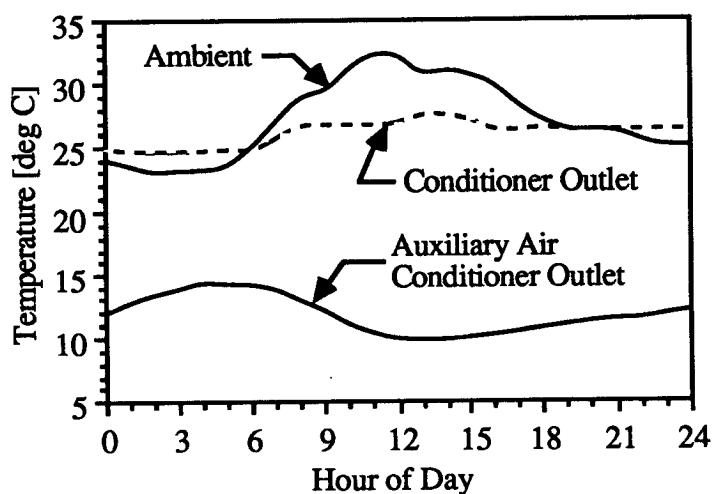


Fig. 5.3a Air Temperatures in the First System at Solution Flow Rate of 3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

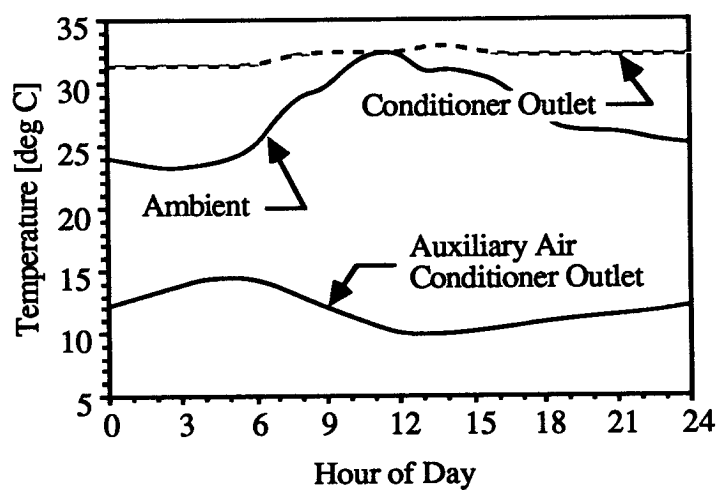


Fig. 5.3b Air Temperatures in the First System at Solution Flow Rate of 0.3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

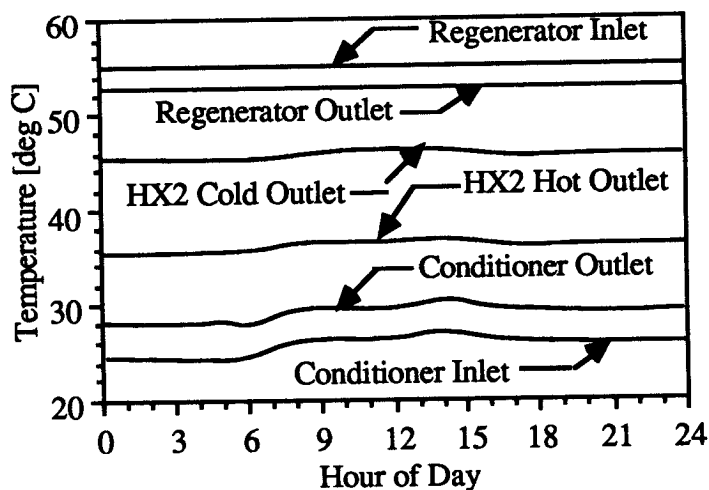


Fig. 5.4a Solution Temperatures in the First System at Solution Flow Rate of 3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

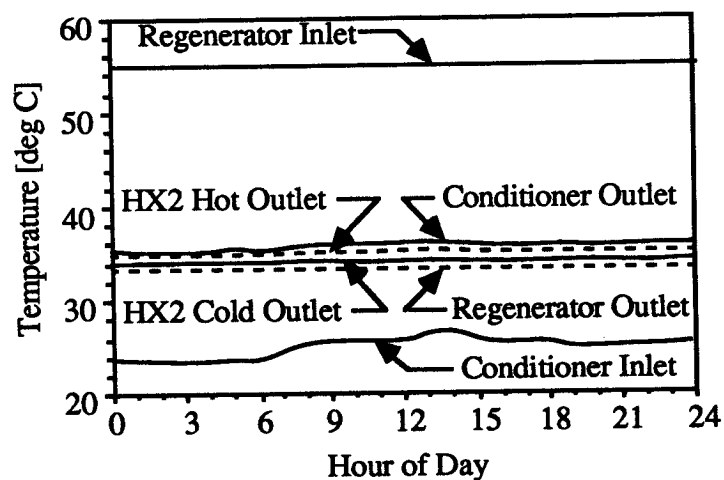


Fig. 5.4b Solution Temperatures in the First System at Solution Flow Rate of 0.3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

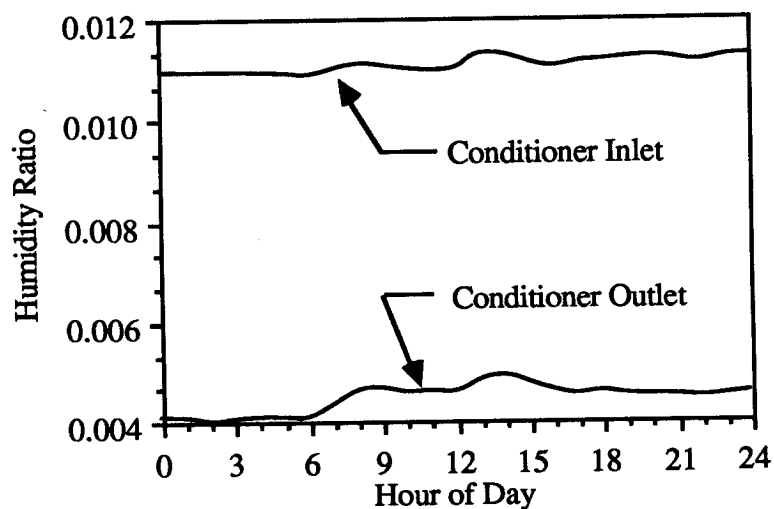


Fig. 5.5a Air Humidity Ratios in the First System at Solution Flow Rate of 3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

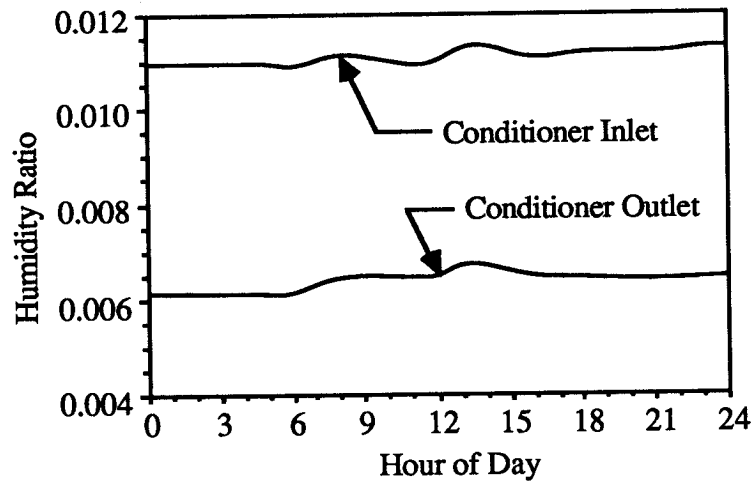


Fig. 5.5b Air Humidity Ratios in the First System at Solution Flow Rate of 0.3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

Table 5.1 shows the results of the simulations of both cases, a and b. The system COPs are based on a total load on the zone of 126,600 kW-hr. Conditioning of the ventilation air is performed, but not included in these calculations.

TABLE 5.1
Results of Simulation of First System

	Required Auxiliary Heat [kW-hr]	Required Auxiliary Cooling [kW-hr]	COP _h	COP _c	Example COP
case a ($\dot{m}_s = 3 \text{ kg/s}$)	328,500	17,710	0.385	7.15	0.30
case b ($\dot{m}_s = 0.3 \text{ kg/s}$)	71,670	27,300	1.77	4.64	0.80

In an attempt to decrease the required auxiliary cooling as well as take advantage of the overdrying capabilities of the conditioner, an evaporative cooler was added to the first system. The addition of an evaporative cooler makes no difference to the solution, so only the effects on the air will be shown. Figures 5.7a and 5.7b show the effect of changing the conditioner size on the required auxiliary chiller power for a single day for the first and second systems. Figures 5.8a and 5.8b show air temperatures in the second system, and Figures 5.9a and 5.9b show humidity ratios. These figures again represent a one day simulation using TMY data from Miami.

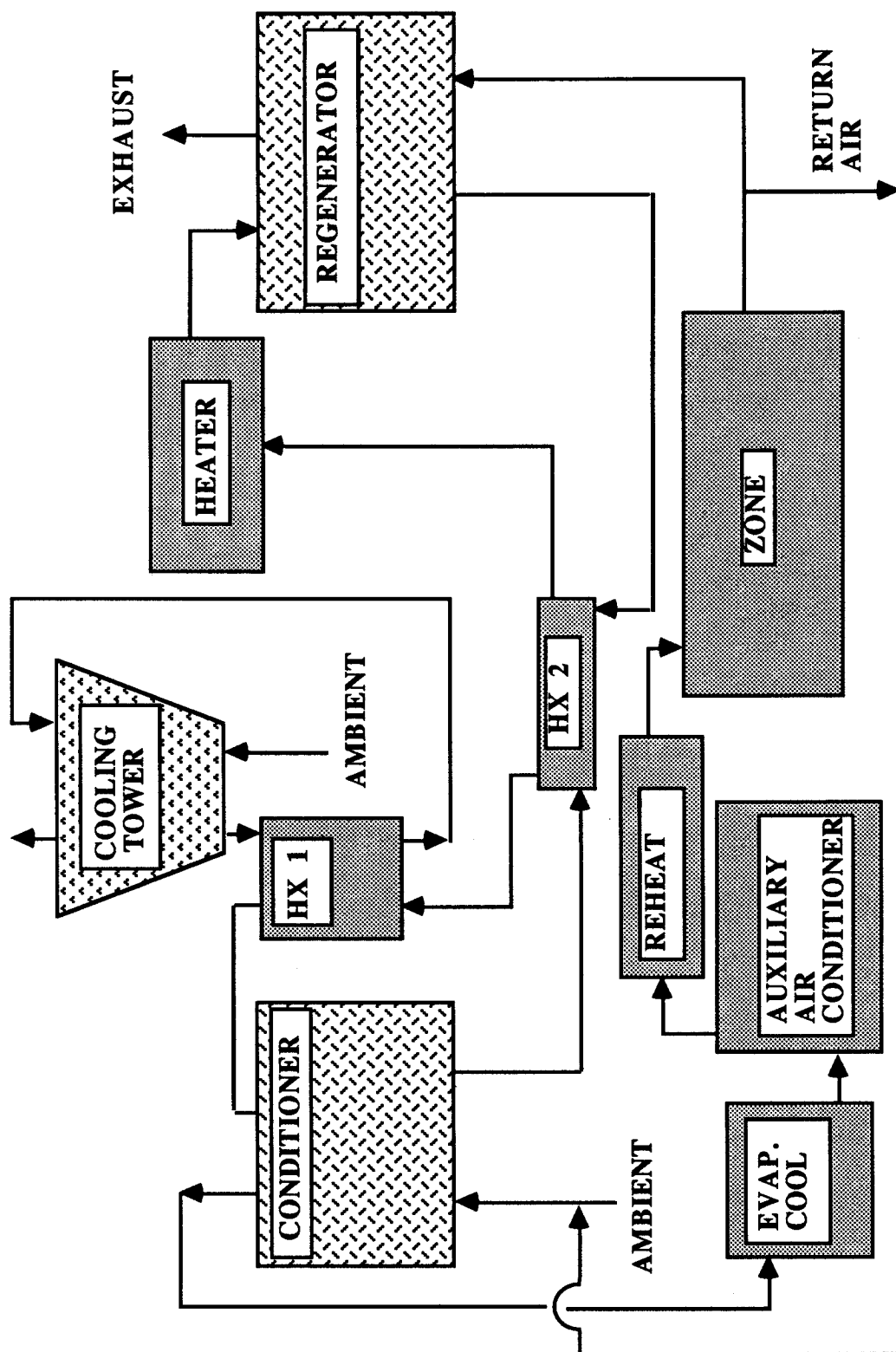


Fig. 5.6 Schematic of Second Liquid-Desiccant System Configuration

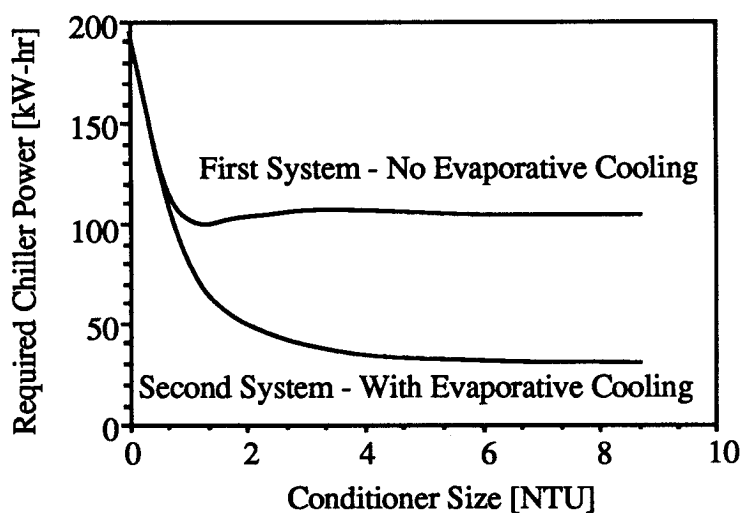


Fig. 5.7a Required Auxiliary Chiller Power vs. Conditioner Size for First and Second Systems at Solution Flow Rate of 3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

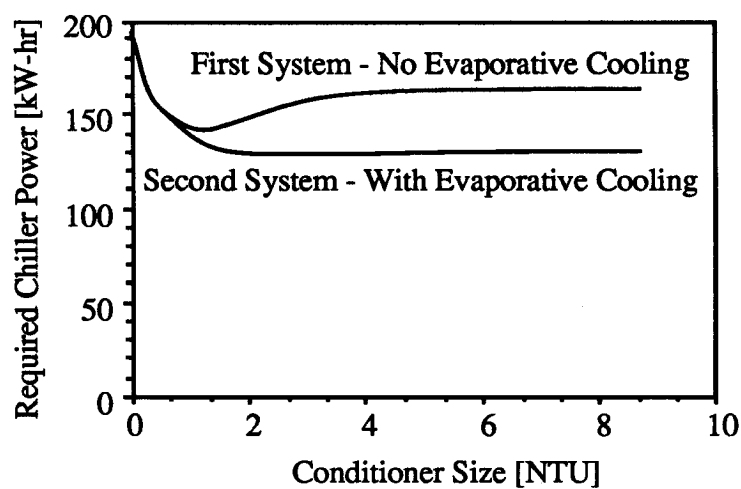


Fig. 5.7b Required Auxiliary Chiller Power vs. Conditioner Size for First and Second Systems at Solution Flow Rate of 0.3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

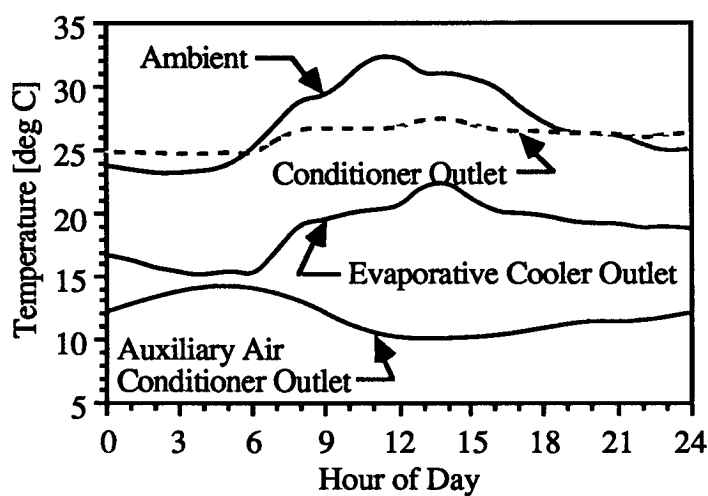


Fig. 5.8a Air Temperatures in the Second System at Solution Flow Rate of 3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

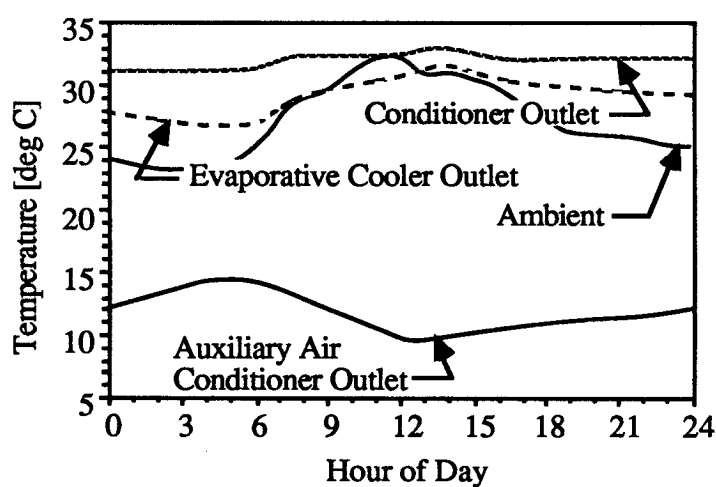


Fig. 5.8b Air Temperatures in the Second System at Solution Flow Rate of 0.3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

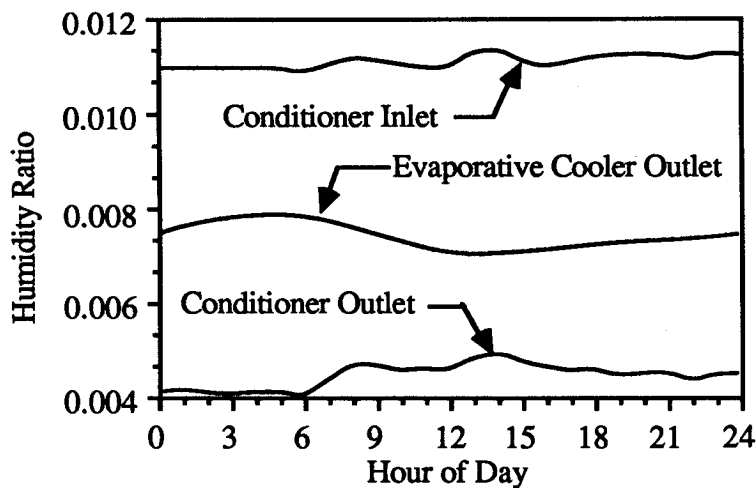


Fig. 5.9a Air Humidity Ratios in the Second System at Solution Flow Rate of 3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

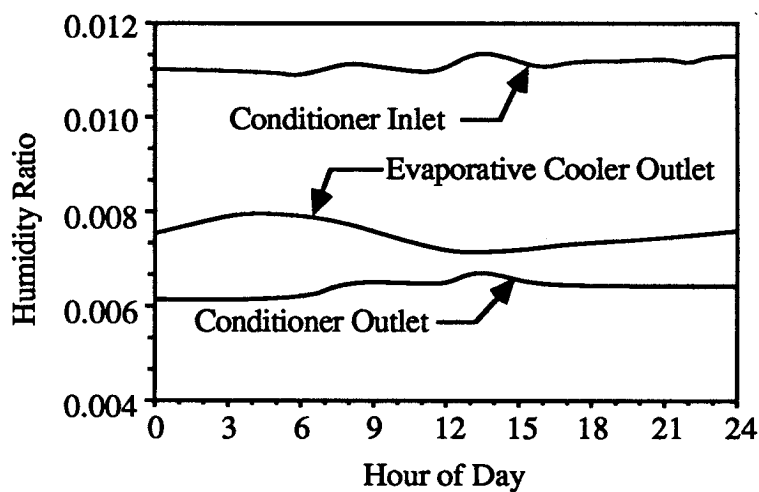


Fig. 5.9b Air Humidity Ratios in the Second System at Solution Flow Rate of 0.3 kg/s, Air Flow Rate of 3000 CFM and LiCl Concentration of 40% using July 1, Miami TMY Data

Like the first system, the second would also benefit from heat exchange between the ambient air and the conditioner outlet air. If such a heat exchanger were present, the desiccant system could meet the entire latent load without increasing the sensible load so much.

Table 5.2 shows the results of the simulations of both cases, a and b, of the second system. Again, the COPs are based on a total load on the zone of 126,600 kW-hr for the six-month summer period.

TABLE 5.2
Results of Simulation of Second System

	Required Auxiliary Heat [kW-hr]	Required Auxiliary Cooling [kW-hr]	COP _h	COP _c	Example COP
case a ($\dot{m}_s = 3 \text{ kg/s}$)	328,500	5,877	0.385	21.5	0.33
case b ($\dot{m}_s = 0.3 \text{ kg/s}$)	71,670	20,660	1.77	6.13	0.92

In order to compare the liquid-desiccant systems with traditional vapor-compression air conditioning, one other system was simulated over a six-month period. The first system (shown in Figure 5.1) was changed so that the conditioner NTU was negligibly small (6×10^{-5}). The result is the simulation of a traditional system. The total air-conditioner power required is 33,870 kW-hr, with 13,110 kW-hr of reheat energy also necessary (resulting in a COP_h of 9.66, a COP_c of 3.74 and an overall COP using the example efficiencies of 1.19).

In addition to changing the overall system configurations, it is also possible to change certain parameters of the chosen components. For example, if a larger heat exchanger is used for HX #2 the amount of auxiliary heat required will change.

From the comparison of cases a and b, it can be seen that there is a trade-off between the required amounts of chiller energy and auxiliary heat. Decreasing the solution mass flow rate decreases the heat required while increasing the required chiller energy. This effect is examined in Tables 5.3 thru 5.6. The simulations in these tables use a conditioner size ($h_D A_V V_T$) of 10 kg/s and a regenerator size of 5 kg/s. While part of the size term (h_D) is known to depend on the air and solution mass flow rates, it is assumed here that this dependency is small compared to the effect of changing the inlet mass flow rates. The cost ratio appearing in Tables 5.3 thru 5.6 is the ratio of unit costs of cooling energy to heating energy that would make the system's energy cost equal to that of a traditional vapor-compression system.

TABLE 5.3

Effect of Changing Air Flow Rate

Solution Flow Rate = 0.3 kg/s, Solution LiCl Concentration = 35%

Air Flow [CFM]	Auxiliary Heat [kW-hr]	Auxiliary Cooling [kW-hr]	COP_h	COP_c	Example COP	cost ratio
2000	70,300	24,460	1.80	5.18	0.87	6.08
3000	85,690	23,870	1.48	5.30	0.78	7.26
4000	87,040	24,610	1.46	5.14	0.76	7.98
8000	87,590	25,950	1.45	4.88	0.75	9.40

TABLE 5.4

Effect of Changing Air Flow Rate

Solution Flow Rate = 0.3 kg/s, Solution LiCl Concentration = 45%

Air Flow [CFM]	Auxiliary Heat [kW-hr]	Auxiliary Cooling [kW-hr]	COP _h	COP _c	Example COP	cost ratio
2000	43,850	23,290	2.89	5.44	1.12	2.91
3000	56,280	20,950	2.25	6.04	1.05	3.34
4000	62,890	21,710	2.01	5.83	0.97	4.09
8000	64,370	25,230	1.97	5.02	0.90	5.93

TABLE 5.5

Effect of Changing Solution Flow Rate

Air Flow Rate = 3000 CFM, Solution LiCl Concentration = 35%

Solution Flow[kg/s]	Auxiliary Heat [kW-hr]	Auxiliary Cooling [kW-hr]	COP _h	COP _c	Example COP	cost ratio
0.3	85,690	23,870	1.48	5.30	0.78	7.26
1.0	152,100	12,520	0.832	10.1	0.61	6.51
3.0	353,900	9,367	0.358	13.5	0.30	13.91

TABLE 5.6

Effect of Changing Solution Flow Rate

Air Flow Rate = 3000 CFM, Solution LiCl Concentration = 45%

Solution Flow [kg/s]	Auxiliary Heat [kW-hr]	Auxiliary Cooling [kW-hr]	COP _h	COP _c	Example COP	cost ratio
0.3	56,280	20,950	2.25	6.04	1.05	3.34
1.0	101,700	5,160	1.25	24.5	0.98	3.09
3.0	298,100	1,816	0.43	69.7	0.37	8.89

From the results shown in Tables 5.3 thru 5.6, it can be seen that the higher concentration, 45% gives better results than the lower, 35%. Unfortunately, raising the concentration any higher than about 45% is impractical because of the solution solubility limit [18]. The cost ratio shown in the tables is an indication of the applicability of the desiccant systems that were simulated. For example, if the cost of chiller electricity is 12¢/kW-hr while the cost of gas heat is 4¢/kW-hr, any system with a cost ratio less than (12/4), or 3 will show an economic benefit relative to the traditional system.

Since any source of heat at a temperature of at least 60 °C can be used to power the auxiliary heater, the greatest potential for desiccant systems is for applications where a waste heat stream or other inexpensive heat source is available. The systems presented here are not intended to represent any 'ideal' configuration, but rather to demonstrate some general performance characteristics. Upon further examination, these and other types of systems are likely to be improved considerably.

Chapter 6

CONCLUSIONS AND RECOMMENDATIONS

The objectives of this study were to develop a model of a packed bed liquid-desiccant heat and mass exchangers and to use this model to show the relative advantages of several liquid-desiccant system configurations. An effectiveness model has been developed for simulating liquid-desiccant component behavior. This model is included in the simulations of two different systems.

6.1 Conclusions

A finite-difference model was developed to verify the results of the effectiveness model. The finite-difference model is useful for looking 'into' the components; it can be used to show the states of the air and the desiccant solution at any point inside the packing material. This is useful at certain operating conditions where there is a local minimum or maximum air temperature or humidity ratio inside the component.

The effectiveness model was compared with the finite-difference model. Agreement between the two models was very close. A method of accounting for Lewis number effects in the effectiveness model was developed. Agreement with the finite-difference model on Lewis number effects was close, but not as precise as for the unity Lewis number case.

Three sets of actual liquid-desiccant data were received and compared with the effectiveness model results. In the first set of data it was impossible to determine an appropriate value of NTU from the air enthalpy change. It was shown that NTU can be determined from the change in another parameter, such as air humidity ratio. The energy balances in the first data set did not close, so the effectiveness model, with its assumption of negligible losses, did not predict the experimental outlet values very closely. When the values of NTU were chosen to make the humidity ratios match, the solution outlet temperatures came close to matching, while the air outlet temperatures differed by 6 to 18 °C.

In the second data set, air enthalpy changes were used to calculate the values of NTU. The agreement between the effectiveness model and these data was fairly close with the biggest difference being between the predicted and actual air outlet temperature. This is probably the result of energy losses from the experimental system.

The third data set was the most extensive, with 3 days of data at 15 minute intervals being the usable portion. An average value of NTU was chosen to represent the data set. The predicted and actual air outlet temperatures differed by about 1 °C, the predicted and actual air outlet humidity ratios differed by about 0.0002 kg/kg dry air. The solution temperatures agreed very closely. When a Lewis number of 1.2 was introduced into the effectiveness model, the predicted and actual values were nearly identical.

The comparison of the effectiveness model with the experimental results showed that the effectiveness model does give an accurate representation of actual liquid-desiccant component performance. The comparison with the finite-difference model showed that the effectiveness model is as accurate as a more detailed model. Together, these comparisons, the computational efficiency of the effectiveness model, and the

theoretical (as opposed to empirical) derivation of the effectiveness model prove the suitability of using the effectiveness model to simulate various configurations of liquid-desiccant systems.

The system simulations showed that the energy use of liquid-desiccant systems is highly dependent on the solution mass flow rate. The energy costs depend on the ratio of cooling to heating energy costs, as well as on the amount of energy. Higher solution concentrations are desirable, but are limited by the salt's solubility. The overall COPs of the systems were lower than that of a traditional system for the example efficiencies used. If actual heating efficiencies are higher, or cooling efficiencies lower, a liquid-desiccant system may have a higher overall COP than a traditional system. An evaporative cooler between the desiccant conditioner and the auxiliary air conditioner provides additional benefits to the system.

At low solution flow rates, further benefit could be derived from removing the solution heat exchanger between the conditioner and regenerator and inserting an air to air heat exchanger between the air leaving the conditioner and ambient air.

6.2 Recommendations

The system simulations performed in this study are intended only to show trends of the possible energy savings for three modifications of a hybrid liquid-desiccant system. Many other configurations are possible, and quantitative results are necessary to show the benefits of air conditioning with liquid desiccants.

One possible configuration that would be interesting would be the addition of a chiller to cool the solution before it enters the conditioner. It would be possible to control this chiller so that the auxiliary air conditioner would not be necessary. This method of cooling the solution could be tested with or without a cooling tower.

Because of the large amount of heating energy required in liquid-desiccant systems, modifications that would decrease the required heat will be very attractive. Alternate methods of heating the solution before it enters the regenerator could be examined. A heat pump could retrieve the energy from the exhaust air stream or from the condensor of the auxiliary air conditioner. Cogeneration could be used to provide this heat as well as the electricity for compressors, pumps and fans. It is possible that changes in the regenerator operation could reduce the heat required. Changing the air flow rate through the regenerator, or heating the air instead of the solution might result in lower heating energy requirements.

Precise controls could be added to the cooling tower and the evaporative cooler. A large solution sump could be included between the conditioner and regenerator, helping the convergence of the simulations, as well as providing a more accurate description of the actual use of liquid-desiccant systems.

In order to produce accurate results, parasitic power for the various components must be included in the simulations. The parasitics for a liquid-desiccant system are likely to make up a substantial part of the required power.

To find the overall value of the different liquid-desiccant systems, it is necessary to perform a detailed economic analysis. This requires the gathering of information about the costs of different system components and different component sizes.

Simulations of liquid-desiccant systems should be performed for different climate types. To what degree the advantages of certain configurations are location specific remains to be seen.

Hopefully, by using the new effectiveness model, it will be possible to find the best system configuration for any given load and location.

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Appendix A

Finite Difference and Effectiveness Model Results for 16 Cases

NTU	Le = 0.8			Le = 1.0			Le = 1.2			Effectiveness		
	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2
	CASE #1											
0.01	15.000	0.00109	14.971	15.000	0.00109	14.971	15	0.00109	14.971	15	0.00109	14.971
0.1	14.989	0.00182	14.725	14.987	0.00182	14.725	14.984	0.00182	14.725	14.987	0.00182	14.725
0.5	14.835	0.00434	13.905	14.806	0.00434	13.905	14.779	0.00434	13.905	14.817	0.00432	13.905
1	14.620	0.00633	13.262	14.576	0.00633	13.262	14.545	0.00632	13.262	14.614	0.00628	13.271
1.5	14.498	0.00756	12.864	14.470	0.00755	12.864	14.463	0.00755	12.864	14.521	0.00749	12.876
2	14.464	0.00833	12.606	14.468	0.00832	12.606	14.491	0.00832	12.606	14.518	0.00825	12.618
4	14.692	0.00948	12.181	14.754	0.00947	12.174	14.805	0.00947	12.168	14.764	0.00942	12.187
6	14.897	0.00969	12.082	14.934	0.00969	12.076	14.962	0.00969	12.076	14.926	0.00966	12.082
8	14.972	0.00973	12.056	14.990	0.00973	12.056	14.999	0.00974	12.056	14.984	0.00972	12.056
10	14.994	0.00974	12.049	14.999	0.00974	12.049	15.001	0.00974	12.049	15.001	0.00973	12.049
CASE #2												
0.01	15	0.00101	14.996	15	0.00101	14.996	15	0.00101	14.996	15	0.00101	14.996
0.1	14.999	0.00107	14.963	14.998	0.00107	14.963	14.998	0.00107	14.963	14.998	0.00107	14.963
0.5	14.978	0.00129	14.853	14.974	0.00129	14.853	14.97	0.00129	14.853	14.977	0.00129	14.853
1	14.951	0.00147	14.768	14.945	0.00147	14.768	14.94	0.00147	14.768	14.952	0.00147	14.768
1.5	14.936	0.00158	14.715	14.932	0.00158	14.715	14.931	0.00158	14.715	14.94	0.00157	14.715
2	14.933	0.00165	14.681	14.933	0.00165	14.681	14.936	0.00164	14.681	14.941	0.00164	14.681
4	14.964	0.00173	14.627	14.974	0.00173	14.627	14.982	0.00173	14.627	14.974	0.00173	14.627
6	14.989	0.00175	14.615	14.995	0.00175	14.615	14.998	0.00175	14.615	14.994	0.00174	14.615
8	14.997	0.00175	14.612	14.999	0.00175	14.612	15	0.00175	14.612	15.001	0.00175	14.612
10	14.999	0.00175	14.612	15	0.00175	14.612	15	0.00175	14.612	15.003	0.00175	14.612

NTU	Le = 0.8			Le = 1.0			Le = 1.2			Effectiveness		
	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2
	CASE #3											
0.01	15.159	0.00132	34.866	15.198	0.00132	34.866	15.238	0.00132	34.866	15.209	0.00132	34.866
0.1	16.493	0.00396	33.813	16.848	0.00396	33.78	17.194	0.00396	33.73	16.934	0.00396	33.76
0.5	20.913	0.01226	30.437	22.061	0.01223	30.303	23.097	0.01219	30.168	22.285	0.01222	30.266
1	24.435	0.01838	27.891	25.878	0.01831	27.712	27.088	0.01827	27.561	26.176	0.01837	27.649
1.5	26.798	0.02218	26.263	28.249	0.02213	26.082	29.377	0.02209	25.931	28.598	0.02226	25.984
2	28.507	0.02474	25.152	29.846	0.0247	24.972	30.83	0.02468	24.834	30.23	0.02491	24.846
4	32.169	0.02962	22.944	32.941	0.02965	22.817	33.422	0.0297	22.733	33.315	0.03007	22.632
6	33.648	0.03144	22.116	34.057	0.0315	22.033	34.273	0.03153	21.977	34.342	0.03189	21.853
8	34.286	0.03224	21.73	34.523	0.03233	21.682	34.627	0.03234	21.646	34.741	0.03263	21.539
10	34.621	0.03269	21.544	34.747	0.03274	21.509	34.814	0.03277	21.489	34.909	0.03295	21.404
CASE #4												
0.01	15.159	0.00106	34.929	15.199	0.00106	34.929	15.238	0.00106	34.929	15.201	0.00106	34.929
0.1	16.516	0.00159	34.399	16.875	0.00159	34.345	17.226	0.00159	34.28	16.892	0.00159	34.336
0.5	21.264	0.00332	32.6	22.462	0.00331	32.39	23.538	0.00329	32.187	22.495	0.0033	32.379
1	25.251	0.00463	31.156	26.771	0.00461	30.882	28.027	0.0046	30.644	26.803	0.0046	30.877
1.5	27.945	0.00545	30.214	29.449	0.00543	29.935	30.603	0.00543	29.717	29.479	0.00541	29.936
2	29.83	0.00598	29.566	31.192	0.00598	29.314	32.155	0.00599	29.122	31.219	0.00596	29.316
4	33.427	0.00694	28.37	34.08	0.00697	28.238	34.425	0.007	28.154	34.114	0.00693	28.245
6	34.514	0.00722	28.016	34.765	0.00724	27.957	34.876	0.00726	27.93	34.82	0.0072	27.966
8	34.849	0.00731	27.902	34.945	0.00732	27.883	34.973	0.00733	27.872	35.01	0.00728	27.887
10	34.949	0.00733	27.864	34.992	0.00735	27.864	34.995	0.00735	27.858	35.063	0.00731	27.864

NTU	Le = 0.8			Le = 1.0			Le = 1.2			Effectiveness		
	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2
	CASE #5											
0.01	15	0.01	15.001	15	0.01	15.001	15	0.01	15.001	15	0.01	15.001
0.1	15	0.00998	15.008	15	0.00998	15.008	15	0.00998	15.008	15	0.00998	15.008
0.5	15.005	0.0099	15.032	15.006	0.0099	15.032	15.007	0.0099	15.032	15.005	0.0099	15.032
1	15.011	0.00984	15.051	15.012	0.00984	15.051	15.013	0.00984	15.051	15.012	0.00984	15.051
1.5	15.014	0.0098	15.063	15.015	0.0098	15.063	15.015	0.0098	15.063	15.015	0.00981	15.063
2	15.015	0.00978	15.07	15.015	0.00978	15.07	15.014	0.00978	15.07	15.015	0.00979	15.07
4	15.009	0.00975	15.083	15.007	0.00975	15.083	15.005	0.00975	15.083	15.008	0.00975	15.083
6	15.003	0.00974	15.086	15.002	0.00974	15.086	15.001	0.00974	15.086	15.002	0.00974	15.086
8	15.001	0.00974	15.087	15	0.00974	15.087	15	0.00974	15.087	15	0.00974	15.087
10	15	0.00974	15.088	15	0.00974	15.088	15	0.00974	15.088	15	0.00974	15.088
CASE #6												
0.01	15	0.00992	15.043	15	0.00992	15.043	15	0.00992	15.043	15	0.00992	15.043
0.1	15.016	0.00922	15.403	15.019	0.00922	15.403	15.023	0.00922	15.403	15.017	0.00922	15.403
0.5	15.243	0.00677	16.618	15.288	0.00677	16.618	15.326	0.00677	16.613	15.249	0.00679	16.612
1	15.556	0.00482	17.568	15.624	0.00482	17.568	15.664	0.00482	17.558	15.533	0.00487	17.548
1.5	15.728	0.00362	18.149	15.771	0.00363	18.149	15.785	0.00363	18.149	15.661	0.0037	18.125
2	15.765	0.00289	18.519	15.762	0.00289	18.519	15.731	0.0029	18.519	15.661	0.00298	18.496
4	15.403	0.0019	19.097	15.317	0.0019	19.109	15.247	0.0019	19.122	15.288	0.00197	19.089
6	15.123	0.00177	19.218	15.072	0.00177	19.227	15.044	0.00177	19.235	15.053	0.0018	19.218
8	15.029	0.00175	19.247	15.006	0.00175	19.247	15.004	0.00175	19.253	14.975	0.00178	19.247
10	15.005	0.00175	19.254	14.999	0.00175	19.254	14.997	0.00175	19.254	14.954	0.00177	19.254

NTU	Le = 0.8			Le = 1.0			Le = 1.2			Effectiveness		
	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2
	CASE #7											
0.01	15.159	0.01023	34.896	15.198	0.01023	34.896	15.238	0.01023	34.896	15.206	0.01023	34.896
0.1	16.503	0.01213	34.083	16.861	0.01213	34.049	17.209	0.01212	33.998	16.921	0.01212	34.032
0.5	21.077	0.01802	31.483	22.253	0.01798	31.343	23.313	0.01794	31.205	22.393	0.01798	31.318
1	24.815	0.02228	29.529	26.299	0.02222	29.342	27.537	0.02217	29.186	26.458	0.02225	29.305
1.5	27.317	0.02493	28.279	28.792	0.02487	28.089	29.928	0.02484	27.936	28.961	0.02496	28.034
2	29.081	0.02672	27.421	30.42	0.02669	27.238	31.388	0.02668	27.1	30.6	0.02682	27.167
4	32.584	0.03028	25.695	33.306	0.03036	25.577	33.73	0.03041	25.496	33.495	0.03059	25.466
6	33.862	0.03173	25.032	34.208	0.03177	24.95	34.419	0.03185	24.906	34.389	0.03203	24.843
8	34.392	0.03238	24.715	34.6	0.03247	24.67	34.687	0.03248	24.636	34.737	0.03266	24.578
10	34.656	0.03273	24.554	34.767	0.03278	24.522	34.825	0.03281	24.504	34.891	0.03295	24.456
CASE #8												
0.01	15.159	0.00997	34.975	15.199	0.00997	34.975	15.238	0.00997	34.975	15.198	0.00997	34.975
0.1	16.533	0.00974	34.822	16.896	0.00974	34.766	17.25	0.00974	34.699	16.884	0.00974	34.761
0.5	21.522	0.00889	34.279	22.764	0.00887	34.059	23.878	0.00886	33.848	22.678	0.00889	34.063
1	25.844	0.00817	33.806	27.428	0.00814	33.518	28.728	0.00813	33.272	27.257	0.00817	33.532
1.5	28.737	0.00772	33.47	30.285	0.00771	33.186	31.447	0.00771	32.959	30.06	0.00775	33.2
2	30.682	0.00746	33.22	32.043	0.00747	32.963	32.979	0.00748	32.773	31.804	0.00752	32.981
4	33.948	0.00723	32.703	34.507	0.00728	32.583	34.775	0.00731	32.512	34.351	0.0073	32.595
6	34.707	0.00729	32.524	34.91	0.00732	32.486	34.972	0.00733	32.463	34.821	0.00733	32.489
8	34.915	0.00733	32.469	34.981	0.00734	32.455	34.995	0.00735	32.449	34.925	0.00735	32.457
10	34.969	0.00734	32.448	34.997	0.00735	32.448	35.004	0.00735	32.448	34.951	0.00736	32.448

NTU	Le = 0.8			Le = 1.0			Le = 1.2			Effectiveness		
	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2
	CASE #9											
0.01	34.841	0.00307	15.005	34.801	0.00307	15.005	34.761	0.00307	15.005	34.799	0.00307	15.005
0.1	33.46	0.00364	14.998	33.095	0.00364	15.037	32.738	0.00364	15.085	33.079	0.00364	15.045
0.5	28.369	0.00568	15.008	27.099	0.00569	15.166	25.956	0.00571	15.319	27.108	0.00567	15.177
1	23.903	0.00733	15.063	22.276	0.00735	15.271	20.945	0.00737	15.451	22.375	0.0073	15.279
1.5	20.917	0.00835	15.13	19.346	0.00836	15.339	18.172	0.00836	15.498	19.513	0.00829	15.342
2	18.934	0.00897	15.2	17.573	0.00895	15.38	16.659	0.00894	15.514	17.773	0.00888	15.383
4	15.784	0.00973	15.382	15.293	0.00968	15.453	15.083	0.00965	15.496	15.417	0.00965	15.453
6	15.171	0.00977	15.455	15.026	0.00974	15.47	14.992	0.00972	15.481	15.073	0.00973	15.472
8	15.045	0.00975	15.477	15.002	0.00974	15.477	14.993	0.00973	15.477	15.019	0.00974	15.477
10	15.01	0.00974	15.479	15	0.00974	15.479	14.998	0.00974	15.479	15.009	0.00974	15.479
CASE #10												
0.01	34.841	0.00299	15.045	34.801	0.00299	15.045	34.762	0.00299	15.045	34.802	0.00299	15.045
0.1	33.475	0.00288	15.364	33.113	0.00288	15.419	32.76	0.00288	15.485	33.12	0.00288	15.425
0.5	28.593	0.00254	16.488	27.369	0.00255	16.702	26.271	0.00255	16.913	27.425	0.00255	16.7
1	24.417	0.00229	17.426	22.855	0.0023	17.706	21.573	0.0023	17.952	22.98	0.0023	17.689
1.5	21.599	0.00213	18.05	20.063	0.00214	18.33	18.902	0.00214	18.555	20.235	0.00214	18.301
2	19.656	0.00203	18.478	18.29	0.00203	18.732	17.338	0.00202	18.918	18.483	0.00203	18.695
4	16.179	0.00183	19.251	15.592	0.00181	19.361	15.298	0.0018	19.419	15.735	0.00183	19.332
6	15.292	0.00177	19.451	15.098	0.00176	19.484	15.031	0.00176	19.497	15.158	0.00177	19.472
8	15.073	0.00175	19.505	15.008	0.00175	19.505	15.001	0.00175	19.51	15.029	0.00176	19.505
10	15.015	0.00175	19.512	15	0.00175	19.512	14.997	0.00175	19.512	15	0.00175	19.512

NTU	Le = 0.8			Le = 1.0			Le = 1.2			Effectiveness		
	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2
CASE #11												
0.01	35	0.0033	34.9	34.999	0.0033	34.9	34.999	0.0033	34.9	34.999	0.0033	34.9
0.1	34.964	0.0058	34.075	34.955	0.0058	34.075	34.947	0.0058	34.075	34.955	0.00579	34.075
0.5	34.448	0.0138	31.471	34.349	0.01379	31.476	34.265	0.01379	31.485	34.385	0.01377	31.482
1	33.715	0.0198	29.547	33.571	0.01979	29.563	33.475	0.01978	29.58	33.715	0.01976	29.556
1.5	33.23	0.02353	28.347	33.137	0.02351	28.368	33.108	0.02348	28.383	33.377	0.0235	28.339
2	33.009	0.026	27.542	33.004	0.02596	27.558	33.059	0.02591	27.566	33.3	0.02598	27.506
4	33.391	0.03042	25.976	33.612	0.03034	25.965	33.81	0.03029	25.952	33.898	0.03053	25.868
6	34.069	0.03187	25.39	34.271	0.03184	25.371	34.413	0.03183	25.355	34.465	0.03205	25.265
8	34.469	0.03247	25.111	34.598	0.03246	25.092	34.699	0.0325	25.084	34.763	0.03267	25.008
10	34.693	0.03278	24.969	34.792	0.03282	24.959	34.841	0.03283	24.949	34.906	0.03295	24.89
CASE #12												
0.01	35	0.00304	34.978	35	0.00304	34.978	35	0.00304	34.978	35	0.00304	34.978
0.1	34.992	0.00341	34.792	34.99	0.00341	34.792	34.988	0.00341	34.792	34.991	0.00341	34.792
0.5	34.875	0.00465	34.179	34.852	0.00465	34.179	34.832	0.00465	34.179	34.873	0.00465	34.179
1	34.715	0.00563	33.711	34.679	0.00563	33.711	34.653	0.00563	33.711	34.733	0.00561	33.711
1.5	34.619	0.00624	33.42	34.594	0.00623	33.42	34.585	0.00623	33.42	34.669	0.0062	33.42
2	34.587	0.00662	33.227	34.584	0.00661	33.227	34.597	0.00661	33.227	34.668	0.00658	33.227
4	34.737	0.0072	32.892	34.793	0.0072	32.892	34.826	0.00719	32.882	34.85	0.00716	32.892
6	34.895	0.00731	32.802	34.938	0.00732	32.802	34.952	0.00731	32.793	34.98	0.00728	32.802
8	34.966	0.00734	32.775	34.987	0.00734	32.775	34.998	0.00735	32.775	35.032	0.00731	32.775
10	34.991	0.00735	32.767	34.999	0.00735	32.767	35.002	0.00735	32.767	35.049	0.00732	32.767

NTU	Le = 0.8			Le = 1.0			Le = 1.2			Effectiveness		
	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2
	CASE #13											
0.01	34.841	0.0298	15.096	34.801	0.0298	15.096	34.762	0.0298	15.096	34.808	0.0298	15.096
0.1	33.494	0.0281	15.875	33.136	0.0281	15.906	32.787	0.0281	15.958	33.194	0.0281	15.906
0.5	28.891	0.02239	18.456	27.712	0.02241	18.608	26.647	0.02242	18.753	27.893	0.02245	18.572
1	25.098	0.018	20.469	23.598	0.01802	20.659	22.351	0.01805	20.821	23.867	0.01814	20.585
1.5	22.506	0.01528	21.734	21.008	0.0153	21.924	19.858	0.01532	22.081	21.341	0.01549	21.823
2	20.643	0.01353	22.578	19.274	0.01354	22.754	18.296	0.01354	22.891	19.642	0.01377	22.631
4	16.802	0.01063	24.056	16.125	0.0106	24.159	15.75	0.01058	24.223	16.452	0.01085	24.039
6	15.534	0.00996	24.459	15.274	0.00993	24.503	15.155	0.00991	24.521	15.465	0.01008	24.431
8	15.137	0.00979	24.565	15.053	0.00977	24.58	15.027	0.00977	24.589	15.138	0.00985	24.55
10	15.031	0.00975	24.595	15.009	0.00975	24.6	15.001	0.00974	24.6	15.031	0.00978	24.588
CASE #14												
0.01	34.841	0.02972	15.186	34.802	0.02972	15.186	34.762	0.02972	15.186	34.811	0.02972	15.186
0.1	33.527	0.02732	16.709	33.178	0.02732	16.765	32.837	0.02732	16.832	33.249	0.02732	16.756
0.5	29.4	0.01901	21.842	28.318	0.01901	22.038	27.341	0.01902	22.228	28.472	0.01908	21.98
1	26.255	0.01242	25.832	24.892	0.01244	26.073	23.747	0.01245	26.29	25.073	0.01261	25.957
1.5	24.013	0.0084	28.301	22.602	0.00841	28.554	21.479	0.00843	28.765	22.819	0.00866	28.391
2	22.215	0.0059	29.905	20.841	0.00591	30.15	19.798	0.00592	30.34	21.109	0.0062	29.955
4	17.575	0.00238	32.535	16.72	0.00237	32.701	16.196	0.00236	32.806	17.037	0.00262	32.521
6	15.735	0.00184	33.146	15.38	0.00184	33.223	15.21	0.00183	33.257	15.497	0.00198	33.13
8	15.176	0.00176	33.294	15.064	0.00176	33.32	15.024	0.00176	33.328	15.011	0.00185	33.285
10	15.036	0.00175	33.33	15.006	0.00175	33.336	14.997	0.00175	33.336	14.873	0.00183	33.325

NTU	Le = 0.8			Le = 1.0			Le = 1.2			Effectiveness		
	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2	Ta2	Wa2	Ts2
	CASE #15											
0.01	35	0.03003	34.989	35	0.03003	34.989	35	0.03003	34.989	35	0.03003	34.989
0.1	34.996	0.0303	34.902	34.995	0.0303	34.902	34.994	0.0303	34.902	34.995	0.0303	34.902
0.5	34.943	0.03115	34.632	34.932	0.03115	34.632	34.923	0.03115	34.632	34.931	0.03114	34.632
1	34.866	0.03178	34.434	34.85	0.03177	34.434	34.838	0.03177	34.434	34.85	0.03175	34.439
1.5	34.814	0.03217	34.312	34.801	0.03216	34.312	34.796	0.03215	34.312	34.81	0.03214	34.315
2	34.788	0.03242	34.231	34.785	0.03241	34.231	34.788	0.0324	34.231	34.793	0.03239	34.234
4	34.816	0.03289	34.069	34.841	0.03288	34.069	34.865	0.03287	34.069	34.84	0.03286	34.071
6	34.887	0.03306	34.005	34.907	0.03304	34.003	34.929	0.03305	34.003	34.903	0.03303	34.005
8	34.933	0.03313	33.972	34.946	0.03312	33.97	34.962	0.03313	33.97	34.942	0.03311	33.972
10	34.96	0.03317	33.954	34.968	0.03316	33.952	34.98	0.03317	33.952	34.965	0.03315	33.954
CASE #16												
0.01	35	0.02977	35.115	35.001	0.02977	35.115	35.001	0.02977	35.115	35	0.02977	35.115
0.1	35.042	0.02787	36.082	35.052	0.02787	36.082	35.062	0.02787	36.082	35.044	0.02787	36.082
0.5	35.65	0.02142	39.233	35.768	0.02142	39.225	35.866	0.02142	39.206	35.664	0.02146	39.218
1	36.499	0.01641	41.595	36.67	0.01642	41.573	36.785	0.01643	41.548	36.465	0.01653	41.549
1.5	37.019	0.01333	43.042	37.137	0.01335	43.024	37.175	0.01337	43.008	36.893	0.01353	42.975
2	37.209	0.01137	43.989	37.228	0.0114	43.984	37.168	0.01143	43.982	36.989	0.01162	43.911
4	36.475	0.00826	45.681	36.235	0.00829	45.712	36.024	0.00832	45.742	36.125	0.00857	45.599
6	35.616	0.00758	46.183	35.432	0.00759	46.217	35.309	0.00759	46.24	35.324	0.00783	46.124
8	35.209	0.00741	46.346	35.12	0.00741	46.366	35.073	0.0074	46.377	34.944	0.0076	46.311
10	35.06	0.00736	46.399	35.023	0.00736	46.406	35.011	0.00736	46.412	34.788	0.00752	46.380

Appendix B

**Additional Calculations for the Effectiveness
Model**

B 1

Rewriting Bracked Portion of Equation 3.16

B 2

Solving the Differential Equations

To algebraically simplify the bracketed portion of Equation 3.16, first add and subtract the same term (the two center terms in 3.16a):

$$\left[c_{p,m} \text{Le} (T_s - T_a) + h_{v,0} \text{Le} (\omega_{T_s, \text{sat}} - \omega_a) - h_{v,0} \text{Le} (\omega_{T_s, \text{sat}} - \omega_a) + h_{v,T_s} (\omega_{T_s, \text{sat}} - \omega_a) \right] \quad (3.16a)$$

Factor out the Lewis number and include a very small term (the last term is very small compared to the others and is subtracted here to simplify the terms in a later step):

$$\text{Le} \left[c_{p,m} (T_s - T_a) + h_{v,0} (\omega_{T_s, \text{sat}} - \omega_a) - h_{v,0} (\omega_{T_s, \text{sat}} - \omega_a) + \frac{h_{v,T_s}}{\text{Le}} (\omega_{T_s, \text{sat}} - \omega_a) - c_{p,v} T_s (\omega_{T_s, \text{sat}} - \omega_a) \right] \quad (3.16b)$$

Expand the vapor enthalpy into its components and rearrange the terms:

$$\text{Le} \left[c_{p,m} (T_s - T_a) + h_{v,0} (\omega_{T_s, \text{sat}} - \omega_a) + \frac{1}{\text{Le}} (h_{v,0} + c_{p,v} T_s) (\omega_{T_s, \text{sat}} - \omega_a) - (h_{v,0} + c_{p,v} T_s) (\omega_{T_s, \text{sat}} - \omega_a) \right] \quad (3.16c)$$

The last two terms have two factors in common:

$$\text{Le} \left[c_{p,m} (T_s - T_a) + h_{v,0} (\omega_{T_s, \text{sat}} - \omega_a) + \left(\frac{1}{\text{Le}} - 1 \right) (h_{v,0} + c_{p,v} T_s) (\omega_{T_s, \text{sat}} - \omega_a) \right] \quad (3.16d)$$

Recognize the middle factor of the third term as the enthalpy of water vapor at solution temperature:

$$Le \left[c_{p,m}(T_s - T_a) + h_{v,0}(\omega_{T_s, sat} - \omega_a) + \left(\frac{1}{Le} - 1\right) h_{v,s}(\omega_{T_s, sat} - \omega_a) \right] \quad (3.16e)$$

Multiply out and regroup the first two terms:

$$Le \left[\left(c_{p,m} T_s + h_{v,0} \omega_{T_s, sat} \right) - \left(c_{p,m} T_a + h_{v,0} \omega_a \right) + \left(\frac{1}{Le} - 1 \right) h_{v,s}(\omega_{T_s, sat} - \omega_a) \right] \quad (3.16f)$$

Recognize that the first term is defined as the air saturation enthalpy at solution temperature and that the second term is the air enthalpy. This gives the final simplification:

$$Le \left[(h_{T_s, sat} - h_a) + \left(\frac{1}{Le} - 1 \right) h_{v,s}(\omega_{T_s, sat} - \omega_a) \right] \quad (3.16g)$$

Equations 3.22 and 3.25 can be manipulated to supply the effectiveness relationship. They are repeated below for convenience.

$$\frac{dh_{T_s, \text{sat}}}{dV} = m^* \frac{dh_a}{dV} \quad 3.25$$

$$\frac{dh_a}{dV} = \frac{NTU}{V_T} (h_{T_s, \text{sat}} - h_a) \quad 3.22$$

Equation 3.25 is integrated:

$$h_{T_s, \text{sat}} = m^* h_a + C \quad B2.1$$

Boundary conditions are established:

$$V = 0: \quad h_a = h_{a,i} \quad h_{T_s, \text{sat}} = h_{T_s, \text{sat},o} \quad B2.2$$

$$V = V_T: \quad h_a = h_{a,o} \quad h_{T_s, \text{sat}} = h_{T_s, \text{sat},i} \quad B2.3$$

The boundary conditions at the air inlet are plugged into Equation B2.1:

$$h_{T_s, \text{sat},o} = m^* h_{a,i} + C \quad B2.4$$

Subtract Equation B2.4 from B2.1:

$$h_{T_s, \text{sat}} - h_{T_s, \text{sat},o} = m^* (h_a - h_{a,i}) \quad B2.5$$

Solve Equation B2.5 for the air enthalpy:

$$h_a = h_{a,i} + \frac{1}{m^*} (h_{T_s, \text{sat}} - h_{T_s, \text{sat},o}) \quad B2.6$$

Combine Equations 3.22 and 3.25:

$$\frac{dh_{T_s, \text{sat}}}{dV} = \frac{-m^* NTU}{V_T} (h_a - h_{T_s, \text{sat}}) \quad \text{B2.7}$$

Substitute B2.6 into B2.7:

$$dh_{T_s, \text{sat}} = \frac{-m^* NTU}{V_T} \left(h_{a,i} + \frac{1}{m^*} (h_{T_s, \text{sat}} - h_{T_s, \text{sat},o}) - h_{T_s, \text{sat}} \right) dV \quad \text{B2.8}$$

Separate the variables in Equation B2.8:

$$\frac{dh_{T_s, \text{sat}}}{h_{a,i} + \frac{1}{m^*} (h_{T_s, \text{sat}} - h_{T_s, \text{sat},o}) - h_{T_s, \text{sat}}} = \frac{-m^* NTU}{V_T} dV \quad \text{B2.9}$$

Rearrange the denominator of Equation B2.9:

$$\frac{dh_{T_s, \text{sat}}}{m^* h_{a,i} + (h_{T_s, \text{sat}} - h_{T_s, \text{sat},o}) - m^* h_{T_s, \text{sat}}} = \frac{-NTU}{V_T} dV \quad \text{B2.10}$$

Again:

$$\frac{dh_{T_s, \text{sat}}}{m^* h_{a,i} + (1 - m^*) h_{T_s, \text{sat}} - h_{T_s, \text{sat},o}} = \frac{-NTU}{V_T} dV \quad \text{B2.11}$$

Integrate both sides from the bottom to the top of the chamber:

$$\int_{h_{T_s, \text{sat},o}}^{h_{T_s, \text{sat},i}} \frac{dh_{T_s, \text{sat}}}{m^* h_{a,i} + (1 - m^*) h_{T_s, \text{sat}} - h_{T_s, \text{sat},o}} = \int_0^{V_T} \frac{-NTU}{V_T} dV \quad \text{B2.12}$$

Solve the integral:

$$\frac{1}{1 - m^*} \ln \left[\frac{m^* h_{a,i} + (1 - m^*) h_{T_s, \text{sat}, i} - h_{T_s, \text{sat}, o}}{m^* h_{a,i} + (1 - m^*) h_{T_s, \text{sat}, o} - h_{T_s, \text{sat}, o}} \right] = -NTU \quad \text{B2.13}$$

Multiply both sides by $(1 - m^*)$:

$$\ln \left[\frac{m^* h_{a,i} + (1 - m^*) h_{T_s, \text{sat}, i} - h_{T_s, \text{sat}, o}}{m^* h_{a,i} + (1 - m^*) h_{T_s, \text{sat}, o} - h_{T_s, \text{sat}, o}} \right] = -NTU (1 - m^*) \quad \text{B2.14}$$

Simplify the terms in the natural log:

$$\ln \left[\frac{h_{T_s, \text{sat}, i} - h_{T_s, \text{sat}, o} - m^* (h_{T_s, \text{sat}, i} - h_{a,i})}{m^* (h_{a,i} - h_{T_s, \text{sat}, o})} \right] = -NTU (1 - m^*) \quad \text{B2.15}$$

Eliminate the natural log:

$$\frac{h_{T_s, \text{sat}, i} - h_{T_s, \text{sat}, o} - m^* (h_{T_s, \text{sat}, i} - h_{a,i})}{m^* (h_{a,i} - h_{T_s, \text{sat}, o})} = e^{-NTU (1 - m^*)} \quad \text{B2.16}$$

Multiply both sides by the left-side denominator:

$$(h_{T_s, \text{sat}, i} - h_{T_s, \text{sat}, o}) - m^* (h_{T_s, \text{sat}, i} - h_{a,i}) = m^* (h_{a,i} - h_{T_s, \text{sat}, o}) e^{-NTU (1 - m^*)} \quad \text{B2.17}$$

The effectiveness is defined as the ratio of actual air enthalpy change to the maximum enthalpy change. Remember that $m^* \Delta h_a = \Delta h_s$ while C_{sat} and $c_{p,s}$ are constant:

$$\varepsilon = \frac{h_{T_s, \text{sat}, i} - h_{T_s, \text{sat}, o}}{m^* (h_{T_s, \text{sat}, i} - h_{a,i})} \quad \text{B2.18}$$

Multiply both sides by m^* , the capacitance ratio:

$$\epsilon m^* = \frac{h_{T_s, \text{sat}, i} - h_{T_s, \text{sat}, o}}{h_{T_s, \text{sat}, i} - h_{a, i}} \quad \text{B2.19}$$

Substitute Equation B2.19 into the first term of B2.17 and rearrange the exponent's coefficient:

$$\epsilon m^* (h_{T_s, \text{sat}, i} - h_{a, i}) - m^* (h_{T_s, \text{sat}, i} - h_{a, i}) = m^* \left((h_{a, i} - h_{T_s, \text{sat}, i}) - (h_{T_s, \text{sat}, o} - h_{T_s, \text{sat}, i}) \right) e^{-NTU(1 - m^*)} \quad \text{B2.20}$$

Divide each term by another term:

$$\frac{\epsilon m^* (h_{T_s, \text{sat}, i} - h_{a, i})}{m^* (h_{T_s, \text{sat}, i} - h_{a, i})} - \frac{m^* (h_{T_s, \text{sat}, i} - h_{a, i})}{m^* (h_{T_s, \text{sat}, i} - h_{a, i})} = m^* \left(\frac{(h_{a, i} - h_{T_s, \text{sat}, i}) - (h_{T_s, \text{sat}, o} - h_{T_s, \text{sat}, i})}{m^* (h_{T_s, \text{sat}, i} - h_{a, i})} \right) e^{-NTU(1 - m^*)} \quad \text{B2.21}$$

Simplify:

$$\epsilon(1) - 1 = \left(-1 - \frac{(h_{T_s, \text{sat}, o} - h_{T_s, \text{sat}, i})}{(h_{T_s, \text{sat}, i} - h_{a, i})} \right) e^{-NTU(1 - m^*)} \quad \text{B2.22}$$

Use the definition of effectiveness (Equation B2.19) to simplify B2.22:

$$\epsilon - 1 = (\epsilon m^* - 1) e^{-NTU(1 - m^*)} \quad \text{B2.23}$$

Expand Equation B2.23:

$$\varepsilon - 1 = \varepsilon m^* e^{-NTU(1-m^*)} - e^{-NTU(1-m^*)} \quad \text{B2.24}$$

Rearrange the terms:

$$\varepsilon - \varepsilon m^* e^{-NTU(1-m^*)} = 1 - e^{-NTU(1-m^*)} \quad \text{B2.25}$$

Again:

$$\varepsilon (1 - m^* e^{-NTU(1-m^*)}) = 1 - e^{-NTU(1-m^*)} \quad \text{B2.26}$$

And solve for the effectiveness:

$$\varepsilon = \frac{1 - e^{-NTU(1-m^*)}}{1 - m^* e^{-NTU(1-m^*)}} \quad \text{B2.27}$$

Appendix C

TRNSYS Component Descriptions

C1	Liquid-Desiccant Component (Conditioner or Regenerator)
C2	Auxiliary Air Conditioner
C3	Evaporative Cooler
C4	Cooling Tower
C5	Constant Data Generator

Liquid-Desiccant Component (Conditioner or Regenerator)

<u>INPUT NUMBER</u>		<u>DESCRIPTION</u>
1.	Ta1	Inlet Air Temperature [$^{\circ}\text{C}$]
2.	ω_{a1}	Inlet Air Humidity Ratio [kg/kg]
3.	Ts1	Inlet Solution Temperature [$^{\circ}\text{C}$]
4.	ma	Dry Air Mass Flow Rate [kg/s]
5.	ms1	Solution Mass Flow Rate [kg/s]
6.	ξ_1	Solution Concentration [weight fraction]

<u>PARAMETERS</u>		<u>DESCRIPTION</u>
1.	$h_D A_V V_T$	Mass Transfer Coefficient - Area Product [kg/s]
2.	Patm	Atmospheric Pressure [Pa]
3.	Tol	Tolerance on Solution Outlet Temperature [$^{\circ}\text{C}$]

<u>OUTPUTS</u>		<u>DESCRIPTION</u>
1.	Ta2	Outlet Air Temperature [$^{\circ}\text{C}$]
2.	ω_{a2}	Outlet Air Humidity Ratio [kg/kg]
3.	Ts2	Outlet Solution Temperature [$^{\circ}\text{C}$]

OUTPUTSDESCRIPTION

4.	ms2	Outlet Solution Mass Flow Rate [kg/s]
5.	ξ_2	Outlet Solution Concentration [weight fraction]

C **C**

C Subroutine TYPE7 C

C **C**

C This program is an effectiveness model of a packed bed counter flow C

C liquid desiccant heat and mass exchanger. It uses Buschulte's property C

C subroutine libraries PSAW and PSLICL. Some additional properties can C

C be found in my property subroutine library PROP. C

C C

C

C 9 FEB 88 Dawne Stevens C

C **C**

[illegible]

C The main program reads in the input values to the model. The model is

C called with an initial guess of the solution outlet temperature. The main

C program compares the model's outlet solution temperature with the guess,

C and iteration proceeds until they are the same within a specified

C tolerance.

[illegible]

C FUNCTIONS REQUIRED:

C HATAWA calculates air enthalpy (kJ/kg)

C given temperature (K) & humidity ratio (kg/kg dry air)

C	HSTSXI	calculates solution enthalpy (kJ/kg)
---	--------	--------------------------------------

C given temperature (K) and concentration (weight fract.)

C	CS	calculates saturation specific heat of air (kJ/kg-C)
---	----	--

C given two solution temperatures (C) and concentration

C HSS calculates the enthalpy of the air in equilibrium with solution
C given solution temperature (C) and concentration

C WSSE calculates the humidity ratio of air at a given enthalpy (kJ/kg)

C that is also saturated relative to a solution concentration

C TAHAWA calculates the temperature of moist air (K)
C given its enthalpy (kJ/kg) and humidity ratio

C TSHSXI calculates the temperature of solution (K)
C given its enthalpy (kJ/kg) and concentration

C note: these functions reference several others that are also located in
C the above mentioned files, PROP, PSAW, and PSLICL

[illegible]

The variables used follow a logical pattern - those suffixed with an 'A' refer to air, those suffixed with an 'S' refer to solution, those ending in '1' are inlet values, those ending in '2' are outlet values, '2G' refers to guessed outlet values. The 'T' variables are temperatures (deg C), the 'H's are enthalpies (kJ/kg), the 'M's are mass flow rates (kg/s), the 'W's are humidity ratios (kg/kg dry air), and the 'XI's are solution concentrations (weight fraction).

C Other variables include:

C	NTU	number of transfer units (dimensionless)
---	-----	--

C PATM atmospheric pressure (Pa)

C	HD	mass transfer coefficient (kg/s-m ²)
---	----	--

C	AV	surface area/unit volume of packing material (m^2/m^3)
---	----	--

C	VT	total volume of packing material (m ³)
---	----	--

C	EFFECT	calculated effectiveness (dimensionless)
1	0.0000	0.0000
2	0.0000	0.0000
3	0.0000	0.0000
4	0.0000	0.0000
5	0.0000	0.0000
6	0.0000	0.0000
7	0.0000	0.0000
8	0.0000	0.0000
9	0.0000	0.0000
10	0.0000	0.0000
11	0.0000	0.0000
12	0.0000	0.0000
13	0.0000	0.0000
14	0.0000	0.0000
15	0.0000	0.0000
16	0.0000	0.0000
17	0.0000	0.0000
18	0.0000	0.0000
19	0.0000	0.0000
20	0.0000	0.0000
21	0.0000	0.0000
22	0.0000	0.0000
23	0.0000	0.0000
24	0.0000	0.0000
25	0.0000	0.0000
26	0.0000	0.0000
27	0.0000	0.0000
28	0.0000	0.0000
29	0.0000	0.0000
30	0.0000	0.0000
31	0.0000	0.0000
32	0.0000	0.0000
33	0.0000	0.0000
34	0.0000	0.0000
35	0.0000	0.0000
36	0.0000	0.0000
37	0.0000	0.0000
38	0.0000	0.0000
39	0.0000	0.0000
40	0.0000	0.0000
41	0.0000	0.0000
42	0.0000	0.0000
43	0.0000	0.0000
44	0.0000	0.0000
45	0.0000	0.0000
46	0.0000	0.0000
47	0.0000	0.0000
48	0.0000	0.0000
49	0.0000	0.0000
50	0.0000	0.0000
51	0.0000	0.0000
52	0.0000	0.0000
53	0.0000	0.0000
54	0.0000	0.0000
55	0.0000	0.0000
56	0.0000	0.0000
57	0.0000	0.0000
58	0.0000	0.0000
59	0.0000	0.0000
60	0.0000	0.0000
61	0.0000	0.0000
62	0.0000	0.0000
63	0.0000	0.0000
64	0.0000	0.0000
65	0.0000	0.0000
66	0.0000	0.0000
67	0.0000	0.0000
68	0.0000	0.0000
69	0.0000	0.0000
70	0.0000	0.0000
71	0.0000	0.0000
72	0.0000	0.0000
73	0.0000	0.0000
74	0.0000	0.0000
75	0.0000	0.0000
76	0.0000	0.0000
77	0.0000	0.0000
78	0.0000	0.0000
79	0.0000	0.0000
80	0.0000	0.0000
81	0.0000	0.0000
82	0.0000	0.0000
83	0.0000	0.0000
84	0.0000	0.0000
85	0.0000	0.0000
86	0.0000	0.0000
87	0.0000	0.0000
88	0.0000	0.0000
89	0.0000	0.0000
90	0.0000	0.0000
91	0.0000	0.0000
92	0.0000	0.0000
93	0.0000	0.0000
94	0.0000	0.0000
95	0.0000	0.0000
96	0.0000	0.0000
97	0.0000	0.0000
98	0.0000	0.0000
99	0.0000	0.0000
100	0.0000	0.0000

C	LOF	logical variable for printing certain error messages to LUN
C	TOL	the specified tolerance that solution temperatures must meet
C	CPS	solution specific heat (kJ/kg-C)
C	CPSAT	saturation specific heat of air (kJ/kg-C)
C	MSTAR	capacitance ratio (dimensionless)

C

SUBROUTINE TYPE7(TIME,XIN,OUT,T,DTDT,PAR,INFO)

DIMENSION XIN(6),OUT(5),PAR(3),INFO(10)

REAL MA, MS, MS2, NTU

LOGICAL LOF

C The common block is used to transfer variables between TYPE7 and EFF

COMMON / EFFINPUT / MA, TA1, WA1, MS, TS1, XI1, HD,

1 N, PATM, EFFECT,

2 TA2, MS2, WA2, HA1, HS1

C If this is the first call to this subroutine, do the usual call to INFO

```
IF (INFO(7).EQ.-1) THEN
```

CALL TYPECK(1,INFO,6,3,0)

INFO(6) = 5

INFO(9) = 0

ENDIF

C Set all of the parameters and inputs to the corresponding variable names

HDAVVT = PAR(1)

PATM = PAR(2)

TOL = PAR(3)

TA1 = XIN(1)

WA1 = XIN(2)

TS1 = XIN(3)

MA = XIN(4)

MS = XIN(5)

XI1 = XIN(6)

- C Set LOF to .TRUE. if error and warning messages from the functions are to be
C printed

LOF = .FALSE.

LUN = 6

- C Guess a value for the solution outlet temperature

TS2G = 20.0

- C Calculate the inlet air and solution specific enthalpies

HA1 = HATAWA(TA1 + 273.15, WA1, LUN, LOF)

HS1 = HSTSXI(TS1 + 273.15, XI1, LUN, LOF)

NTU = HDAVVT / MA

- C Since the solution concentration doesn't change very much, use the inlet value
C as the guessed outlet

XI2 = XI1

10 CONTINUE

- C Iterate on solution outlet temperature until the guessed outlet matches the
C calculated outlet

TS2 = TS2G

CALL EFF(NTU,TS2,XI2,LUN,LOF)

IF (ABS(TS2 - TS2G).GE.TOL) THEN

```

      TS2G = TS2
      GO TO 10
END IF

```

C Return outputs

```

      OUT(1) = TA2
      OUT(2) = WA2
      OUT(3) = TS2
      OUT(4) = MS2
      OUT(5) = XI2
      RETURN
      END

```

C*****

```

      SUBROUTINE EFF(NTU,TS2,XI2,LUN,LOF)
      LOGICAL LOF
      REAL MA, MS, NTU, MSTAR, MS2
      COMMON / EFFINPUT / MA, TA1, WA1, MS, TS1, XI1, HD,
1        N, PATM, EFFECT,
2        TA2, MS2, WA2, HA1, HS1

```

C If the guessed outlet is exactly equal to the inlet temperature, their difference
C will be zero, and cause a fatal error. If this is the case, change the guessed
C outlet a little bit

```

      IF (TS1.EQ.TS2) TS2 = TS1 + 1
      HS2 = HSTSXI(TS2 + 273.15,XI2,LUN,LOF)
      CPS = (HS2 - HS1)/(TS2 - TS1)
      CPSAT = CS(TS1,TS2,XI1,PATM)
      MSTAR = (MA * CPSAT)/(MS * CPS)
      TERM = EXP(0.0 - NTU * (1.0 - MSTAR))
      EFFECT = (1 - TERM)/(1 - (MSTAR * TERM))
      ENTHA1 = HA1

```

```
HSS1 = HSS(TS1,XI1,PATM)
ENTHA2 = ENTHA1 + EFFECT * (HSS1 - ENTHA1)
ENTHASS2 = ENTHA1 + (ENTHA2 - ENTHA1)/(1-EXP(0.0-NTU))
HUMASS2 = WSSE(ENTHASS2,XI2,PATM)
WA2 = HUMASS2 + (WA1 - HUMASS2) * EXP(0.0-NTU)
MS2 = MS + MA * (WA1 - WA2)
HS2 = (HS1*MS + MA*(ENTHA1-ENTHA2))/MS2
XI2 = (MS * XI1)/MS2
TS2 = TSHSXI(HS2,XI2,LUN,LOF) - 273.15
TA2 = TAHAWA(ENTHA2,WA2,LUN,LOF) - 273.15
RETURN
END
```

Auxiliary Air Conditioner**INPUTS**

1.	Ta1	Inlet Air Temperature [°F]
2.	ωa1	Inlet Air Humidity Ratio [lb/lb]
3.	ma	Dry Air Mass Flow Rate [lb/hr]
4.	Tc	Condenser Air Temperature [°F] (set equal to ambient temperature)
5.	Tb	Temperature of Air that Bypasses Air Conditioner [°F]
6.	ωb	Humidity Ratio of Air that Bypasses Air Conditioner [lb/lb]
7.	mb	Mass Flow Rate of Air that Bypasses Air Conditioner [lb/hr]
8.	Qsens	Sensible Load on Zone [Btu/hr]
9.	Qlat	Latent Load on Zone [Btu/hr]

PARAMETERS**DESCRIPTION**

1.	AC2on	1 if separate, sensible air conditioner is also running 0 if not
2.	CAP	Rated Capacity [Btu/hr]
3.	COP	Rated COP [dimensionless]

PARAMETERS

4.	BP	Rated Bypass Factor [dimensionless]
5.	CFM	Rated Flow Rate [CFM]
6.	Tset	Zone Temperature Set Point [°F]
7.	ωset	Zone Humidity Ratio Set Point [lb/lb]

OUTPUTS

		<u>DESCRIPTION</u>
1.	Ta2	Outlet Air Temperature [°F]
2.	ωa2	Outlet Air Humidity Ratio [lb/lb]
3.	ma2	Outlet Air Mass Flow Rate [lb/hr]
4.	Pchill	Chiller Power Required [kW]
5.	Qsens	Sensible Cooling Performed [Btu/hr]
6.	Qlat	Latent Cooling Performed [Btu/hr]
7.	plr	Part Load Ratio

Evaporative CoolerINPUTS

1.	Ta1	Inlet Air Temperature [°F]
2.	ωa1	Inlet Air Humidity Ratio [lb/lb]
3.	ma	Dry Air Mass Flow Rate [lb/hr]
4.	Tb	Temperature of Air that Bypasses Air Conditioner [°F]
5.	ωb	Humidity Ratio of Air that Bypasses Air Conditioner [lb/lb]
6.	mb	Mass Flow Rate of Air that Bypasses Air Conditioner [lb/hr]
7.	Qsens	Sensible Load on Zone [Btu/hr]
8.	Qlat	Latent Load on Zone [Btu/hr]

PARAMETERSDESCRIPTION

1.	Tset	Zone Temperature Set Point [°F]
2.	ωset	Zone Humidity Ratio Set Point [lb/lb]
3.	F	Fraction of Available Cooling to be Used
4.	Tw	Temperature of Make-Up Water [°F]

OUTPUTS**DESCRIPTION**

- | | | |
|----|-------------|--|
| 1. | Ta2 | Outlet Air Temperature [°F] |
| 2. | $\omega a2$ | Outlet Air Humidity Ratio [lb/lb] |
| 3. | ma2 | Outlet Dry Air Mass Flow Rate
[lb/hr] |
| 4. | mw | Mass Flow Rate of Make-Up
Water [lb/hr] |

```

subroutine type13(time,xin,out,t,dtdt,par,info)
c*****
c
c   This subroutine simulates the performance of an evaporative cooling unit.
c
c   All quantities are to be expressed in English units.
c
c   (DIS--March 1988)
c*****

implicit none                                !disallow default variable definitions

c   define the standard TRNSYS variables:
real    xin(8),out(4),par(4),dtdt(1),t,time
integer info(10)

c   define variables needed for this routine:

real    hwliq                                !function that calculates water enthalpy
real    fract                                !fraction of humidity difference to make-up
real    Tset                                  !store temperature set point (deg F)
real    Wset                                  !store humidity ratio set point
real    Tin                                   !dry bulb temperature of incoming air
real    Win                                   !humidity ratio of incoming air
real    Hin                                   !enthalpy of incoming air
real    mdot1                                 !mass flow-rate of incoming air (lb/hr)
real    Tret                                  !dry-bulb temp. of return air
real    Wret                                  !humidity ratio of return air
real    mdot2                                 !mass flow-rate of air bypassing ac1
real    Qsens                                 !store sensible load, neglecting ventil.
!                                             !      (Btu/hr)
real    Qlat                                  !store latent load, neglecting ventil.

```



```

real    mdottot      !total circulation mass flow-rate:
                        ! (mdot1 + mdot2)

real    Texit        !temperature of exiting air
real    Wexit        !humidity ratio of exiting air
real    Hexit        !enthalpy of exiting air
real    Tdes         !desired inlet temperature of zone air
real    Wdes         !desired inlet humidity ratio of zone air
real    Wdes2        !saturated air at inlet enthalpy
real    Wsat         !saturation humidity at Tdes
real    TW           !make-up water temperature (deg F)
real    TWK          !make-up water temperature (deg K)
real    mdotW        !make-up water flow rate (lb/hr)
real    hW           !enthalpy of make-up water
integer LUN          !logical unit number for output
logical LOF          !=.true. if output desired

```

c define variables necessary for psych subroutine:

```

parameter    iunits=2      !unit flag to psych subroutine: 2=English
integer      wbmode        !wet bulb mode flag to psych subroutine:
                        !  1--calculate Twb
                        !  0--don't calculate Twb

integer mode              !mode flag to psych subroutine:
                        !  1--Tdb and Twb input
                        !  2--Tdb and rh input
                        !  4--Tdb and W input

real    psydat(9)        !array of psychometric data:
                        !  (1)--Patm (atm)
                        !  (2)--Tdb (F)
                        !  (3)--Twb (F)
                        !  (4)--relative humidity fraction
                        !  (5)--Tdp (F)
                        !  (6)--humidity ratio (lbm water/
                        !      lbm dry air

```

```

! (7)--enthalpy (Btu/lbm dry air)
! (8)--dens. of air-water mixture
! (lbm/ft**3)
! (9)--density of air in mixture
! (lbm/ft**3)

```

LOF = .false.

```

c  if this is the first call to this unit, do the usual TRNSYS stuff:
    if (info(7) .eq. -1) then
        call typeck(1,info,8,4,0)    !check no. of inputs, parameters
        info(6) = 4                  !four outputs
        info(9) = 0                  !outputs depend only on inputs, not time
    endif

c  get parameters, inputs from arrays:
    Tset = par(1)
    Wset = par(2)
    fract = par(3)
    TW = par(4)
    Tin = xin(1)
    Win = xin(2)
    mdot1 = xin(3)
    Tret = xin(4)
    Wret = xin(5)
    mdot2 = xin(6)
    Qsens = xin(7)
    Qlat = xin(8)

c  call psych to get complete information on incoming air:
    psydat(1) = 1
    psydat(2) = Tin
    psydat(6) = Win

```

```

mode = 4
wbmode = 1
call psych(iunits,mode,wbmode,psydat)
Win = psydat(6)           !this is a fix in case the incoming
                           ! humidity is physically impossible--
                           ! at least one of the TMY weather files
                           ! has mistakes
Hin = psydat(7)           !this is the incoming air enthalpy

c    convert water make-up temperature to deg K for calculating enthalpy,
c    calculate total mass flow, as this quantity will often be used:
TWK = (TW + 460) / 1.8
mdottot = mdot1 + mdot2

c    calculate desired exit state :
Wdes = ( (Wset - Qlat/(mdottot * 1061.0)) * mdottot
+        - mdot2 * Wret ) / mdot1
Tdes = ( (Tset - Qsens/(mdottot * 0.24)) * mdottot
+        - mdot2 * Tret ) / mdot1

c    if humidity is greater than desired humidity, we don't want to use
c    evaporative cooling

if (Win .gt. Wdes) then

    mdotW = 0

c    put out output

    out(1) = Texit
    out(2) = Wexit
    out(3) = mdot1
    out(4) = mdotW

```

```
return
```

```
endif
```

- c compare Wdes with saturation humidity at Tdes. If Wdes is greater,
- c then evaporatively cool to saturation or to Wdes, whichever is lower

```
psydat(1) = 1
```

```
psydat(2) = Tdes
```

```
psydat(4) = 0.99
```

```
mode = 2
```

```
wbmode = 0
```

```
call psych(iunits,mode,wbmode,psydat)
```

```
Wsat = psydat(6)
```

```
if (Wdes.gt.Wsat) then
```

```
    psydat(6) = Win
```

```
    psydat(7) = Hin
```

```
    wbmode = 1
```

```
    mode = 6
```

```
    call psych(iunits,mode,wbmode,psydat)
```

```
    psydat(2) = psydat(3)
```

```
    mode = 1
```

```
    call psych(iunits,mode,wbmode,psydat)
```

```
    Wdes2 = psydat(6)
```

```
    if (Wdes2.le.Wdes) Wdes = Wdes2
```

```
endif
```

- c set exit state to input state, with the expectation that most of the
- c the time the exit conditions will be reassigned:

```
Texit = Tin
```

```
Wexit = Win
```

- c if humidity is less than desired humidity, use evaporative cooling
- c to make up a fraction of the difference

$\text{mdotW} = \text{fract} * \text{mdot1} * (\text{Wdes} - \text{Win})$

$\text{hW} = \text{hwliq}(\text{TWK}, \text{LUN}, \text{LOF}) * 0.43$!convert from kJ/kg to Btu/lb

$\text{Wexit} = \text{Win} + \text{mdotW} / \text{mdot1}$

$\text{Hexit} = (\text{mdotW} * \text{hW} + \text{mdot1} * \text{Hin}) / \text{mdot1}$

- c call psych to get complete information on exiting air:

$\text{psydat}(1) = 1$

$\text{psydat}(6) = \text{Wexit}$

$\text{psydat}(7) = \text{Hexit}$

$\text{mode} = 6$

$\text{wbmode} = 1$

call psych(iunits,mode,wbmode,psydat)

$\text{Texit} = \text{psydat}(2)$

if ($\text{Texit} < \text{Tdes}$) then

write (6,*) 'Texit less than Tdes in evaporative cooler'

endif

- c put out output

$\text{out}(1) = \text{Texit}$

$\text{out}(2) = \text{Wexit}$

$\text{out}(3) = \text{mdot1}$

$\text{out}(4) = \text{mdotW}$

return

end

Cooling Tower**INPUTS**

1.	Tw1	Inlet Water Temperature [°C]
2.	Mw1	Inlet Water Mass Flow Rate [kg/hr]
3.	Tdb1	Inlet Air Dry Bulb Temperature [°C]
4.	Twb1	Inlet Air Wet Bulb Temperature [°C]
5.	Tw	Temperature of Make-Up Water [°C]
6.	γ	Fan Speed Control Function [Fraction of Maximum]

PARAMETERS**DESCRIPTION**

1.	Units	1 = SI, 2 = English
2.	Mode	1 = Enter Coefficients
3.	Geometry	1 = Counterflow
4.	N	# of Cells
5.	Vmax	Maximum Air Flow per Cell [m ³ /hr]
6.	Pmax	Maximum Fan Power [kW]
7.	Fanoff	Used if Air Flows Naturally

PARAMETERS

8.	V	Sump Volume [m^3]
9.	Tsump	Initial Sump Temperature [$^{\circ}\text{C}$]
10.	c	NTU Relationship Coefficient
11.	n	NTU Relationship Exponent

OUTPUTSDESCRIPTION

1.	TS	Sump Temperature, Outlet Water Temperature [$^{\circ}\text{C}$]
2.	mw2	Outlet Water Mass Flow Rate [kg/hr]
3.	Pfan	Fan Power Required [kW]
4.	Qcells	Total Heat Rejection from Cells [kJ/hr]
5.	Tw	Temperature of Water into Sump [$^{\circ}\text{C}$]
6.	mloss	Water Loss Rate [kg/hr]
7.	Tdb2	Outlet Air Dry Bulb Temperature [$^{\circ}\text{C}$]
8.	Twb2	Outlet Air Wet Bulb Temperature [$^{\circ}\text{C}$]
9.	ω_{a2}	Outlet Air Humidity Ratio [kg/kg]

OUTPUTSDESCRIPTION

10.	ma2	Outlet Dry Air Mass Flow Rate [kg/hr]
11.	U	Internal Energy Change of Sump since Beginning of Simulation [kJ]

Constant Data Generator**PARAMETERS**

1.	CFM	Volumetric Air Flow Rate to Zone [CFM]
2.	Tset	Zone Temperature Set Point [°C]
3.	ω set	Zone Humidity Ratio Set Point [kg/kg]
4.	Patm	Atmospheric Pressure [Atm]

OUTPUTS**DESCRIPTION**

1.	CFM	Volumetric Air Flow Rate to Zone [CFM]
2.	ma	Air Mass Flow Rate to Zone [lb/hr]
3.	ma	Air Mass Flow Rate to Zone [kg/s]

Appendix D

Sample TRNSYS Deck

NOLIST

*This is a deck for a simple Liquid-Desiccant system with
*evaporativecooling

*

DeckC

*31 MAR 88

*

SIM 2160 6551 1.0

*

TOL 0.001 0.001

*

LIM 100 100 90

*

CON 15

QLA = 33000, QSE = 67000, M2 = 0, CFM = 3000, TWA = 30, RFS = 1,
TSE = 24, TSF = 75.2, WSE = 0.01, T65 = 65, PAT = 1.0,
PPA = 1.0135E+05, XIR = 0.45, MRE = 0.1, GAM = 0.15

*

*

UNI 1 TYP 42 Cooling Tower

*

PAR 11

1 1 1 1 13400 2.5 0 0 25 3 -0.5

*units mode geom #cell maxflow Pmax fanoff sumpvol sumptemp c n

*

INP 6

4,3 13,1 9,1 11,2 0,0 0,0

41.5 6700 23.9 22.2 TWA RFS

*Twi Mwi Tai Taiwb TWater relfanspd

*

*

UNI 2 TYP 7 Conditioner

*

PAR 3

10 PPA 1E-04

*HdAvVt Patm Tol

*

INP 6

19,1 19,2 4,1 19,3 4,2 3,5

23.9 0.0164 26.1 1.72 30.4

*Tai Wai TsiMa Ms Xli

*

*

UNI 3 TYP 7 Regenerator

*

PAR 3

5 PPA 1E-04

*HdAvVt Patm Tol

*

INP 6

18,4 18,5 7,1 18,6 0,0 0,0

24 0.01 55 1.72 MRE XIR

*Tai Wai Tsi Ma Ms Xli

*

*

UNI 4 TYP 5 Heat Exchanger #1

*

PAR 4

4 0.9 2.6 4.2

*mode eff Cph Cpc

*

INP 4

6,1 6,2 1,1 14,1

35 3 26.1 1.86

*Thi Mh Tci Mc

*

*

UNI 5 TYP 30 Generates Constant Data

*

PAR 4

CFM TSE WSE PAT

*

*

UNI 6 TYP 5 Heat Exchanger #2

*

PAR 4

4 0.7 2.6 2.6

*mode eff Cph Cpc

*

INP 4

3,3 3,4 2,3 2,4

35 3 25 3

*Thi Mh Tci Mc

*

*

UNI 7 TYP 6 Auxiliary Heater

*

PAR 3

150 55 2.6

*Qmax Tset Cpf

*

INP 3

6,3 6,4 12,1

35 3 1.0

*Ti Mi on/off

*

*

UNI 8 TYP 28 Simulation Summary

LAB 6

Pchill PcoolT Pheat Ptot QHX1 QHX2

*

INP 5

10,4 1,3 7,3 4,5 6,5

*Pchill Pfan Qheat Qhx1 Qhx2

*

PAR 18

48 2160 8076 0 2 0,-3 0,-3 0,-3 3 3 -4 0,-4 0,-4

*interval Ton Toff LUN mode

*

*

UNI 9 TYP 9 Data Reader (TMY Data)

*

PAR 10

2 1 -1 0.1 0 -2 0.0001 0 11 1

(T20,F4.0,T25,F6.0)

*#val interval LUN fmt?

*format

*

*

UNI 10 TYP 11 Auxiliary Air Conditioner

*

PAR 7

0 150000 4 0.05 5000 TSF WSE

*ac2on cap COP bypass CFM Tset Wset

*

INP 9

29,1 29,2 29,3 15,1 16,1 2,2 0,0 20,1 20,2

94 0.005 13650 75 94 0.005 M2 QSE QLA

*Tin Win Lb/hr Tcond Tmix Wmix Mmix Qsens Qlat

*

*

UNI 11 TYP 33 Psychrometrics

*

PAR 3

4 1 PAT

*mode units Patm

*

INP 2

9,1 9,2

23.9 0.0164

*Tdb W

*

*

UNI 12 TYP 2 On/off Controller (Aux Heat)

*

PAR 3

3 12 10

*NSTK DTh DTI

*

INP 3

0,0 6,3 12,1

T65 35 1

*Th TI control

*

*

UNI 13 TYP 15 Algebra kg/s - kg/hr

*

PAR 5

0 -1 3600 1 -4

*

INP 1

4,4

1

*

*

UNI 14 TYP 15 Algebra kg/hr - kg/s

*

PAR 5

0 -1 3600 2 -4

*

INP 1

1,2

6700

```

*
*****
*
UNI 15  TYP 15  Algebra deg C - deg F
*
PAR 8
-1  1.8  0  1  -1  32  3  -4
*
INP 1
9,1
25
*
*****
*
UNI 16  TYP 15  Algebra deg C - deg F
*
PAR 8
-1  1.8  0  1  -1  32  3  -4
*
INP 1
2,1
25
*
*****
*
UNI 17  TYP 14  Time Dependent Load Generator
*
PAR 26
0,1.0  2,0.9  4,0.82  6,0.82  8,0.95  10,1.1  12,1.2
14,1.2  16,1.15  18,1.1  20,1.08  22,1.05  24,1.0
*t0,v0  ti,vi
*
*****
*
UNI 18  TYP 12  Mixer #1
*
PAR 1

```



```

7
*mode
*
INP 4
0,0 0,0 5,3 0,0
TSE WSE 1.72 GAM
*Tin Win kg/s gamma
*
*****
*
UNI 19 TYP 12 Mixer #2
*
PAR 1
6
*mode
*
INP 6
9,1 9,2 18,6 18,1 18,2 18,3
24 0.0164 0.27 24 0.010 1.45
*T1 W1 M1 T2 W2 M2
*
*****
*
UNI 20 TYP 15 Algebra - Calculates Qsens,Qlat
*
PAR 10
0 -1 QSE 1 -4 -11 -1 QLA 1 -4
*inp par # * out inp par # * out
*
INP 1
17,1
1.0
*Load fraction
*
*****
*
UNI 21 TYP 15 Algebra - AC outputs to SI (T,Qsens,Qlat)
*

```

PAR 18

0 -1 32 4 -1 1.8 2 -4 0 -1 3413 2 -4 0 -1 3413 2 -4

*inp par # - par # / out inp par # / out inp par # / out

*

INP 3

10,1 10,5 10,6

59 5300 0

*Tac_out Qsens_ac Qlat_ac

*

*

UNI 22 TYP 15 Algebra - Calculates Reheat

*

* first output--leaving temperature [deg F]

* Tset-Qsens/mdotcirc*0.24 or Tcold--whichever greater

* second output Qreheat (kW)

* (out1 - Tcold) * 0.24 * Mdotcirc

PAR 22

0 0 0 2 -1 0.24 2 4 0 12 -3 -14 4 -1 0.24 1 -13 1 -1 3413 2 -4

*

INP 4

0,0 20,1 5,2 10,1

TSF QSE 13650 59

* Tset Qsens mdotcirc Tcold

*

*

UNI 23 TYP 25 Printer - Air Temperatures

*

PAR 4

48 0 9000 8

*Tpr Ton Toff LUN

*

INP 5

9,1 2,1 21,1 30,1 24,1

tamb ta2c ta2a ta2ec ta2rh

*

```

*****
*
UNI 24 TYP 15 Algebra - deg F to deg C
*
PAR 8
0 -1 32 4 -1 1.8 2 -4
*inp par # - par # / out
*
INP 1
22,1
58
*Ta2_rh
*
*****
*
UNI 25 TYP 25 Printer - Solution Conditions
*
PAR 4
48 0 9000 9
*Tpr Ton Toff LUN
*
INP 6
2,3 6,3 7,1 3,3 6,1 4,1
Ts2C TcoHX2 ToAux Ts2R ThoHX2 ThoHX1
*
*****
*
UNI 26 TYP 25 Printer - System Parameters
*
PAR 4
48 0 9000 10
*Tpr Ton Toff LUN
*
INP 2
10,7 2,5
PLR XI2C
*

```

*

UNI 27 TYP 28 Sim Sum - Qlat,Qsens,Qrh

*

PAR 11

48 0 9000 12 2 0,-4 0,-4 0,-4

*Tpr Ton Toff LUN mode i/o i/o i/o

*

INP 3

10,6 10,5 22,2

*Qlat Qsen QRH

*

*

UNI 28 TYP 25 Printer - Humidity Ratios

*

PAR 4

48 0 9000 13

*Tpr Ton Toff LUN

*

INP 4

19,2 2,2 10,2 29,2

Wa1 Wa2 Wa3 Wa4

*

*

UNI 29 TYP 13 Evaporative cooler

*

PAR 4

TSF WSE 0.95 TWA

*Tset Wset fract Tmake-up

*

INP 8

16,1 2,2 5,2 16,1 2,2 0,0 20,1 20,2

94 0.005 13650 94 0.005 M2 QSE QLA

*Tin Win Lb/hr Tmix Wmix Mmix Qsens Qlat

*

```
*
UNI 30 TYP 15 Algebra - Deg F to deg C
*
PAR 8
0 -1 32 4 -1 1.8 2 -4
*inp par # - par # / out
*
INP 1
29,1
70
*Temperature out of evaporative cooler
*
*****
END
*****
```

Appendix E

FORTRAN Code - Property Subroutines

```

*****
c
c This set of functions can be used with Buschulte's property subroutine libraries for
c either lithium chloride or lithium bromide solution
c
*****
c
c Function WSS finds the air humidity ratio in equilibrium with solution
c at temperature T (deg C) and concentration X (wt. fract)
c
    function WSS(T,X,Patm)
    logical LOF
    LUN = 6
    TK = T + 273.15
    P = pwtsxi(TK,X,LUN,LOF)
    WSS = wapw(P,Patm,LUN,LOF)
    return
    end
*****
c
c Function CP calculates the specific heat of solution at temperature T
c (deg C) and concentration X (wt. fract.)
c
    function CP(T,X)
    logical LOF
    LUN = 6
    TK = T + 273.15
    H1 = hstsxi(TK - 1,X,LUN,LOF)
    H2 = hstsxi(TK + 1,X,LUN,LOF)
    CP = (H2 - H1) / 2.0
    return
    end
*****
c
c Function HSS calculates the enthalpy of air that is in equilibrium with
c solution at temperature T (deg C) and concentration X (wt. fract.)
c

```

```

function HSS(T,X,Patm)
logical LOF
LUN = 6
TK = T + 273.15
W = WSS(T,X,Patm)
HSS = hatawa(TK,W,LUN,LOF)
return
end

```

c*****

```

c
c Function CS calculates the saturation specific heat of air. T1 and T2
c correspond to the inlet and outlet conditions of the LiCl solution (deg C)
c and X is the concentration (wt. fract.)
c

```

```

function CS(T1,T2,X,Patm)
H1 = HSS(T1,X,Patm)
H2 = HSS(T2,X,Patm)
CS = (H2 - H1)/(T2 - T1)
return
end

```

c*****

```

c
c Function WSSE calculates the effective humidity of a air/solution mixture.
c WSSE is the humidity ratio that is in equilibrium with solution at a given
c concentration. It is determined from the known enthalpy at this condition.
c

```

```

function WSSE(H,XI,Patm)
logical LOF
LOF = .true.
if (h.gt.150) then
relax = 0.3
else
relax = 0.8
end if
LUN = 6
WTOL = 1e-05
WSOL = 1.0

```



```

WAIR = 0.0
TG = 20.0
do while (abs(WSOL - WAIR).ge.WTOL)
  TGK = TG + 273.15
  WSSP = WSS(TG + 1,XI,Patm)
  WSOL = WSS(TG,XI,Patm)
  SOLSLOPE = WSSP - WSOL
  HUMP = wahata(H,TGK + 1,LUN,LOF)
  WAIR = wahata(H,TGK,LUN,LOF)
  AIRSLOPE = HUMP - WAIR
C      write (9,*) tg,wsol,wair,solslope,airslope
  Wdiff = WSOL - WAIR
  SLOPEdif = AIRSLOPE - SOLSLOPE
  if (slopedif.eq.0.0) then
    if (abs(WSOL - WAIR).le.WTOL) then
      WSSE = WSOL
      return
    else
      write (6,*) ' function wsse '
      write (6,*) ' h = ',h,' xi = ',xi,' Patm = ',Patm
      write (6,*) ' wsol = ',wsol,' wair = ',wair
      write (6,*) ' airslope = ',airslope,' solslope = ',solslope
      write (6,*) ' Wdiff = ',Wdiff,' SLOPEdif = ',SLOPEdif
      write (6,*) ' error in function WSSE'
      stop
    end if
  end if
  TG = TG + relax*(WSOL-WAIR)/(AIRSLOPE-SOLSLOPE)
end do
WSSE = WSOL
return
end
C*****

```

Appendix F

FORTRAN Code - Finite Difference and Effectiveness Model Comparison

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C                                                                 C
C   This program runs a comparison of the effectiveness model with the   C
C   finite difference model for liquid desiccant chambers.                 C
C                                                                 C
C   The finite difference model originates from a paper by Factor and     C
C   Grossman, though it has seen many revisions. The effectiveness model  C
C   is from a paper by Jim Braun devoted to the modeling of cooling towers. C
C   Of course, some changes were necessary in order to accomodate a salt   C
C   solution instead of water, but the essence of the equations remains the C
C   same.                                                                 C
C                                                                 C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
REAL MA, MS, MS2, MS2G, LAMBDA, MDOTSALT, MDOTS, NTU, MSTAR
REAL MS2FD, MS2A, MS2B, MS2C, MS2D, MS2EF, Le
LOGICAL LOF, iflag
COMMON / INPUT / MA, TA1, WA1, MS, TS1, XI1, HD,
1      S, DZ, MDOTSALT, N, PATM, Le, Ts2,
2      TA2, MS2, XI2, NTU, WA2, HA1, HS1
LOF = .FALSE.
  iflag = .false.
LUN = 6
TOLX = 0.0001
TOLT = 0.001
WRITE (6,*) ' ENTER RELAXATION COEFFICIENT '
READ (6,*) RELAX
READ (8,*) M,ZMAX,AV,VT,N,PATM
DO 15 K=1,M
  READ (8,*) MA,TA1,TA2G,WA1,WA2G,MS,TS1,TS2G,XI1,HD
  HA1 = HATAWA(TA1 + 273.15, WA1, LUN, LOF)
  HS1 = HSTSXI(TS1 + 273.15, XI1, LUN, LOF)
  XI2G = XI1
  Le = 0.8
  NTU = HD * AV * VT * Le / MA
10  CALL EFF(TS2g,XI2g)
    if (abs(ts2-ts2g).ge.tolT) then
      ts2g = ts2

```

```

        iflag = .true.
    end if
        if (abs(xi2-xi2g).ge.tolX) then
            xi2g = xi2
            iflag = .true.
        end if
        if (iflag) then
            iflag = .false.
            go to 10
        end if
        TS2EF_8 = TS2
        TA2EF_8 = TA2
        WA2EF_8 = WA2
        XI2EF_8 = XI2
        MS2EF_8 = MS2
        Le = 1.0
        NTU = HD * AV * VT * Le / MA
20    CALL EFF(TS2g,XI2g)
        if (abs(ts2-ts2g).ge.tolT) then
            ts2g = ts2
            iflag = .true.
        end if
        if (abs(xi2-xi2g).ge.tolX) then
            xi2g = xi2
            iflag = .true.
        end if
        if (iflag) then
            iflag = .false.
            go to 20
        end if
        TS2EF = TS2
        TA2EF = TA2
        WA2EF = WA2
        XI2EF = XI2
        MS2EF = MS2
        Le = 1.2
        NTU = HD * AV * VT * Le / MA

```

```

30  CALL EFF(TS2g,XI2g)
      if (abs(ts2-ts2g).ge.tolT) then
        ts2g = ts2
        iflag = .true.
      end if
      if (abs(xi2-xi2g).ge.tolX) then
        xi2g = xi2
        iflag = .true.
      end if
      if (iflag) then
        iflag = .false.
        go to 30
      end if
      TS2EF_12 = TS2
      TA2EF_12 = TA2
      WA2EF_12 = WA2
      XI2EF_12 = XI2
      MS2EF_12 = MS2

```

C*****

```

      DZ = ZMAX/N
      S = MA * ZMAX / (VT * AV)
      MDOTSALT = MS * XI1
      Le = 0.8
      CALL FGFD(ts2,xi2,RELAX)
      TS2FD_8 = ts2
      WA2FD_8 = WA2
      TA2FD_8 = TA2
      XI2FD_8 = xi2
      MS2FD_8 = MS2
      Le = 1.0
      CALL FGFD(ts2,xi2,RELAX)
      TS2FD = ts2
      WA2FD = WA2
      TA2FD = TA2
      XI2FD = xi2
      MS2FD = MS2
      Le = 1.2
      CALL FGFD(ts2,xi2,RELAX)

```

```

    TS2FD_12 = ts2
    WA2FD_12 = WA2
    TA2FD_12 = TA2
    XI2FD_12 = xi2
    MS2FD_12 = MS2
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
    WRITE (9,25) NTU,TA2FD,WA2FD,TS2FD,TA2EF,WA2EF,TS2EF
        write (11,35) ntu,ta2fd_8,wa2fd_8,ts2fd_8,ta2fd,wa2fd,ts2fd,
1            ta2fd_12,wa2fd_12,ts2fd_12
        write (12,35) ntu,ta2ef_8,wa2ef_8,ts2ef_8,ta2ef,wa2ef,ts2ef,
1            ta2ef_12,wa2ef_12,ts2ef_12
15 CONTINUE
    25 FORMAT (' ',F7.3,2X,F9.3,2X,F8.4,2X,F9.3,2X,F9.3,2X,F8.4,2X,F9.3)
    35 format (' ',f6.2,x,3(f7.3,x,f6.5,f7.3))
100 FORMAT (' ',2(F7.2,2X,F10.3,2X,F10.4,2X),F8.3)
200 FORMAT (' ',F10.4,2X,F8.3,2X,F8.3,2X,I6)
300 FORMAT (' ',F12.5,2X,F9.5,2X,F12.6,2X,F10.5,2X,F10.6)
    STOP
    END
C*****
    SUBROUTINE FGFD(ts2g,xi2g,RELAX)
C
C This subroutine calculates the finite difference solution of a liquid desiccant
C chamber
C
    LOGICAL LOF, iflag, IFLAG2
    REAL MA, MS, MDOTSALT, MDOTS, MS2, NTU, LAMBDA, Le
    COMMON / INPUT / MA, TA1, WA1, MS, TS1, XI1, HD,
1        S, DZ, MDOTSALT, N, PATM, Le, Ts2,
2        TA2, MS2, XI2, NTU, WA2, HA1, HS1
    LUN = 6
    LOF = .FALSE.
    iflag = .false.
    IFLAG2 = .FALSE.
    tolX = 0.0001
    TOLT = 0.001
    T0 = 25.0

```

```

    LAMBDA = 2442.3
40 continue
    tl = ts2g
    TG = TA1
    TGK = TG + 273.15
    TLK = TL + 273.15
    WA = WA1
    HS2 = HSTSXI(TLK,XI2G,LUN,LOF)
    ENTHSOL = HS2
    ENTHAIR = HA1
    MDOTS = MDOTSALT/XI2g
    MS2 = MDOTS
    xi = xi2g
    DO 20 I=1,N
        TGK = TG + 273.15
        TLK = TL + 273.15
        cpm=(hatawa(tgk+1,wa,lun,lof)-hatawa(tgk-1,wa,lun,lof))/2
        HG = HD * cpm * Le
        CV0 = (ENTHAIR - HATAWA(TGK,0,LUN,LOF))/WA
        DWADZ = (HD/S) * (WSS(TL,XI,PATM) - WA)
        Q = HG * (TG - TL)
        DHADZ = DWADZ * CV0 - Q/S
        DMSDZ = MA * DWADZ
        MDOTS = MDOTS + DMSDZ * DZ
        DHSDZ = (MA/MDOTS) * (CV0*DWADZ - ENTHSOL*DWADZ - Q/S)
        WA = WA + DWADZ * DZ
        ENTHSOL = ENTHSOL + DHSDZ * DZ
        ENTHAIR = ENTHAIR + DHADZ * DZ
        XI = MDOTSALT/MDOTS
        TL = TSHSXI(ENTHSOL,XI,LUN,LOF) - 273.15
        TG = TAHAWA(ENTHAIR,WA,LUN,LOF) - 273.15
        IF (IFLAG2) THEN
c          WRITE (12,60) TG,TL,WA,entsol,enthair,q
60          format (' ',2(f8.3),f9.5,2(f8.3),f12.6)
          END IF
20 CONTINUE
    IF (IFLAG2) GO TO 50
    if (abs(tl-ts1).ge.tolT) then

```

```

      ts2g = ts2g + RELAX*(ts1-tl)
      IFLAG = .TRUE.
    end if
      if (abs(xi-xi1).ge.tolX) then
        xi2g = xi2g + RELAX*(xi1-xi)
        iflag = .true.
      end if
      if (iflag) then
        iflag = .false.
        go to 40
      ELSE
        IFLAG2 = .TRUE.
c      WRITE (12,*) ' '
        GO TO 40
      end if
50 CONTINUE
      TA2 = TG
      ts2 = ts2g
      wa2 = wa
      xi2 = xi2g
      dHa = ma * (ha1 - enthair)
      dHs = ms*hs1 - ms2*hs2
100 FORMAT (' ',2(F7.2,2X,F10.3,2X,F10.4,2X),F8.3)
200 FORMAT (' ',F10.4,2X,F8.3,2X,F8.3,2X,I6)
300 FORMAT (' ',F12.5,2X,F9.5,2X,F10.6,2X,F10.5,2X,F10.6)
      RETURN
      END
C*****
      SUBROUTINE EFF(ts2g,xi2g)
C
C   This subroutine calculates the effectiveness model solution of a liquid-desiccant
C   chamber
C
      LOGICAL LOF
      REAL MA, MS, NTU, MSTAR, MS2, MDOTSALT, Le, Ts2
      COMMON / INPUT / MA, TA1, WA1, MS, TS1, XI1, HD,
1      S, DZ, MDOTSALT, N, PATM, Le, Ts2,

```



```
2          TA2, MS2, XI2, NTU, WA2, HA1, HS1
LOF = .FALSE.
LUN = 6
HS2 = HSTSXI(TS2g + 273.15,XI2g,LUN,LOF)
CPS = (HS2 - HS1)/(TS2g - TS1)
CPSAT = CS(TS1,TS2g,XI1,PATM)
MSTAR = (MA * CPSAT)/(MS * CPS)
TERM = EXP(0.0 - NTU * (1.0 - MSTAR))
EFFECT = (1 - TERM)/(1 - (MSTAR * TERM))
ENTHA1 = HA1
HSS1 = HSS(TS1,XI1,PATM)
ENTHA2 = ENTHA1 + EFFECT * (HSS1 - ENTHA1)
ENTHASS2 = ENTHA1 + (ENTHA2 - ENTHA1)/(1-EXP(0.0-NTU))
HUMASS2 = WSSE(ENTHASS2,XI2,PATM)
WA2 = HUMASS2 + (WA1 - HUMASS2) * EXP(0.0-NTU/Le)
MS2 = MS + MA * (WA1 - WA2)
HS2 = (HS1*MS + MA*(ENTHA1-ENTHA2))/MS2
XI2 = (MS * XI1)/MS2
TS2 = TSHSXI(HS2,XI2,LUN,LOF) - 273.15
TA2 = TAHAWA(ENTHA2,WA2,LUN,LOF) - 273.15
100 FORMAT (' ',2(F7.2,2X,F10.3,2X,F10.4,2X),F8.3)
200 FORMAT (' ',F10.4,2X,F8.3,2X,F8.3,2X,I6)
300 FORMAT (' ',F12.5,2X,F9.5,2X,F10.6,2X,F10.5,2X,F10.6)
RETURN
END
```