PERFORMANCE OF SOLAR REGENERATED ROTATING BEDS OF SILICA GEL

by

Gopal P. Ananth

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Approved by:

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Major Professor

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

CHAPTER I

INTRODUCTION

Dehumidified air finds numerous applications--in food processing industries for production and storage of food ; in chemical industries such as paper and textile where humidity control is essential to achieve quality end products; in ships where circulation of dehumidified air is to be maintained to preserve interior structures; in the preservation of grain and lumber for long term storage; in human comfort, etc. Processes such as brewing, candy making, packaging of cereals, drying of leather, and manufacture of certain chemicals need to be carried out under dry atmosphere conditions.

Conventionally, dehumidification of air is achieved by cooling the air to the dew point temperature corresponding to the final humidity ratio of air. This forces the excess moisture to condense out, leaving behind drier air. In certain instances, such as drying of lumber, it may be required to reheat the dried air sensibly, to carry out the desired process more effectively. The process of cooling the air to dry it is called refrigeration dehumidification.

Almost all the refrigeration dehumidification apparatus is dependent on fossil fuel as the energy source. With increases in the requirements of dehumidified air and increasing costs of fossil fuel, a dehumidification process that could utilize solar energy as a main energy source attracted the attention of researchers.

Materials that could extract and hold water vapor from moist air when it is passed through them and release this vapor to another stream of hot air seem to promise to effectively use solar power. Materials that remove moisture, or in general any 'adsorbate', are called sorbents, and may be either solid or liquid.

A dehumidification system that uses silica gel as the sorbent has been built by Richard Singer, a former student of this school, to study the adsorption dehumidification process. Singer has also developed a computer program to predict the performance of the system. Preliminary runs taken on the actual system show good comparison between predicted and experimental values.

The objectives of this research were:

1. Additional data to be collected from the experimental apparatus mentioned above with different air flow rates, temperatures, and humidity ratios to study the performance of the system over a wider range of parameters.
2. The computer program to be modified such that the modeling of dehumidification systems with differing air stream flow rates, inlet conditions, etc, are permitted.
3. Once the revised computer program credibility has been established, predicted performance curves over all possible range of interest of the variables are to be obtained.
4. The use of the performance curves was to be demonstrated by the design of an adsorption air conditioning system.

CHAPTER II

ADSORPTION DEHUMIDIFICATION

Process Description

An adsorption process takes place through an operation which deals primarily with the utilization of surface forces, and the concentration of materials on the surfaces of solid bodies (referred to as adsorbents). Adsorption should be differentiated from absorption in that adsorption is commonly without any chemical reaction between the adsorbents and the material adsorbed (called the adsorbate), whereas in absorption, there is a chemical reaction or phase change taking place as a result of the process.

On a microscopic scale the sorption process can be explained as follows: When equilibrium is established between a gas and a solid surface, the concentration of gas molecules is always greater near the surface than it is in the bulk phase. This results in unbalanced forces being exerted on molecules near the surface of the solid normal to the solid plane. The balance of forces is partially restored by the migration of gas molecules to the surface. The process by which this concentrated surface layer of molecules is formed is called adsorption.

However, viewing the process on a larger scale, the mass transfer takes place due to a potential difference of vapor pressure between the sorbent and the flowing fluid. The vapor pressure of an adsorbent is negligible when it is anhydrous. Hence, when an active adsorbent is brought in contact with a gas of high humidity, the vapor pressure of water in the adsorbent tends to reach equilibrium with the partial pressure of water in the surrounding gas. This results in a decrease in the moisture content of the gas and an increase in moisture content of the adsorbent.

The sorption process always generates heat, which is mainly the result of condensation of water vapor. One other factor due to which heat is liberated is known as heat of wetting, which is developed when the liquid (water droplets) and solid (adsorbent) contact one another. The heat of wetting is large when the adsorbent is freshly reactivated and tapers off to a low value as the adsorbent approaches saturation. The heat of condensation and heat of wetting together is called the heat of sorption which is dissipated into the desiccant and the flowing air stream.

The equilibrium capacity of a given adsorbent, or a desiccant, for water, is basically a function of three variables--temperature of the air, moisture content of the air, and percent water adsorbed on the desiccant. To attempt to predict the performance of a desiccant analytically is a complex problem because the equations describing the conservation of energy and mass in the model are coupled and non linear. The equations are coupled because of liberation of heat of sorption and also because the equilibrium capacity of the sorbent depends on its temperature. The equations are non linear because the equilibrium concentrations of the sorbate (the material that is adsorbed, in this case, water) in air and the desiccant are not linearly related and also because the heat of sorption and the specific heats of the flowing fluid and desiccant are not constant.

Desiccants may be solid or liquid. Solid sorbents are generally an extremely porous 'solid foam', with large internal surface areas. When alumina is specially prepared or when silica is produced in its gel form, these solid materials may be used to take up moisture at room temperatures, from an air stream flowing through them. The process of adsorption by solid desiccants is reversible. If the vapor pressure of the adsorbed water becomes greater than the partial pressure of the vapor in the surrounding atmosphere, water will be released by the adsorbent.

Popular liquid sorbents are solutions in water of lithium chloride and of triethylene glycol. In this arrangement moist air is brought into direct contact with the adsorbent solution in a spray chamber. Liquid sorbents too can be reactivated, like solid sorbents.

Both solid and liquid desiccant dehumidification systems have their own merits and disadvantages. The present study concerns only solid desiccants and hence further discussions will be restricted only to them.

Sufficient reactivation or regeneration of solid sorbents can be made with air at temperatures of the order of 180°F. And since temperatures of this magnitude are attainable with the use of solar collectors, a dehumidification-regeneration system using solar energy has become particularly attractive.

As mentioned earlier, the air that is dehumidified, comes out warmer. In certain cases, this dry and warm air may find direct application, as in the case of drying grain, lumber, etc. But if cool and dry air is required, as for instance, in comfort air conditioning, the dry warm air may be sensibly cooled to preheat the regeneration air stream, thus improving the efficiency of the process.

Desiccant beds may be either stationary or rotating. Normally, in the case of stationary bed systems, there will be two beds; while one bed dehumidifies process air, the other bed is being regenerated.

And when the active bed nears its saturation, the process stream is diverted to the freshly regenerated bed and the other one is then fed the regeneration air. In a rotary dynamic dehumidifier, such as the system for the present study, a circular rotating bed structure is divided into two sections as shown in Figure 2-1. Through one of the sections flows the process air which needs to be dehumidified and through the other flows the hot regeneration air stream. Due to the rotary motion, freshly activated portions of the desiccant are always available for drying. A rotary dynamic dehumidifier system is simpler in construction because there are no reversing valves, and hence the piping layout is simplified. But, the problem of effectively sealing the leakage paths around the rotating bed is not insignificant. However, rotary dehumidifiers enjoy a lot more popularity than stationary ones.

A space cooling system can be contrived with three basic physical components--a rotary dehumidifier, rotary heat exchangers (regenerators) and spray/evaporative coolers, arranged in a variety of combinations.

Rotary heat exchangers are sensible heat exchangers with no moisture hold up capacity. These regenerators are made of porous matrix and have physically the same shape and size as the dehumidifier, because often these two are enclosed in the same housing. Common materials used in rotary sensible heat regenerators are aluminum wire, polyester film, steel wool, etc. The performance of these heat exchangers have been thoroughly studied and can be predicted with considerable accuracy.

The dehumidified air out of the desiccant will be warm and dry and cooled in the regenerator mentioned above. However, its dry bulb temperature may still not be low enough for comfort cooling applications.

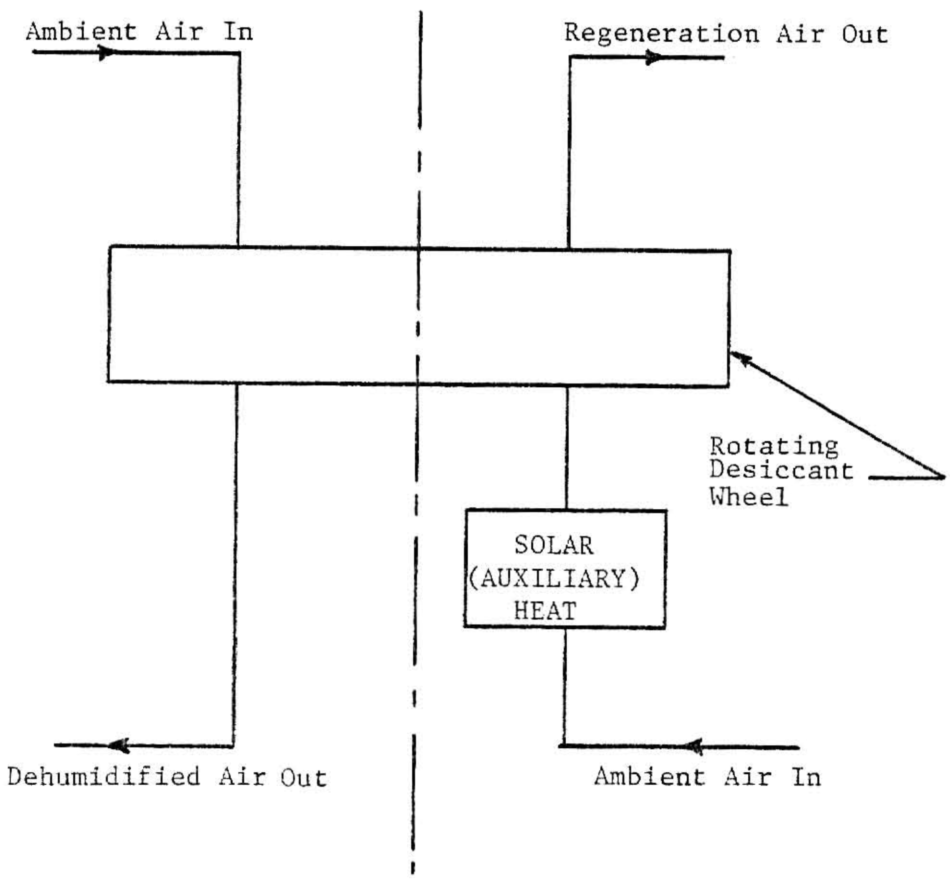

FIGURE 2-1 A ROTARY BED DEHUMIDIFIER

But by spray cooling or evaporative cooling, it is possible to obtain cool and dry air that can pick up sensible and latent loads in the conditioned space and maintain it within the comfort range.

The air conditioning system may be operated in a vent mode or recirculation mode. In a vent mode of operation, fresh air is conditioned in a dehumidifier, sensible heat exchanger and an evaporative cooler before it is led into the comfort space. After this air picks up the load in the room, it is evaporatively cooled and passed through the sensible heat exchanger (where it cools the supply air) and is heated with solar and/or other kinds of energy for regeneration of the desiccant.

Figure 2-2 shows a layout of the various components and Figure 2-3 shows a plot of a psychrometric chart indicating various statepoints, for a vent mode arrangement for comfort cooling.

The recirculation mode conditions the return air from the space and feeds it back to the room. Ambient air is preheated in the sensible heat exchanger and further heated to the desired regeneration temperature before it is used for the reactivation of the desiccant. Evaporative coolers are included in the arrangement to improve the performance of the system, as indicated in Figure 2-4. The statepoints of air at various locations is indicated in the psychrometric plot in Figure 2-5.

Now that an adsorption dehumidification air conditioning system has been presented, a feel for the energy requirement may be had. Primarily, the energy inputs are for heating regeneration air stream, for the blowers and for the desiccant and regenerator drive motors. If the regeneration air heating is supplemented with solar energy, there will definitely be a substantial saving. The blowers need to overcome,

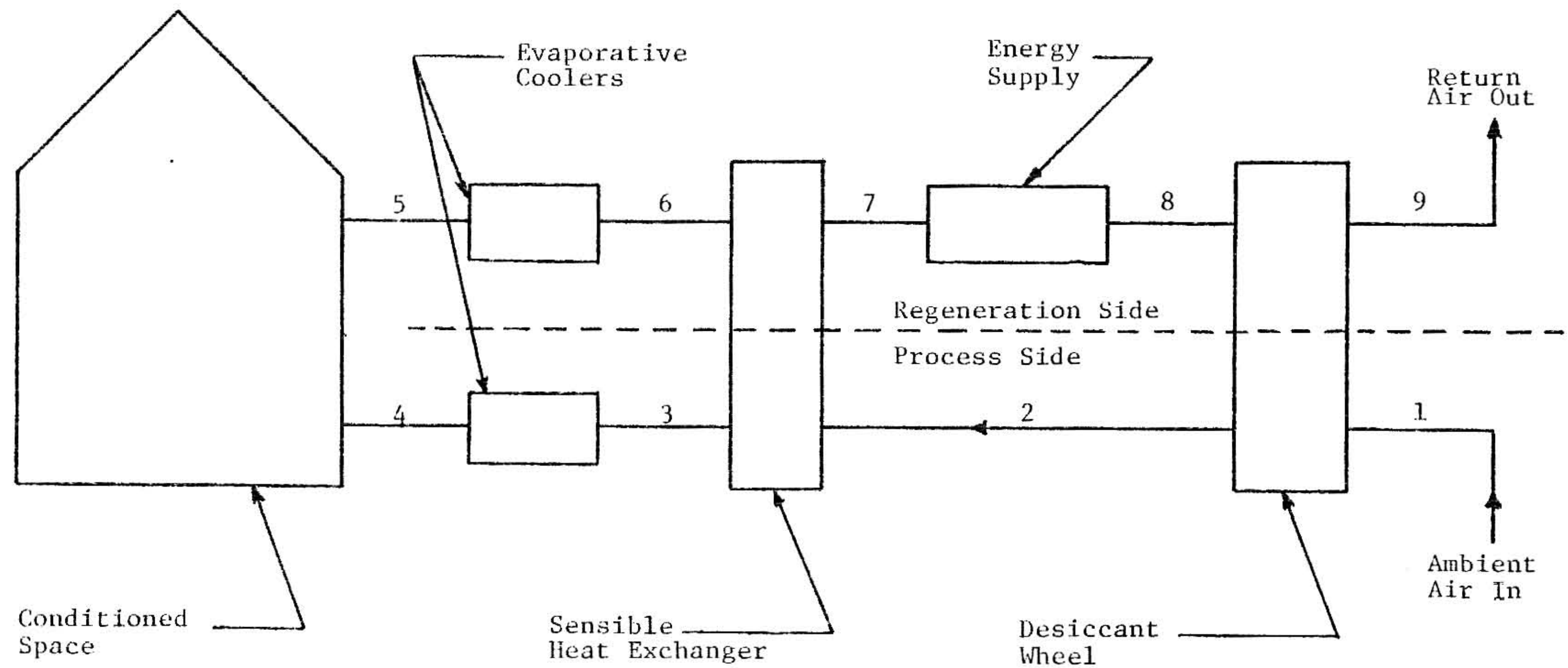

FIGURE 2-2 AN ADSORPTION AIR CONDITIONING SYSTEM OPERATING IN VENT MODE

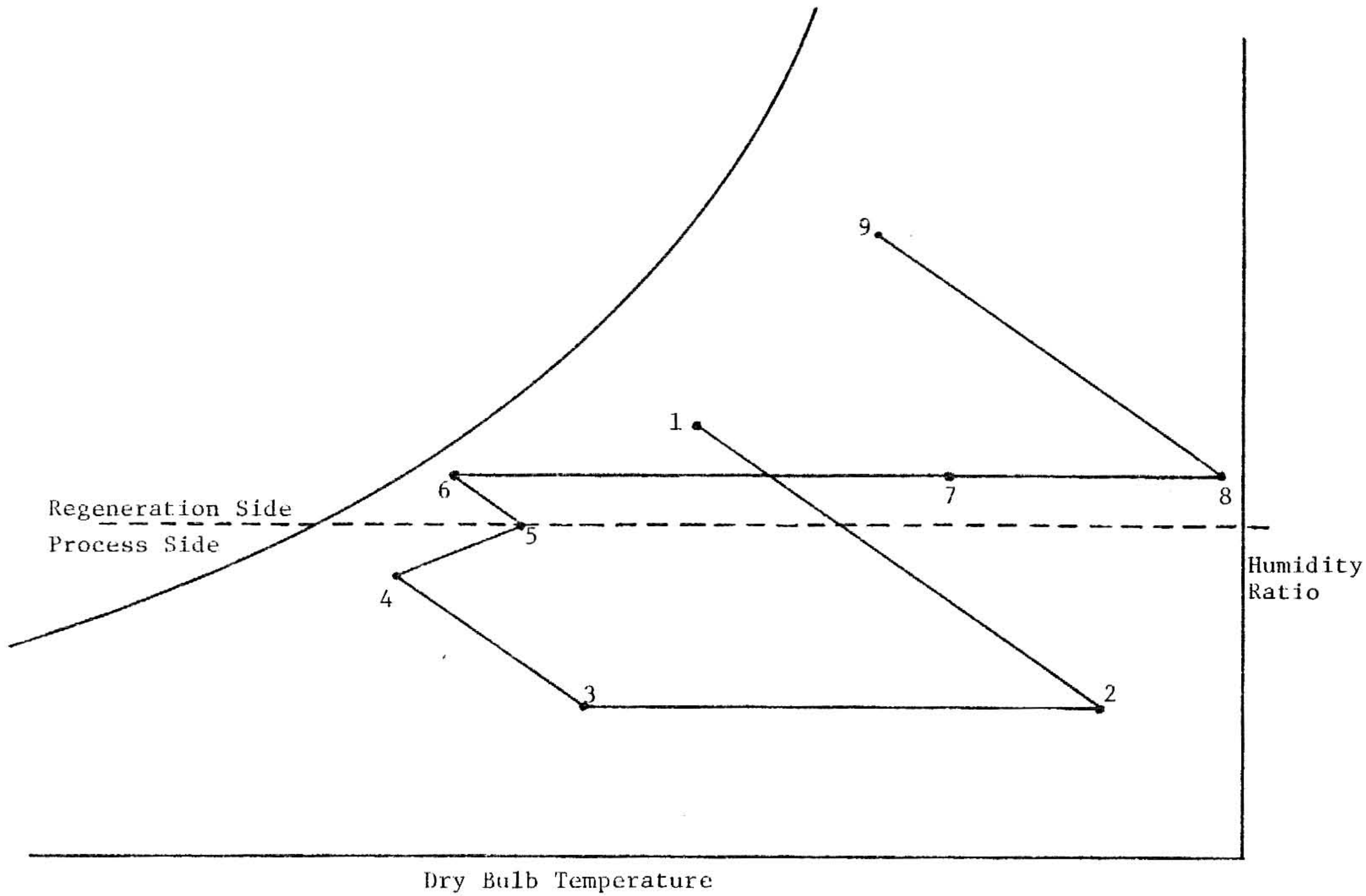

FIGURE 2-3 PSYCHROMETRIC CHART OF AN ADSORPTION AIR CONDITIONING SYSTEM OPERATING IN VENT MODE

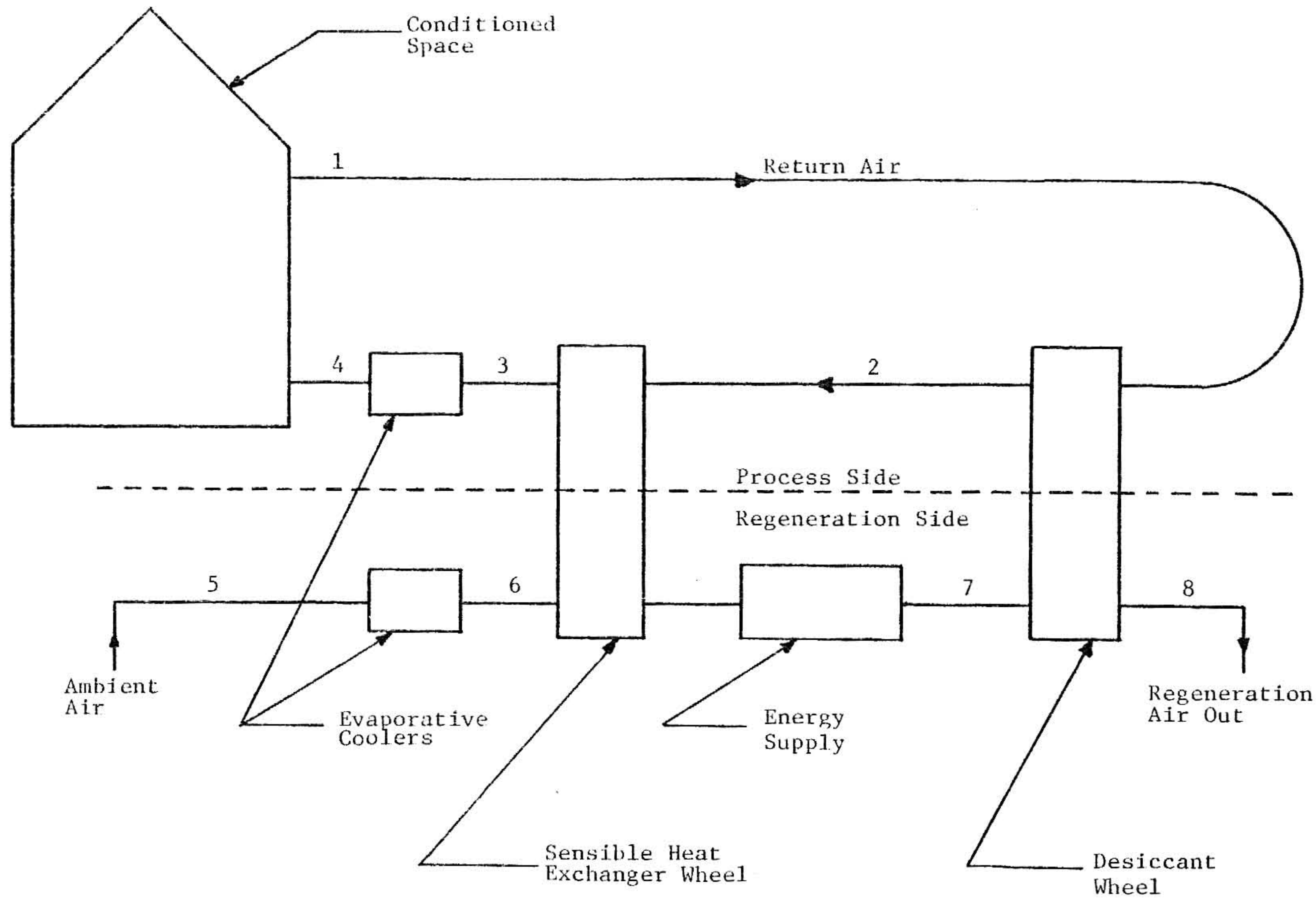

FIGURE 2-4 AN ADSORPTION AIR CONDITIONING SYSTEM OPERATING IN RECIRCULATION MODE

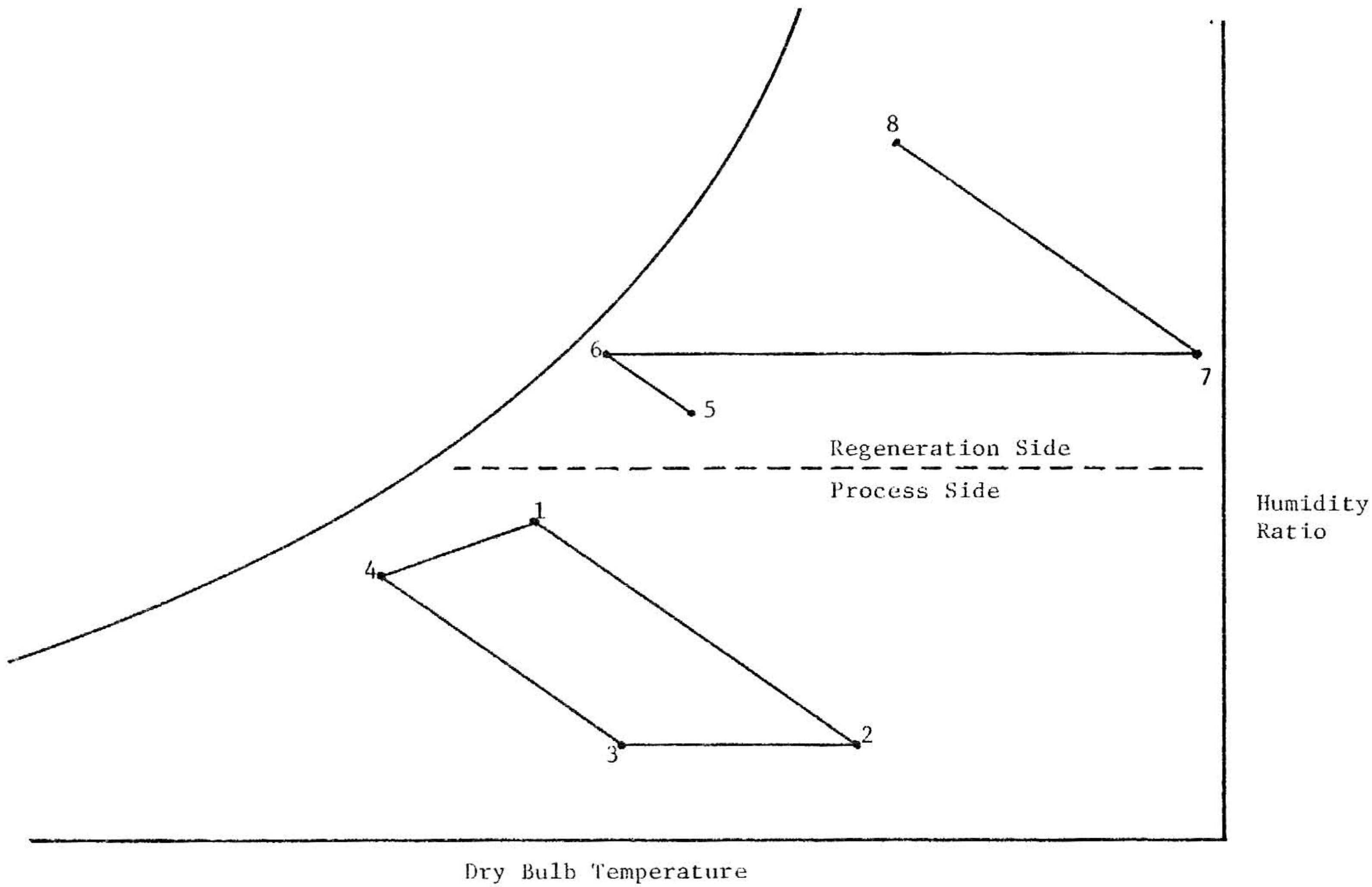

FIGURE 2-5 PSYCHROMETRIC CHART OF AN ADSORPTION AIR CONDITIONING SYSTEM OPERATING IN RECIRCULATION MODE

mainly, the pressure drops in the desiccant bed and the regenerator. These materials being extremely porous, the pressure drops across them are not considerable. Also, both these beds rotate at a very slow speed (of the order of 400 secs/rev to 600 secs/rev) and hence their drive motors can be expected not to consume much power.

Thus a preliminary analysis shows that an adsorption dehumidification air conditioning system could prove to be an economically viable system and merits further study.

Discussion of Papers

There are very many variables that decide the rate of moisture pick up by the desiccant. These variables concern the desiccant bed (type, particle size, area of the bed normal to the fluid flow, depth of bed, etc.), the process air (flow rate, temperature, moisture content, contact time, etc.), the quality of regeneration air stream, and other parameters such as the cycle time or rotational speed, leakage, etc.

In view of the large number of variables involved in the sorption process, it has not been easy to predict the performance of the dehumidifier exactly. Attempts at an analytical solution of the adsorption process are limited and the results available in the literature have been mainly of empirical nature only.

Following discussions will mainly concern an air-water vapor-silica gel system, which is the case for the present study.

Hubard (1) has published charts that tabulate data on the equilibrium between water vapor and commercial type silica gel over a wide range of temperatures and partial pressures.

Ross and McLaughlin (2) derived empirical equations for the adsorption-desorption characteristics of a stationary silica gel bed with the help of test data from an experimental set up. The obvious drawback of this method is that the reliability of predictions decreases when we move away from the standard values at which the readings were taken, and each different system would require an experimental investigation and analysis.

Bullock and Threlkeld (3) were probably the first persons to develop a fundamental analysis describing the adsorption process and to come up with a numerical solution for the analysis. Their experimental unit utilized a stationary bed and their predictions compared favourably with the experimental results.

Pla Barby (4) too developed a numerical model to simulate the performance of the air dryer. By using the finite difference scheme to solve the continuum equations with appropriate boundary conditions, his numerical method is capable of predicting the air properties at any instant of time at any depth of the bed. However, the computer model requires a significant amount of time (up to 1000 secs.).

The early analytical approach to the sorption process was provided by P. J. Banks (5). He attempted to solve the mass and energy conservation equations by the method of characteristics. When these equations are transformed to characteristic form, they are found to represent two kinematic wave equations, each describing the propagation of a small change in potential that depends on temperature and adsorbate concentrations. Each velocity of propagation is seen to depend on a "specific capacity ratio", and a "characteristic potential". An analogy with the heat transfer alone case suggests that the specific capacity ratio is similar to the ratio specfic heats of the porous medium and the fluid mixture which determines the velocity of a kinematic temperature wave and the characteristic potential is analogous to temperature. Banks' analysis was for transfer of a single adsorbate between a stationary porous medium and a single phase fluid mixture.

Close and Banks (6) studied further on Banks' work to obtain a compatible set of thermal and sorption properties for reversible sorption. They presented characteristic potentials and specific capacity ratios as plots on psychrometric charts, for an air-water vapor-silica gel system, which may be used to predict its behaviour under equilibrium conditions.

MacLaine-Cross and Banks (7) extended the above model to a rotating bed. The resulting solution is approximate because of the assumption that the heat/mass transfer is described by a single film transfer coefficient and diffusion in the fluid flow and sorption hysteresis are neglected.

Nelson (8) computerized the theory developed by (7) to provide the mean outlet temperatures and humidity ratios of air on process and regeneration sides. Nelson's computer model is different from Pla Barby's, in that Nelson's approach is analytical, and cannot provide the states of air at any instant and depth of bed, like Pla Barby's. But Nelson's program requires considerably less amount of computer time, with no loss in significant output.

Singer (9) modified Nelson's computer program and also included software for the performance of a solar collector. This enabled a comprehensive system of silica gel air dryer with solar energy powered regeneration to be modeled. A physical system was built comprising of a desiccant wheel, solar collectors and auxiliary heaters. Preliminary runs show that the performances of the air dryer and collectors are very well predicted by the computer program.

CHAPTER III

THE COMPUTER MODEL

The original program written by Singer (9) is in FORTRAN WATFIV language suitable for IBM 370 digital computer.

It was felt that it would be convenient to excute the program from an interactive terminal in order to be able to make changes in values of desired parameters from the keyboard. As a Hewlett Packard Model HP 9845 B desk top computer with such a facility was available in this Department's laboratory, the FORTRAN program was rewritten in BASIC language for use with this mini computer.

Although the computation time in BASIC is significantly more than that when executed in FORTRAN, this disadvantage is offset by many advantages that go with any interactive terminal.

The original FORTRAN program has been modified to accept humidity ratios and mass flow rates of air that are different on process and regeneration streams. The inlet process air temperature to the desiccant could be specified independent of the ambient temperature, whereas the original program assumed that this inlet air temperature is always 3°F above that of the ambient, which is assumed to be due to heat of compression in the blower.

The mathematical modeling of the rotary dehumidifier closely follows the analysis of Close and Banks (6) and MacLaine-Cross and Banks (7). The theory developed by them assumes, amongst other things, that there is no diffusion of water vapor in the flow direction, the fluid mixture is at constant pressure and there is no hysteresis. Although the analysis gets much simplified owing to these assumptions, accuracy of the results is not appeared to be lost.

The problem is to determine two outlet states (temperatures and humidity ratios) corresponding to two given inlet states of air--i.e. process and regeneration air states.

The analysis begins by first assuming that a part of the bed, in equilibrium with regeneration air stream is suddenly exposed to the process air and some other part of the bed in equilibrium with the process air stream is exposed to the regeneration air stream.

Close and Banks (6) have shown that the outlet states would follow paths of constant characteristic potentials, when a step change has occured in the inlet state of the fluid to the bed. For the present case of two component (enthalpy and water vapor) transfer system, the surfaces of constant characteristic potentials F_i, $i = 1,2$, become two sets of lines when plotted on a psychrometric type chart. The F_1 lines lie close to adiabatic saturation lines and F_2 lines to relative humidity lines. The assigned values of F_i can be completely arbitrary.

It should be noted that α_i, $i = 1,2$, which are slopes of F_i lines, are derivable from the thermal and sorption properties of the air-water vapor-silica gel system.

An analysis of the governing parameters for a real situation shows that F_1 waves propagate at a faster velocity than F_2 waves. In a rotary dehumidifier system, if the inlet fluid states of process and regeneration air streams are represented by points 1 and 2 respectively in Figure 3-1, then the outlet state of process air will be 3 and that of the regeneration stream will be 4, if the matrix rotational speed is such that only F_2 change in the outlet states occurs. In a practical situation, the actual outlet states on process and regeneration sides would be 3' and 4' respectively, which may be determined from states 3 and 4 by applying

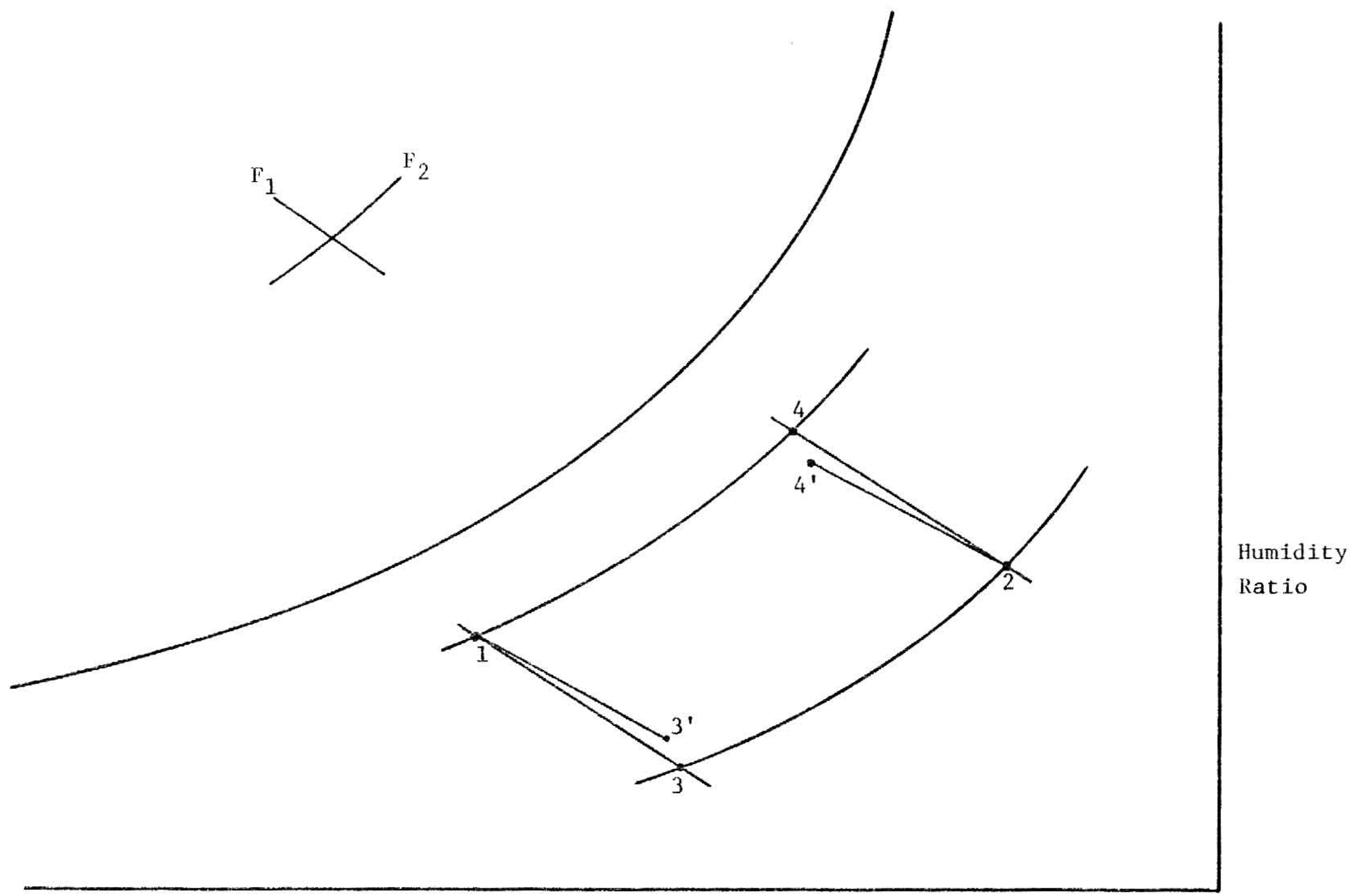

FIGURE 3-1 CHART SHOWING F_i LINES AND INLET AND OUTLET STATES

the theory developed by MacLaine-Cross and Banks (7).

The computer model first estimates points 3 and 4 by finding the points of intersection of straight lines with slopes α_i, through the points 1 and 2. Then with the assumption that the F_i lines through 1 and 2 are linear with slopes α_i for a very small temperature increment (assumed to be 1/60 of the inlet and estimated outlet points), the value of α_i at the new states are determined, and the process repeated until the two pairs of points converge on the two points 3 and 4.

Leakage around and across the dehumidifier bed could not be avoided. The original program was modified to enable the specification of peripheral leakage values on process and regeneration stream independent of each other, apart from the bypass and recirculation leaks. The possible leakage paths are indicated in Figure 3-2.

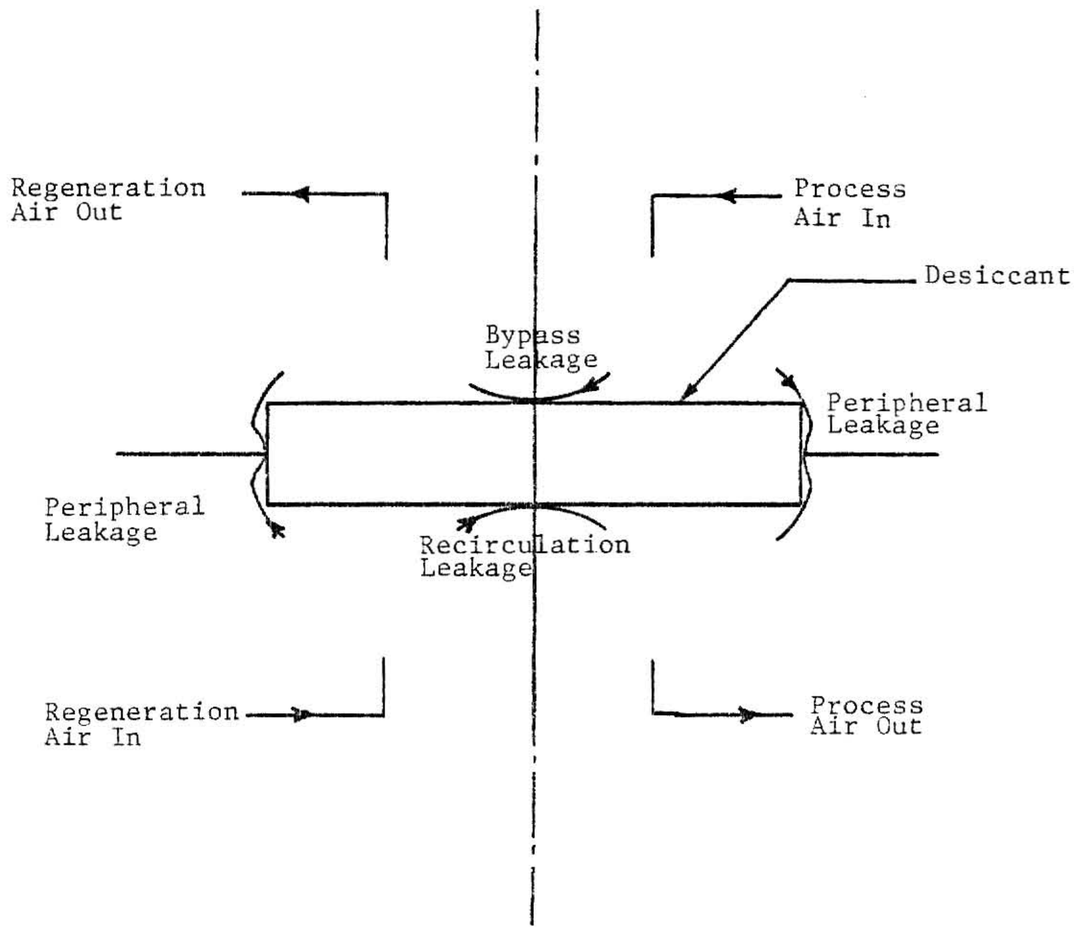

FIGURE 3-2 SCHEMATIC OF LEAKAGE PATHS IN A DEHUMIDIFICATION SYSTEM

CHAPTER IV

EXPERIMENTAL APPARATUS & RESULTS

Set Up

As one of the objectives of this research was to generate additional data from the existing desiccant system, the basic layout of the components was left unaltered. Some minor modifications were done to be able to change the humidity ratio of air on the process side. A brief description of the set up is given below. The layout of the system can be seen in Figure 4-1.

The main component of the physical model is the desiccant wheel made of silica gel sandwiched between two layers of perforated aluminum sheet. The gel bed is 17mm (0.67") thick, 290mm (11.42") in diameter and of grade 01 (3-8 mesh). The estimated particle diameter is 3.23mm (0.127"), surface to volume ratio is 714.6 m^2/m^3 (218 ft^2/ft^3) with a void fraction of 0.75.

A sensible heat exchanger wheel is provided, so that the process air which gets heated up after dehumidification, can be used to indirectly preheat the regeneration stream before the regeneration stream is further heated in the solar collectors. This sensible heat exchanger wheel is constructed from a 25mm (1") thick steel wool bed on a plastic support and a perforated aluminum sheet cover. This bed also is 290mm (11.42") in diameter with a fill density of 191 kg/m^3 (12 lbm/ft^3) and 98% void fraction. The materials comprising this sensible heat exchanger wheel are hydrophobic so that there is no latent heat exchange.

Air seals to prevent leakages of air between stationary and moving parts of the system are made from brass, felt and vinyl strips. The

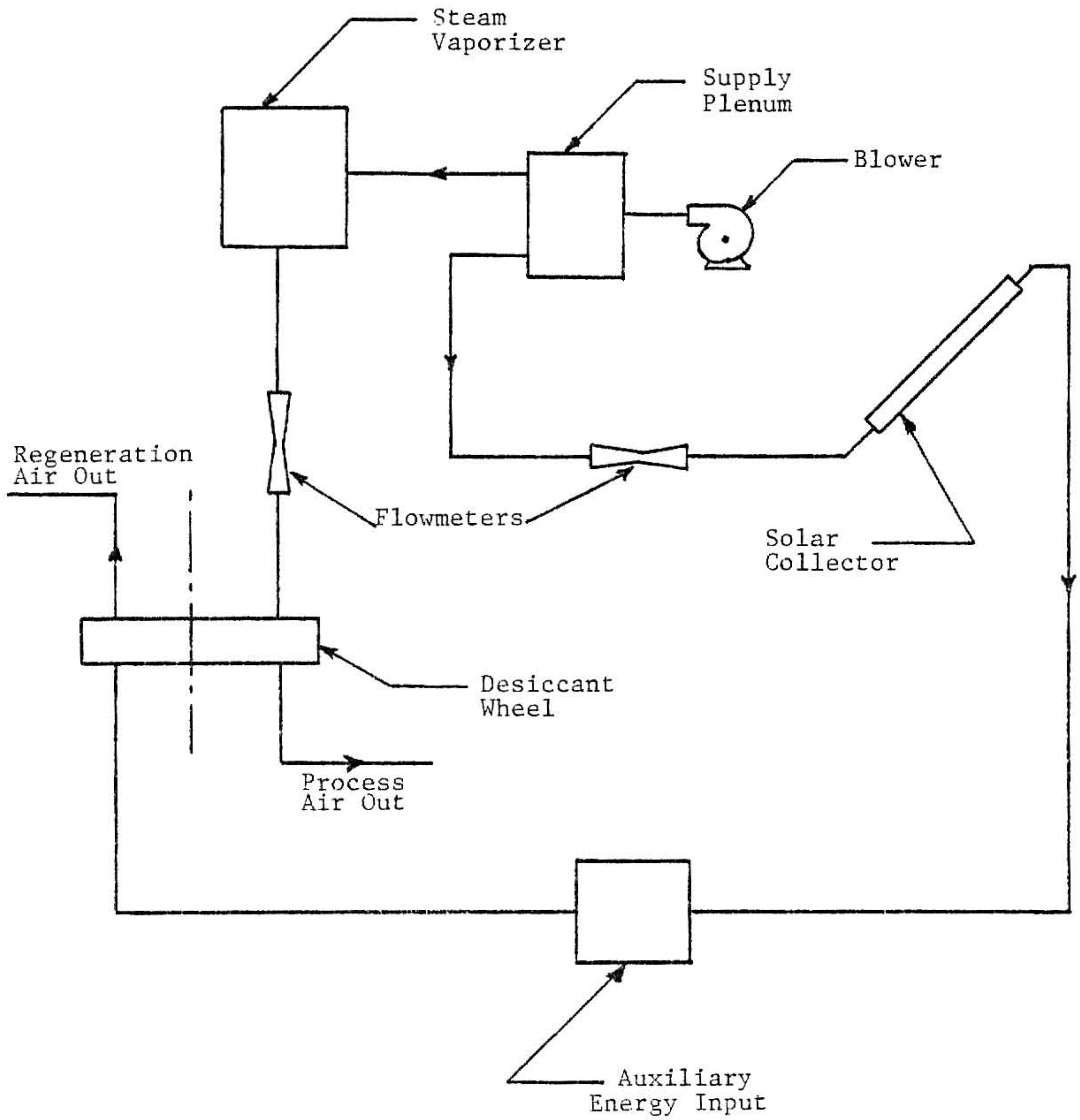

FIGURE 4-1 LAYOUT OF THE DEHUMIDIFICATION SYSTEM

possible leakage paths are indicated in Figure 3-2.

The two wheels are mounted on shafts and could be rotated at any desired independently controllable speeds with the aid of two DC motor drives, control systems, pulleys and belts.

The desiccant and sensible heat exchanger wheels, the motors and associated drives are enclosed in a sealed metallic enclosure made from 24 guage sheet metal. The chambers are well insulated to make the process get as close to adiabatic as possible.

The regeneration air is heated in two Solaron Series 3000 Air Collectors, each having an aperture area of 1.586 m^2 (17.07 ft^2), with a single 3.2mm (0.125") thick tempered glass glazing. The absorptivity and emissivity of the selective surface are 0.95 and 0.15 respectively. Recommended air flow rate through each collector is 17.7 liters/sec (37.4 scfm) to 26.5 liters/sec (56.2 scfm).

Air supply to process and regeneration streams is provided by a single blower that has a capacity to discharge 322 liters/sec (682 scfm) at a static head of 3"wg at a rated speed of 3450 rpm. The blower discharges air into a supply plenum made from 24 gauge sheet metal. Two outlets--one for process air and the other for regeneration air are provided. Air flow rate to each stream can be regulated by operating the guillotine type dampers provided on these outlets.

An auxiliary (electric) heater of 660 watts capacity can be energized in the event the solar collectors being unable to meet the desired regeneration air temperature. This cone glo coil is contained in a sheet metal enclosure down stream to the collector. A temperature controller senses the air temperature at the inlet of the desiccant wheel on the regeneration side and energizes the auxiliary heater, if this temperature falls below a pre set regeneration temperature. To ensure safety in operation, a sail switch is installed in the fan discharge plenum, that deenergizes a power relay cutting off power to the auxiliary heater in case air flow is insufficient.

To have the ability to increase the moisture content in the process air, if necessary, a commercial 7.57 liter (2 gallon) capacity electric steam vaporizer is placed in the circuit. This vaporizer is contained in an insulated box provided with inlet and outlet ports on adjacent faces so that thorough mixing of steam with air can be expected. The steam generation rate and hence humidity ratio can be controlled by controlling the voltage to the vaporizer.

The existing pipings were made of 102mm (4") diameter flexible plastic duct. Wherever possible, this duct was replaced with straight galvanized duct of the same size. Batt insulation is wrapped around all duct work which is then covered with polythene sheet to protect it from environment.

Thermocouples made from T-Copper-Constantan are installed at various locations throughout the system. At least two thermocouples are located in every location to minimize any error due to stratification. The lead wires of the thermocouples are covered with aluminum foil to shield them from radio frequency interference.

From each of the four chambers of the desiccant wheel--inlet and outlet chambers on process and regeneration sides, air can be tapped off for sampling, through four ¼" copper tubings. The air sample is used to determine its moisture content using an electronic dew point meter. A five way motor operated timer actuated valve lets an air sample flow from any one of the four tubings to the dew point analyzer. To avoid condensation in the sampling lines that could affect the reading of the dew point meter, a 160 watt heating tape is wrapped around the copper tubings. A differential temperature thermostat energizes this tape if the temperature drop in the copper tubing exceeds the set temperature difference in the thermostat.

Solar radiation on the plane of the collectors is measured using two methods--an Epply Precision Spectral pyranometer and a Hollis Obervatory Model MR 5A 2092 pyranometer with its solar integrator that provides instantaneous as well as integrated outputs.

Wind direction and speed are measured with the help of a Climatronics Wind measuring system, the output appearing on a pressure sensitive paper.

Flow rates of process and regeneration air streams are measured using a 2½" annubar and 2" ASME long radius flow nozzle respectively. Oil filled inclined manometers are used to measure the pressure differentials in these flow measuring devices.

All the thermocouple leads as well as outputs from the dew point meter and the precision spectral pyranometer, were led to a data logger which is the heart of the data acquisition system. The data logger can be programmed to scan the channels at any desired rate, average out individual channel signals, convert them to the specified units and print out the average values at the set rate. During this experiment, the scanning rate was set at 15 seconds and printer rate at 60 minutes.

It was observed that leakages across the sensible heat exchanger and dehumidifier could not be effectively controlled. Since the performance of the sensible heat exchanger is very well known and its exclusion would not be affecting the study of performance of the dehumidifier, it was decided to bypass the sensible heat exchanger wheel.

Data Taking Procedure

Ambient conditions need to be conducive to run the experiment. Hence when the early morning hours appear promising and the forecast for the day predicted good sunshine and other desirable weather conditions, the following procedure was adopted to gather data.

The desiccant wheel, the blower and all instrumentation are turned on a couple of hours before the start of the actual data taking time. This allowed warming up of all equipment and enables dehumidification process to reach a steady state. All the instruments are tested for zero setting and balanced. The guillotine dampers in the supply plenum are then adjusted to provide required, preferably balanced, flow rates on process and regeneration streams.

The ambient dry bulb and dew point temperatures are measured. From these values, and knowing the mass flow rate of air on the process side, the amount of moisture that needs to be added in this stream is computed to arrive at a final desired relative humidity. This value is of course only indicative in nature, as, as the day wears on, the dry bulb and wet bulb temperatures could change requiring different quantity of steam to be evaporated. The steam vaporizer is switched on and the variable transformer is adjusted so that steam evaporation rate is close to what is required. The reading of dew point meter, that continously samples air from the inlet on the process side during this initial period, is compared with the dry bulb temperature at the same location, read by the data logger. The heating tape that heats the copper tubing is left permanently on as a safety measure to avoid any condensation in the copper tubing.

Just prior to actual data taking, the timer controlling the five way dew point sampling valve, is adjusted such that the first sampling line leading to the dew point meter is exposed 15 minutes after the start of the test and subsequent lines at 15 minute intervals thereafter. Thus the data logger prints out the first dew point temperature starting 30 minutes after the beginning of the test, and subsequent values at 15 minute intervals thereafter. Hence the last dew point reading for a given hour would be printed 15 minutes later because it represents the effect of the preceding hour better due to lagging dehumidification-humidification characteristics of the desiccant wheel.

Discussion of Results

As mentioned earlier, all the data gathered were with the sensible heat exchanger bypassed. The prediction of the performance of solar collectors used in this research has been discussed adequately by Singer (9), and so, it was decided to concentrate data gathering solely around the dehumidifier.

A copy of the results from the computer analysis of the experimental data obtained on June 19, 1982, is presented in Table 4-1. Table 4-2 compares the experimental results with the predicted values.

Data gathering began after an initial warm up period of about 3 hours. At the time of start of the experiment, the process and regeneration flow rates were adjusted to the desired values. It was decided to have different flow rates in each stream to check the validity of the computer model which has been modified to analyze

TABLE 4-1 ANALYSIS OF DATA TAKEN ON JUNE 19, 1982--COPY OF OUTPUT

Analysis of data taken on JUNE 19, 1982

Process side flow rate	: .01778 kg/s (141.1 lb/hr)
Regeneration side flow rate	: .02836 kg/s (225.1 lb/hr)
Matrix rotational speed	: .0105 rad/s (600 s/rev)
Thickness of bed/Frontal area	: .017 m / .0364 m2
Pr. drop, proc./regn. sides	: .571 / 1.177 inches of water
Recirculation/Bypass leaks	: .400 / .400
Peripheral leak, proc./regn.	: .200 / .200

PROCESS SIDE PARAMETERS

	INLET CONDITIONS		OUTLET CONDITIONS	
HOUR ENDING	Temperature (Deg C)	Humidity Ratio (kg/kg)	Temperature (Deg C)	Humidity Ratio (kg/kg)
12	32.3	0.0159	59.6	0.0093
13	34.7	0.0161	63.0	0.0100
14	35.6	0.0156	62.5	0.0097
15	36.9	0.0154	63.0	0.0095
16	35.9	0.0146	58.8	0.0093
17	35.8	0.0135	55.4	0.0089

REGENERATION SIDE PARAMETERS

	INLET CONDITIONS		OUTLET CONDITIONS	
HOUR ENDING	Temperature (Deg C)	Humidity Ratio (kg/kg)	Temperature (Deg C)	Humidity Ratio (kg/kg)
12	75.1	0.0078	46.3	0.0147
13	80.6	0.0087	50.9	0.0152
14	79.8	0.0081	51.5	0.0144
15	80.8	0.0074	53.3	0.0136
16	72.8	0.0079	48.9	0.0134
17	67.0	0.0077	46.8	0.0124

TABLE 4-2 ANALYSIS OF DATA TAKEN ON JUNE 19, 1982--COMPARISON WITH EXPERIMENTAL VALUES

		Outlet Temperatures				Outlet Humidity Ratios		
Time CDT	T_{in} (°C)	T_{exp} (°C)	T_{cal} (°C)	ΔT (°C)	W_{in} g/Kg	W_{exp} g/Kg	W_{cal} g/Kg	ΔW g/Kg
Process Side								
12	32.3	58.0	59.6	+1.6	15.9	10.7	9.3	-1.4
13	34.7	62.8	63.0	+0.2	16.1	11.3	10.0	-1.3
14	35.6	62.8	62.5	-0.3	15.6	10.1	9.7	-0.4
15	36.9	64.3	63.0	-1.3	15.4	9.9	9.5	-0.4
16	35.8	58.6	58.8	+0.2	14.6	9.2	9.3	+0.1
17	35.8	55.6	55.4	-0.2	13.5	8.9	8.9	0.0
Regeneration Side								
12	75.1	46.4	46.3	-0.1	7.8	14.0	14.7	+0.7
13	80.6	51.8	50.9	-0.9	8.7	14.0	15.2	+1.2
14	79.8	52.6	51.5	-1.1	8.1	13.9	14.4	+0.5
15	80.8	54.2	53.3	-0.9	7.4	14.0	13.6	-0.4
16	72.8	49.7	48.9	-0.8	7.9	13.8	13.4	-0.4
17	67.0	47.9	46.8	-1.1	7.7	12.8	12.4	-0.4

systems with unbalanced flow rates. The steam vaporizer in the process line was also turned on and its steam generation rate was set to a level that would not lead to any condensation of moisture anywhere in the system.

The temperature readings are the averaged values of thermocouples, if there are more than one in a location. The four thermocouples located in the outlet chamber of the desiccant wheel on the process side differed in readings by up to $10^{o}F$, which could have been due to thermal stratification of air in this compartment. The values presented in the tables are the averaged value of the four thermocouples.

A study of Table 4-2 leads to the following discussion:

The computer model almost always predicts the air to come out of the dehumidifier drier than experimental values. This may lead one to conclude that the peripheral leak, that is, air flowing to the outlet along the periphery of the wheel without passing through the desiccant, is more than that has been accounted for in the computer program. However, it may be observed that after longer hours of operation, the predicted and experimental values are very close to each other.

The outlet temperatures on process and regeneration sides have been predicted with considerable accuracy. As for the small variations of temperatures between actual and predicted values on the process side, no significant conclusions are able to be drawn, since the variations do not indicate any trend. However, the computer model predicts that the regeneration air always comes out cooler. This may be due to some heat transfer across the insulated compartment walls, which has not been provided for, in the computer program. The regeneration outlet chamber is adjacent to the relatively cooler process inlet chamber which might lead to cooling of the regeneration stream.

The moisture picked up by the regeneration stream during the first three hours of operation is lesser than that predicted, but during the subsequent three hours, it is greater. There is no apparent reason for this trend; however, the differences are less than the measurement accuracy. Additionally, the trend of the above results is consistent with that of Singer (9). The leakage rates assumed by him and in this present analysis are the same.

The order of magnitude of errors in the predicted humidity ratios are also the same in the present study as was in Singer's (9). However, it should be noted that the moisture content in the inlet air is about 16 g/kg and the moisture removed is about 6.5 g/kg, whereas at the time Singer conducted his experiment, without artificially humidifying ambient air, the humidity ratio was of the order of 5 g/kg and the removal rate was about 1.5 g/kg.

The dew point meter, which was used to measure the humidity ratios, was observed to drift through 1°C. When the humidity ratios are of the order of 16 g/kg, a discrepency of 1°C in the dew point meter reading could produce an error of 1.2 g/kg in the humidity ratio reading, which is very significant compared to the error between the predicted and experimental humidity ratios of air. This error could be minimized probably by letting the data logger average the dew point meter readings over a period and print this averaged value, instead of printing the instantaneous value of the dew point meter every 15th minute.

The process air in the above discussed experiment was at relative humidities of about 40%. The following experiment was conducted with air that was having relative humidities of up to 85%.

The data presented were taken on June 23 and June 24, 1982. The intention was to study the performance of the dehumidifier on a long-term operation. Readings were gathered after a 12 hour warm up period, and it was decided to run the experiment continuously for 24 hours. There was very little sunshine on June 23rd. Therefore, hardly any solar energy was gathered by the collectors. The ambient air was also very humid. These two factors had a detrimental effect on the regeneration capacity of air, and hence the dehumidification was not very effective.

Table 4-3 lists the predicted and actual values over a 26 hour period starting at 7 AM on June 23. It may be noted that the humidity ratios on process and regeneration inlets are the same, because the steam vaporizer was not turned on due to very humid conditions.

Unlike in the previous case, where the predicted process outlet temperatures showed no consistency in variation with the actual values, the present case indicates that process air in the experimental set up always comes out a little cooler than the calculated values. It cannot be said that this is due to insufficient dehumidification or larger peripheral leakage than assumed, because, then the outlet humidity ratios should consistently be higher than the predicted values, which is not the case. The humidity ratios on the process outlet side vary on both the sides of the calculated values, on more occasions coming out wetter than predicted.

On the regeneration side, the air comes out warmer and the stream does not pick up as much moisture as the computer model predicts it

TABLE 4-3 COMPARISON OF PREDICTED AND EXPERIMENTAL VALUES FOR JUNE 23, JUNE 24, 1982

		Outlet Temperatures				Outlet Humidity Ratios		
Time CDT	T_{in} (^oC)	T_{exp} (^oC)	T_{cal} (^oC)	ΔT (^oC)	W_{in} g/Kg	W_{exp} g/Kg	W_{cal} g/Kg	ΔW g/Kg
Process Side								
7	19.6	29.7	30.6	+0.9	10.1	9.1	8.7	-0.4
8	18.2	28.4	28.9	+0.5	11.4	9.9	10.0	+0.1
9	17.5	27.4	28.1	+0.7	11.4	10.0	10.0	0.0
10	18.3	29.2	31.2	+2.0	12.2	11.1	10.4	-0.7
11	21.2	35.7	37.8	+2.1	13.2	11.7	11.0	-0.7
12	23.8	39.9	42.7	+2.8	14.1	10.9	11.6	+0.7
13	26.4	46.8	49.8	+3.0	12.8	10.8	10.0	-0.8
14	26.9	47.4	49.0	+1.6	12.5	10.1	9.9	-0.2
15	27.8	48.3	50.0	+1.7	13.2	10.5	10.5	0.0
16	26.8	46.9	48.3	+1.4	13.7	10.3	11.0	+0.7
17	27.6	50.8	53.2	+2.4	13.0	10.6	10.0	-0.6
18	28.7	51.4	52.3	+0.9	13.2	11.3	10.4	-0.9
19	29.3	47.6	48.3	+0.7	14.2	12.2	11.8	-0.4
20	28.3	43.6	44.2	+0.6	14.6	12.2	12.5	+0.3
21	26.6	39.9	40.4	+0.5	14.0	12.2	12.1	-0.1
22	25.2	36.6	37.1	+0.5	14.3	12.3	12.6	+0.3
23	23.1	34.3	34.8	+0.5	14.0	12.1	12.4	+0.3
24	21.9	33.0	33.7	+0.7	13.7	12.2	12.1	-0.1
1	21.2	32.1	33.0	+0.9	13.6	11.8	12.0	+0.2
2	20.9	31.8	32.9	+1.1	13.3	11.8	11.6	-0.2
3	20.8	31.6	32.8	+1.2	13.1	11.6	11.4	-0.2

Time CDT	T_{in} (^oC)	T_{exp} (^oC)	T_{cal} (^oC)	ΔT (^oC)	W_{in} g/Kg	W_{exp} g/Kg	W_{cal} g/Kg	ΔW g/Kg
		Outlet Temperatures				Outlet Humidity Ratios		
4	20.7	31.5	32.6	+1.1	13.0	11.5	11.4	-0.1
5	20.3	31.1	32.3	+1.2	12.9	11.6	11.2	-0.4
6	20.1	31.0	32.2	+1.2	12.9	11.0	11.2	+0.2
7	20.1	31.0	32.4	+1.4	12.9	11.7	11.2	-0.5
8	21.9	33.9	36.2	+2.3	13.4	12.1	11.4	-0.7
Regeneration Side								
7	36.9	25.8	25.7	-0.1	10.1	10.4	11.5	+1.1
8	34.6	24.2	23.8	-0.4	11.4	11.8	12.8	+1.0
9	33.7	23.6	23.0	-0.6	11.4	11.7	12.8	+1.1
10	38.0	25.1	24.9	-0.2	12.2	12.3	13.9	+1.6
11	46.8	29.6	29.9	+0.3	13.2	14.0	15.4	+1.4
12	52.9	33.3	33.5	+0.2	14.1	16.0	16.7	+0.7
13	63.7	38.2	39.7	+1.5	12.8	15.2	16.0	+0.8
14	62.3	39.4	39.4	0.0	12.5	14.2	15.3	+1.1
15	63.2	40.8	40.3	-0.5	13.2	15.1	16.2	+1.1
16	60.3	39.7	38.2	-1.5	13.7	15.2	16.5	+1.3
17	68.6	43.1	42.3	-0.8	13.0	14.2	16.4	+2.2
18	66.6	44.4	42.2	-2.2	13.2	14.6	16.2	+1.6
19	59.1	37.4	39.6	+2.2	14.2	15.4	16.9	+1.5
20	52.9	38.2	36.6	-1.6	14.6	15.5	16.8	+1.3
21	47.6	35.0	33.6	-1.4	14.0	14.4	15.9	+1.5
22	43.2	32.4	31.2	-1.2	14.3	14.8	16.0	+1.2
23	40.8	30.2	29.0	-1.2	14.0	14.6	15.7	+1.1
24	39.7	28.8	27.7	-1.1	13.7	14.3	15.3	+1.0

Time CDT	T_{in} (^oC)	T_{exp} (^oC)	T_{cal} (^oC)	ΔT (^oC)	W_{in} g/Kg	W_{exp} g/Kg	W_{cal} g/Kg	ΔW g/Kg
		Outlet Temperatures				Outlet Humidity Ratios		
1	39.1	28.0	27.1	-0.9	13.6	14.2	15.2	+1.0
2	39.0	27.7	26.9	-0.8	13.3	13.9	14.9	+1.0
3	39.0	27.4	26.8	-0.6	13.1	13.8	14.7	+0.9
4	38.8	27.2	26.7	-0.5	13.0	13.8	14.6	+0.8
5	38.5	26.8	26.3	-0.5	12.9	13.6	14.5	+0.9
6	38.4	26.6	26.2	-0.4	12.9	13.3	14.5	+1.2
7	38.8	26.6	26.3	-0.3	12.9	13.4	14.5	+1.1
8	43.6	29.1	29.1	0.0	13.4	14.6	15.3	+0.7

should. That is, actually the regeneration is less effective. This may be due to larger peripheral leakages by this stream than assumed in the program.

It was therefore decided to re-run the computer model with the regeneration side peripheral leak increased from 20% to 30%. The results of this change are presented in Table 4-4. It must be observed that the predicted values on the regeneration side are much closer to the actual values now, but the values on the process stream remain unaltered.

The data for June 19, 1982 also has been analyzed with the above revised leakage rates and are presented in Table 4-5. In this case, it cannot be conclusively said that the predicted performance has shown an improvement.

The peripheral seal on the desiccant had to be replaced because the original seal peeled off. Presumably for this reason, leakage rate different from those used by Singer (9) has resulted in more accurate predictions.

Due to the limited amount of reaearch done on this area, very little literature was available for comparison of present results with those of others. Pla Barby (4) has carried out investigations on the performance of rotating beds of silica gel. But, due to insufficient data, for instance, lack of information on flow rates, and also because of the unsymmetrical nature of his desiccant bed (30% regeneration area and 70% process area), his results could not be directly compared with the results of the present study.

Even though the results of the present study could not be compared with those of others, the good agreement between the experimental values

TABLE 4-4 EFFECT OF REVISION IN LEAKAGE RATES ON COMPUTED VALUES FOR JUNE 23, JUNE 24, 1982.

Time	PROCESS SIDE				REGENERATION SIDE			
	ΔT (°C)		ΔW (g/Kg)		ΔT (°C)		ΔW (g/Kg)	
CDT	Orig.	Rev.	Orig.	Rev.	Orig.	Rev.	Orig.	Rev.
7	+0.9	+0.9	-0.4	-0.4	-0.1	+1.0	+1.1	+0.7
8	+0.5	+0.5	+0.1	+0.1	-0.4	+0.6	+1.0	+0.6
9	+0.7	+0.7	0.0	0.0	-0.6	+0.4	+1.1	+0.7
10	+2.0	+2.0	-0.7	-0.7	-0.2	+1.1	+1.6	+1.2
11	+2.1	+2.1	-0.7	-0.7	+0.3	+1.9	+1.4	+0.9
12	+2.8	+2.8	+0.7	+0.7	+0.2	+2.2	+0.7	+0.1
13	+3.0	+3.0	-0.8	-0.8	+1.5	+2.7	+0.8	0.0
14	+1.6	+1.6	-0.2	-0.2	0.0	+2.2	+1.1	+0.4
15	+1.7	+1.7	0.0	0.0	-0.5	+1.7	+1.1	+0.4
16	+1.4	+1.4	+0.7	+0.7	-1.5	+0.7	+1.3	+0.6
17	+2.4	+2.4	-0.6	-0.6	-0.8	+1.7	+2.2	+1.4
18	+0.9	+0.9	-0.9	-0.9	-2.2	+0.1	+1.6	+0.8
19	+0.6	+0.6	-0.4	-0.4	-2.2	-0.3	+1.5	+0.8
20	+0.6	+0.6	+0.2	+0.2	-1.6	0.0	+1.3	+0.9
21	+0.5	+0.5	-0.1	-0.1	-1.4	0.0	+1.5	+1.0
22	+0.5	+0.5	+0.3	+0.3	-1.2	0.0	+1.2	+0.8
23	+0.5	+0.5	+0.3	+0.3	-1.2	-0.1	+1.1	+0.7
24	+0.7	+0.7	+0.1	+0.1	-1.1	-0.1	+1.0	+0.6
1	+0.9	+0.9	+0.2	+0.2	-0.9	+0.3	+1.0	+0.6
2	+1.1	+1.1	+0.2	+0.2	-0.8	+0.4	+1.0	+0.6
3	+1.2	+1.2	-0.2	-0.2	-0.6	+0.6	+0.9	+0.5
4	+1.1	+1.1	-0.1	-0.1	-0.5	+0.7	+0.8	+0.4

Time CDT	PROCESS SIDE ΔT (°C) Orig.	Rev.	ΔW (g/Kg) Orig.	Rev.	REGENERATION SIDE ΔT (°C) Orig.	Rev.	ΔW (g/Kg) Orig.	Rev.
5	+1.2	+1.2	-0.4	-0.4	-0.5	+0.7	+0.9	+0.5
6	+1.2	+1.2	+0.2	+0.2	-0.4	+0.8	+1.2	+0.8
7	+1.4	+1.4	-0.5	-0.5	-0.3	+1.0	+1.1	+0.7
8	+2.3	+2.3	-0.7	-0.7	0.0	+1.4	+0.7	+0.3

TABLE 4-5 EFFECT OF REVISION IN LEAKAGE RATES ON COMPUTED VALUES FOR JUNE 19, 1982.

	PROCESS SIDE				REGENERATION SIDE			
	ΔT (°C)		ΔW (g/Kg)		ΔT (°C)		ΔW (g/Kg)	
Time CDT	Orig.	Rev.	Orig.	Rev.	Orig.	Rev.	Orig.	Rev.
12	+1.6	+1.6	-1.4	-1.4	-0.1	+2.8	+0.7	-0.2
13	+0.2	+0.2	-1.3	-1.3	-0.9	+1.9	+1.2	+0.3
14	-0.3	-0.3	-0.4	-0.4	-1.1	+1.6	+0.5	-0.4
15	-1.3	-1.3	-0.4	-0.4	-0.9	+1.6	-0.4	-1.1
16	+0.2	+0.2	+0.1	+0.1	-0.8	+1.5	-0.4	-1.1
17	-0.2	-0.2	0.0	0.0	-1.1	+0.8	-0.4	-1.0

and computer predictions over a wide range of operating conditions, is a proof of validity of the computer model.

CHAPTER V

AIR DRYER PERFORMANCE

It was shown in the preceding chapter that the computer program is capable of predicting the performance of the dehumidifier system with reasonable accuracy. The objective of this chapter is to utilize the computer model in generating performance characteristics of the silica gel air dryer over a wide range of operation. Such graphs would help in the choice of optimum parameters for a given set of operating conditions.

There are numerous variables that play roles in deciding the extent to which process air is dehumidified. Certain variables, such as bed depth, frontal exposed area, rotational speed of the bed, etc., can be optimized for the best performance. However, there are a few independent variables, such as the regeneration air temperature (if the source of regeneration energy is solar power) or the regeneration air humidity ratio, over which the designer has no control.

The performance of the silica gel dryer is primarily determined by its effectiveness in dehumidifying the process air and hence all the plots presented here are drawn to indicate the variations in outlet humidity ratio of process air with certain parameters held constant and varying one specified parameter. The air streams are assumed to be balanced, in counterflow arrangement, with equal process and regeneration areas.

It is obvious that a more effective regeneration of the silica gel can be achieved using air at higher regeneration temperatures. A well regenerated desiccant performs better dehumidification because the partial pressure of the water vapor in it is very low. Figures 5-1, 5-2, and

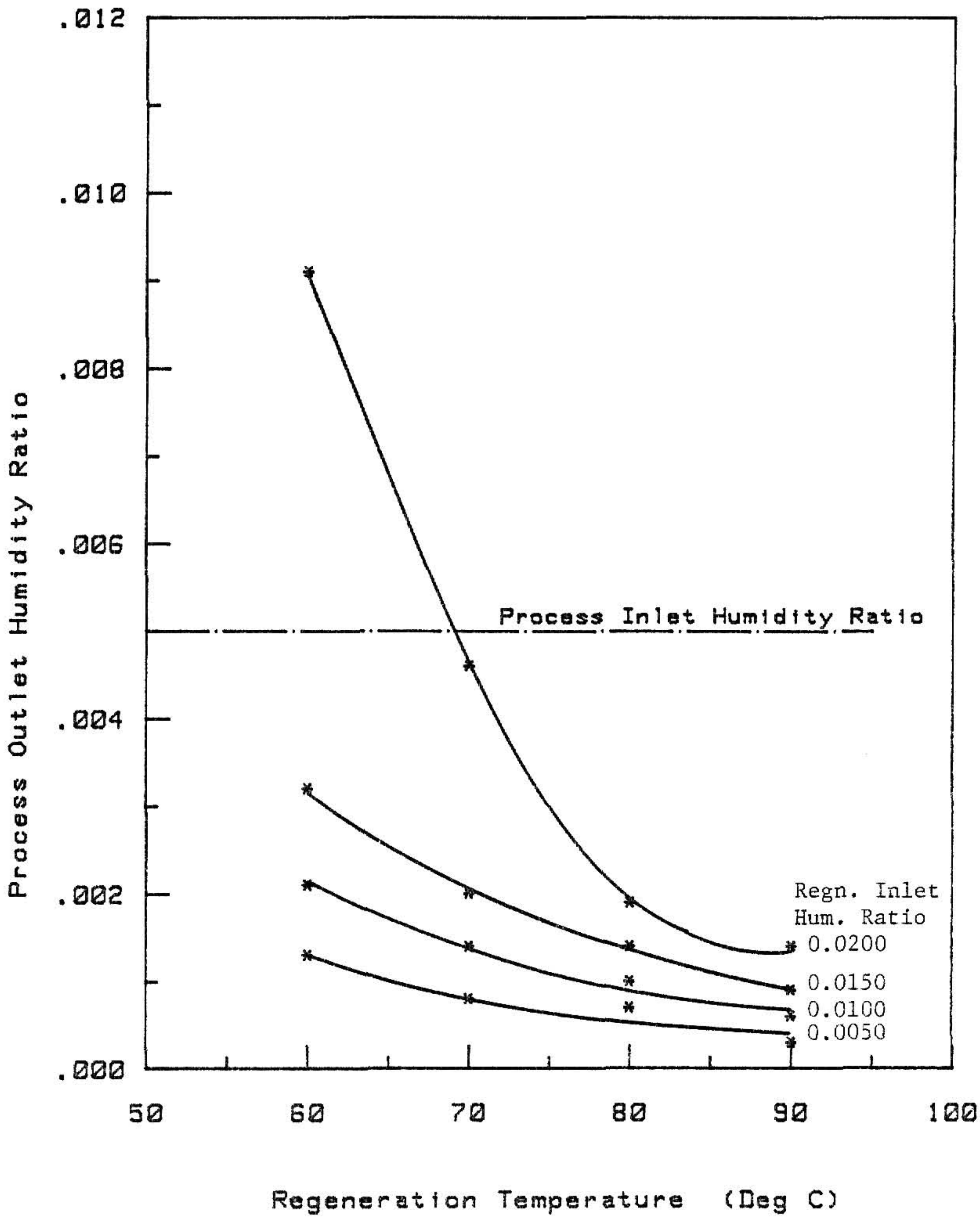

FIGURE 5-1 EFFECT OF VARIATION OF REGENERATION TEMPERATURE-PROCESS INLET HUMIDITY RATIO = 0.0050

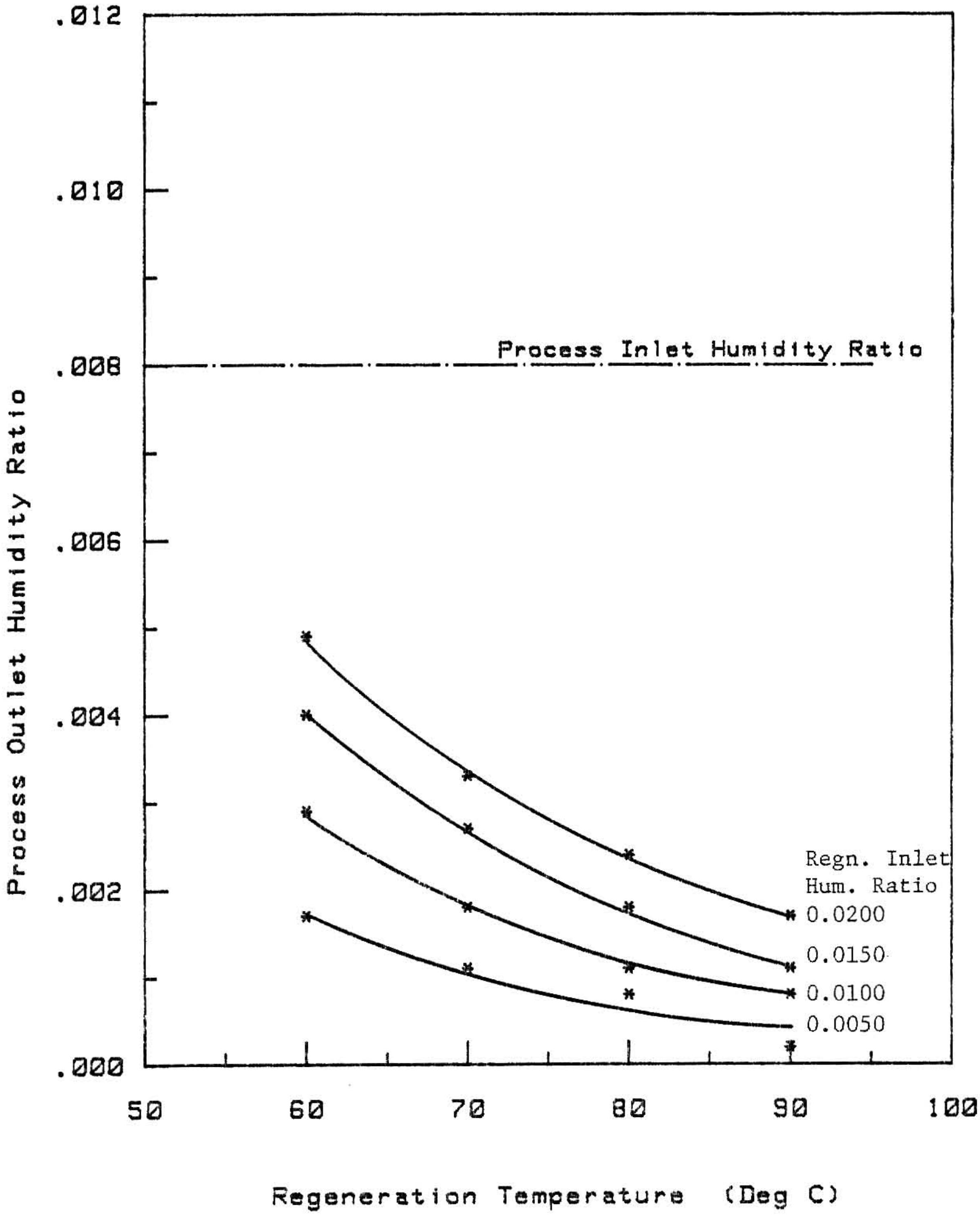

FIGURE 5-2 EFFECT OF VARIATION OF REGENERATION TEMPERATURE-
PROCESS INLET HUMIDITY RATIO = 0.0080

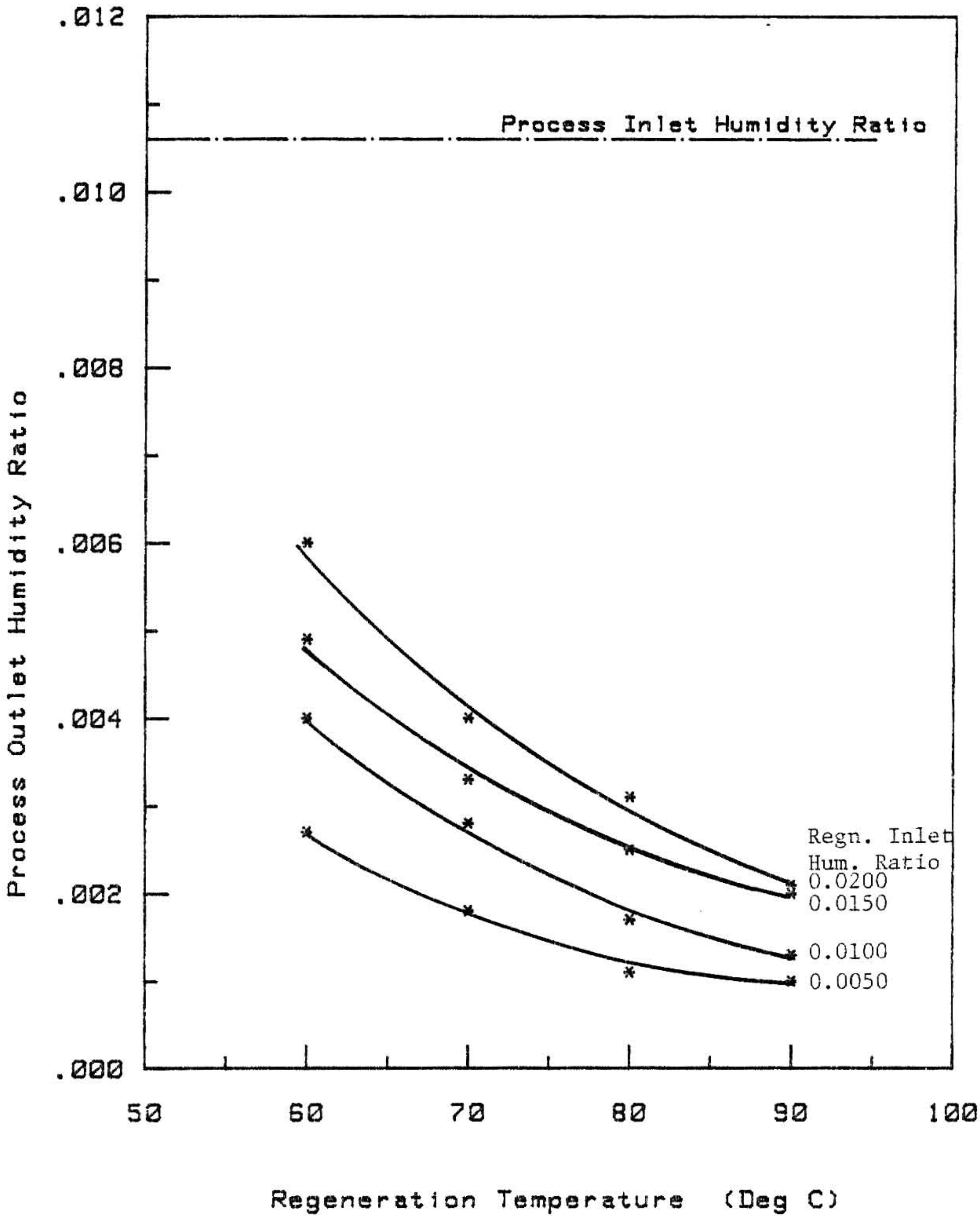

FIGURE 5-3 EFFECT OF VARIATION OF REGENERATION TEMPERATURE-
PROCESS INLET HUMIDITY RATIO = 0.0106

5-3 show the effect of variation of regeneration air remperatures on the process outlet humidity ratios for varying moisture contents in the regeneration stream, for three different process inlet humidity ratios. The trend in all the three plots are the same: the dehumidification process shows rapid improvement with increase in regeneration temperatures. Hubard's equilibrium data for silica gel and water vapor (1) show that, at the equilibrium water vapor partial pressure in the region of our interest (0.2" Hg to 1.0" Hg corresponding to dew point temperatures of 2°C to 26°C), the useful water content adsorbed by silica gel decreases at a very fast rate with increase in silica gel temperatures of up to 80°C. Hence regeneration of silica gel at around 80°C results in a very well reactivated bed.

One other fact that may be observed from these three figures is the small sensitivity of the dehumidification capacity of the bed to the moisture content of regeneration air stream at high regeneration air temperatures. At such high temperatures, the regeneration air temperature and not its moisture content, is the dominating factor on the regeneration efficiency.

A particular case in Figure 5-1 shows the process air getting humidified during its passage through the bed. With regeneration air of about 68°C or lower and with humidity ratio of 0.0200, the partial pressure of water vapor in the regenerated bed is greater than that in the process air entering the bed. Hence the mass transfer is from the bed to the air.

Figure 5-4 shows the effect of inlet temperature of process air on the dehumidification capacity of the silica gel bed. Increase in inlet air temperature from 15° to 35°C increases the outlet air humidity ratio from 0.0012 to 0.0044, with all other parameters constant. Thus if the

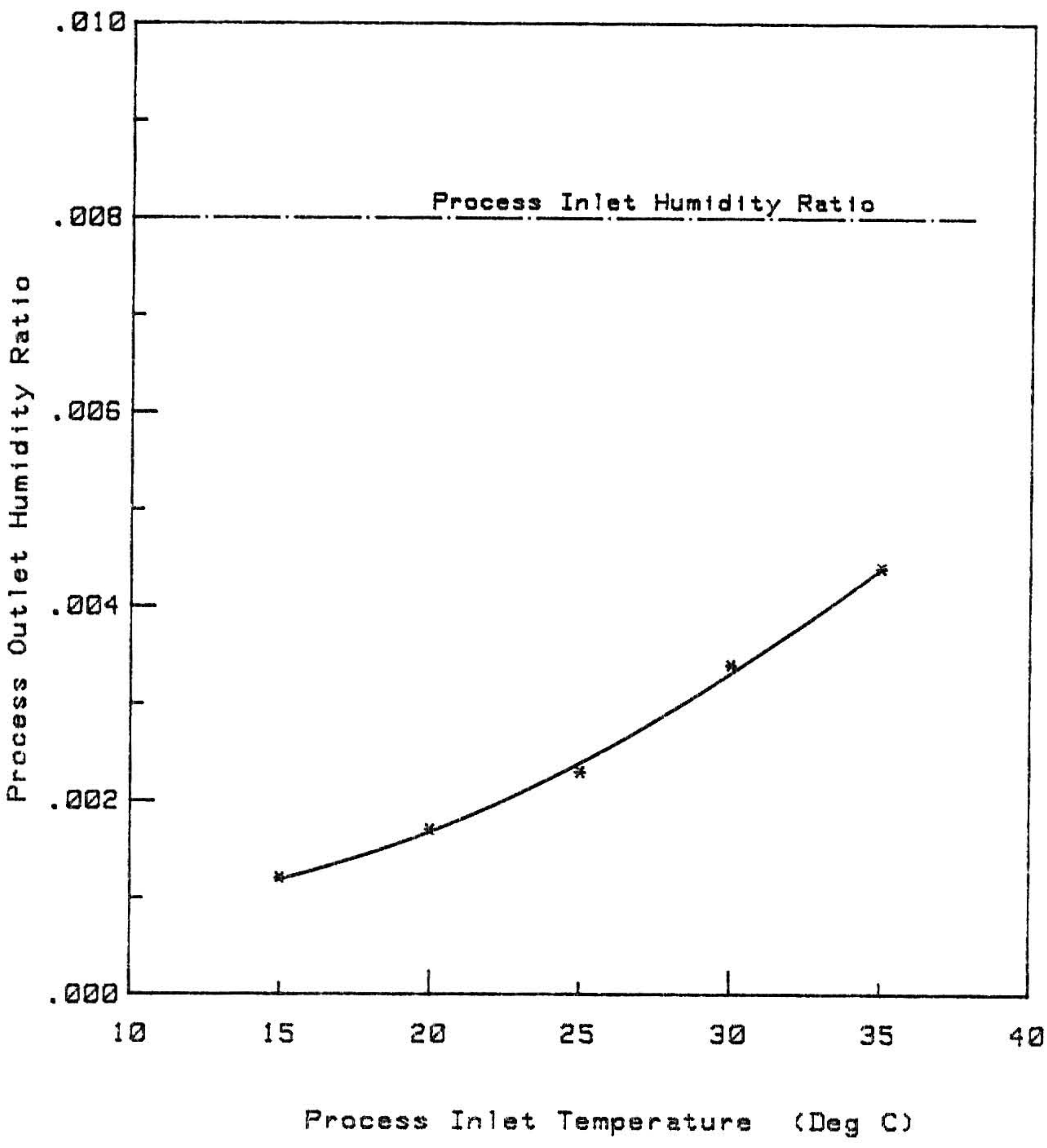

FIGURE 5-4 EFFECT OF VARIATION OF PROCESS INLET TEMPERATURE

dryer dehumidifies ambient air, say, for an air conditioning application, it can be expected to be less efficient during the day time, when the air temperature goes up. But it is likely that the regeneration steam temperature too will go up for the same reason and this is going to counteract the decreasing effectiveness of dehumidification.

Figure 5-5 is a plot of process outlet humidity ratio vs process inlet humidity ratio, with the process inlet air temperature fixed at 25^{o}C and regeneration air at 90^{o}C. Variations in inlet humidity ratio does not appear to significantly change moisture content at the outlet of the desiccant bed. For instance, a change in the inlet humidity ratio of 0.09 (from 0.005 to 0.014) changes the outlet condition by only 0.0023 (from 0.0006 to 0.0029) for the particular case of regeneration air having a humidity ratio at inlet of 0.0100.

Figure 5-6 shows the effect of speed of rotation of the matrix on the outlet humidity ratio for three different bed thicknesses. For the bed with the smallest thickness (8mm), the dehumidification process gets rapidly inefficient as the speed of revolution is slowed down. At very low speeds of revolution, a thin bed could get saturated before it moves over to the regeneration chamber and the process air will come out without getting dehumidified. However, as the thickness of the bed increases, its ability to hold moisture increases and so the dehumidification is effective even with speeds of revolution as low as 1/1000 rev/sec.

High rotational speeds, of say less than 200 secs/rev, result in reduced performance because of insufficient regeneration period. At very high rotational speeds, the contact time between the air and the desiccant will be too small to result in any dehumidification/regeneration.

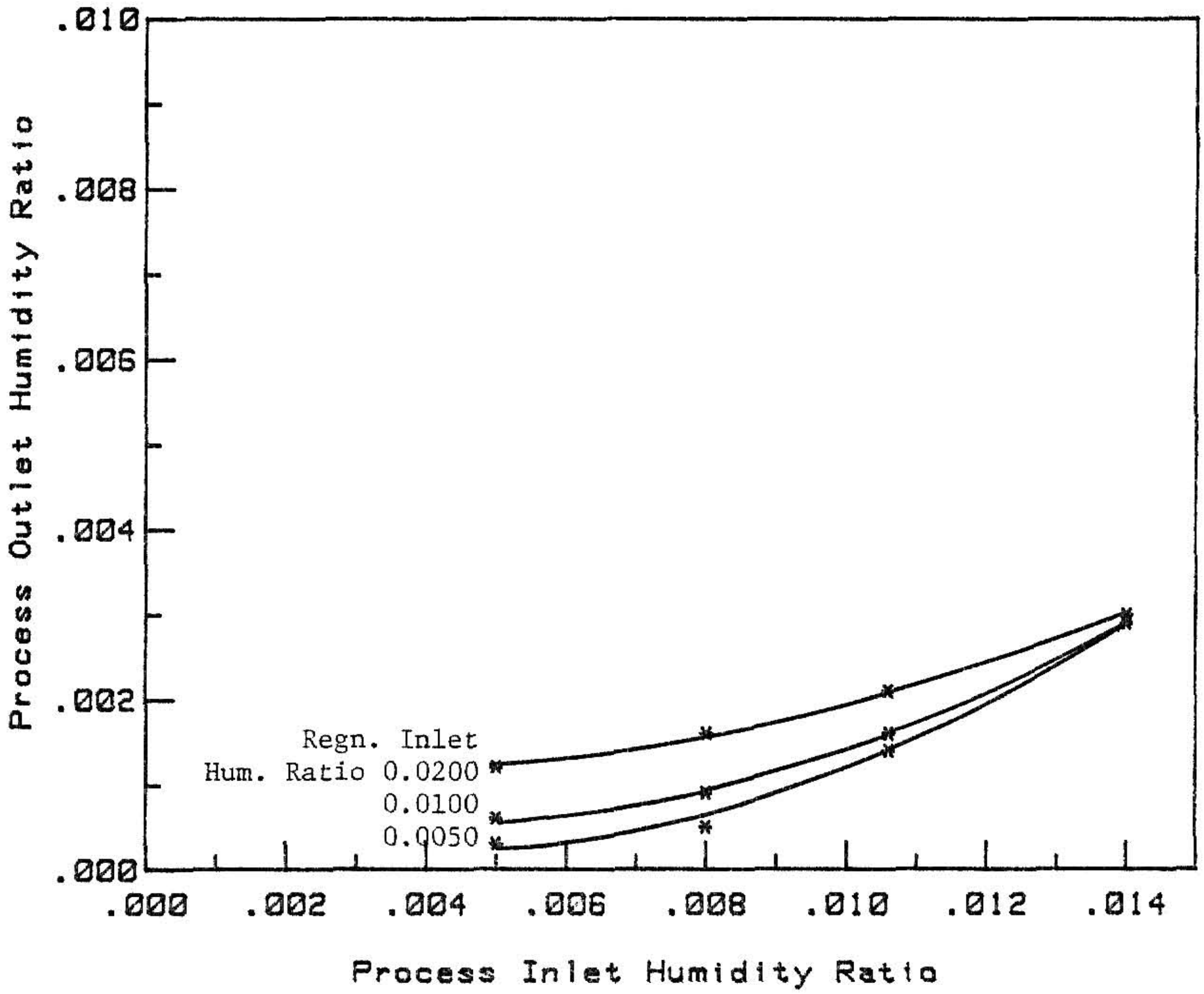

FIGURE 5-5 EFFECT OF VARIATION OF PROCESS INLET HUMIDITY RATIO

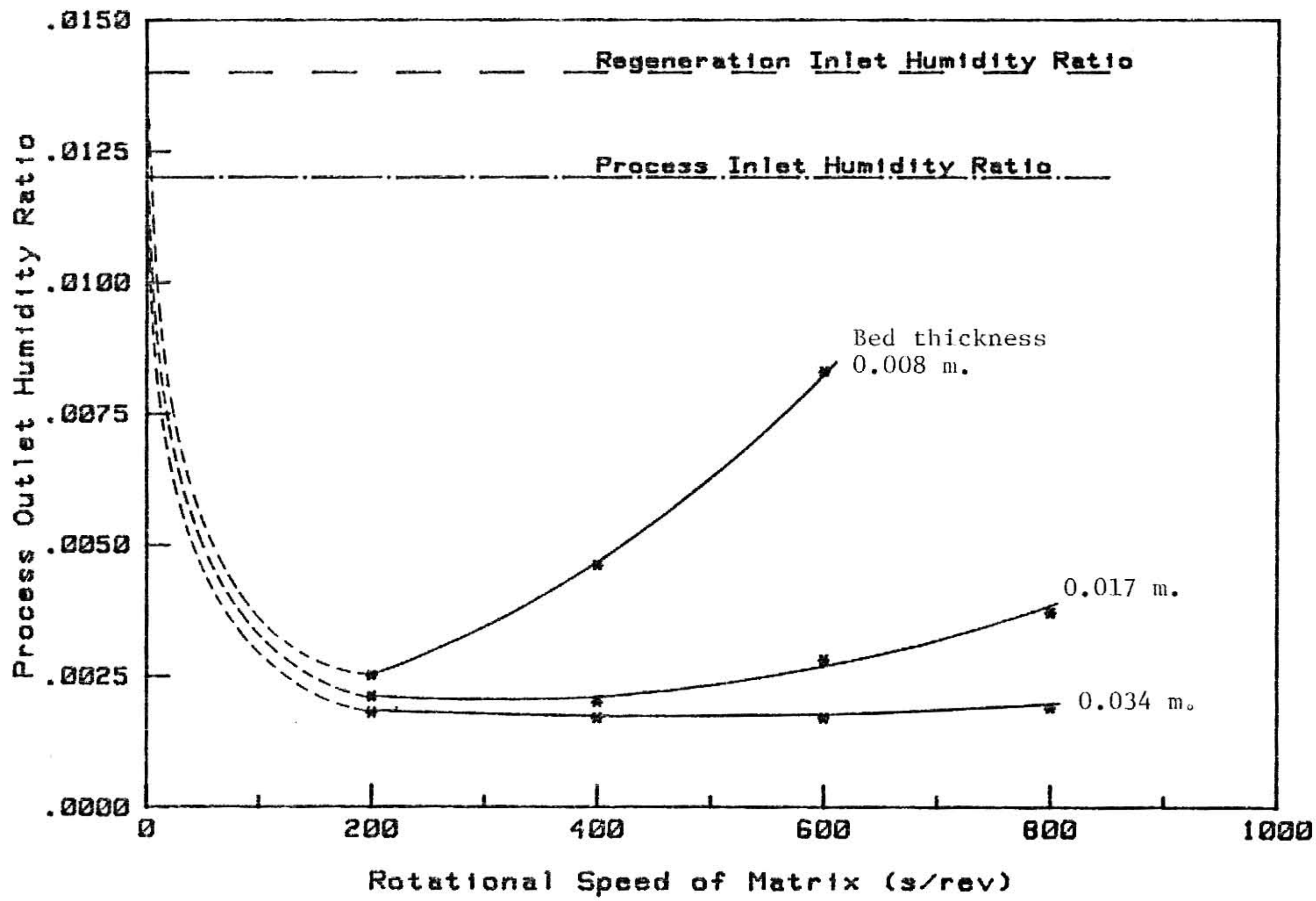

FIGURE 5-6 EFFECT OF VARIATION OF SPEED OF ROTATION

The two streams could get completely mixed at such high speeds due to carry over and so that the plots are extrapolated to converge at the point

$$\frac{W_{process\ in} + W_{regn.\ in}}{2}$$

Thus the resulting state on both the process and regeneration sides will be the mean of the two states.

Figure 5-7 brings out the consequences of increasing the face velocity on the desiccant bed. When the mass flow rate is such that the velocity of air through the bed is low, there is adequate contact time between the desiccant and the fluid to have an effective dehumidification. At higher velocities, thinner beds become less effective than thicker ones. A closer observation of the curve for a bed of 51mm shows that an optimum velocity of air is about 1.5 mts/sec, and when we move away on either side , the dehumidification performance deteriorates. It is possible that at this velocity, the contact time is not too small to result in insufficient dehumidification nor too large that the increase in bed temperature due to adsorption actually acts against dehumidification process.

Finally, the effect of bed thickness on the performance is presented in Figure 5-8, for two different flow rates. The trend in both the cases is the same; as bed thickness increases the dehumidification performance becomes better, but for large bed thickness, there is actually a small decrease in the dehumidification effect. This is because, thicker beds tend to dehumidify air as close to equilibrium moisture content as possible, but due to heat of adsorption, the beds become hotter. This inturn reduces the beds' ability to adsorb moisture, thus resulting in decreased performance.

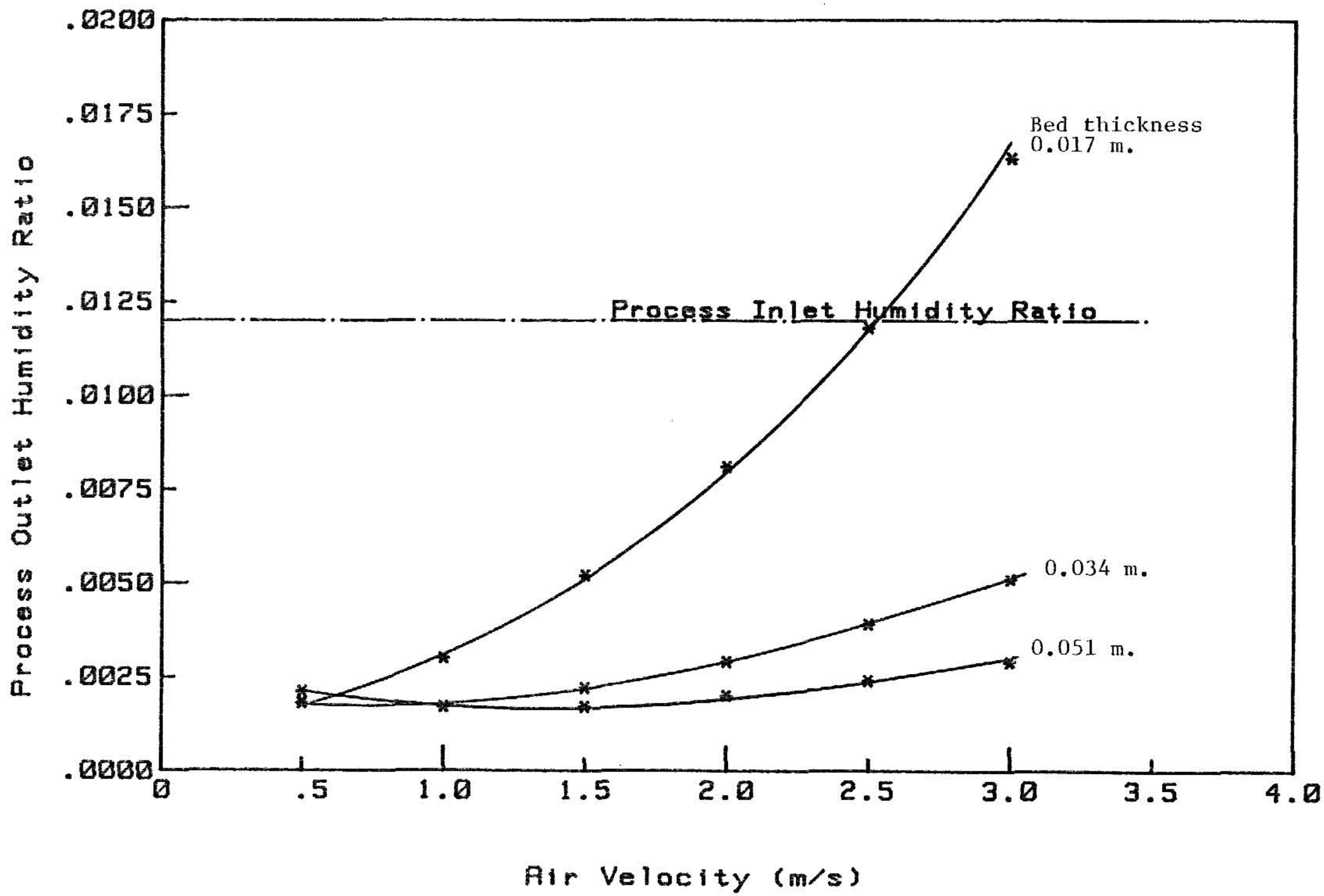

FIGURE 5-7 EFFECT OF VARIATION OF FACE VELOCITY ON DESICCANT

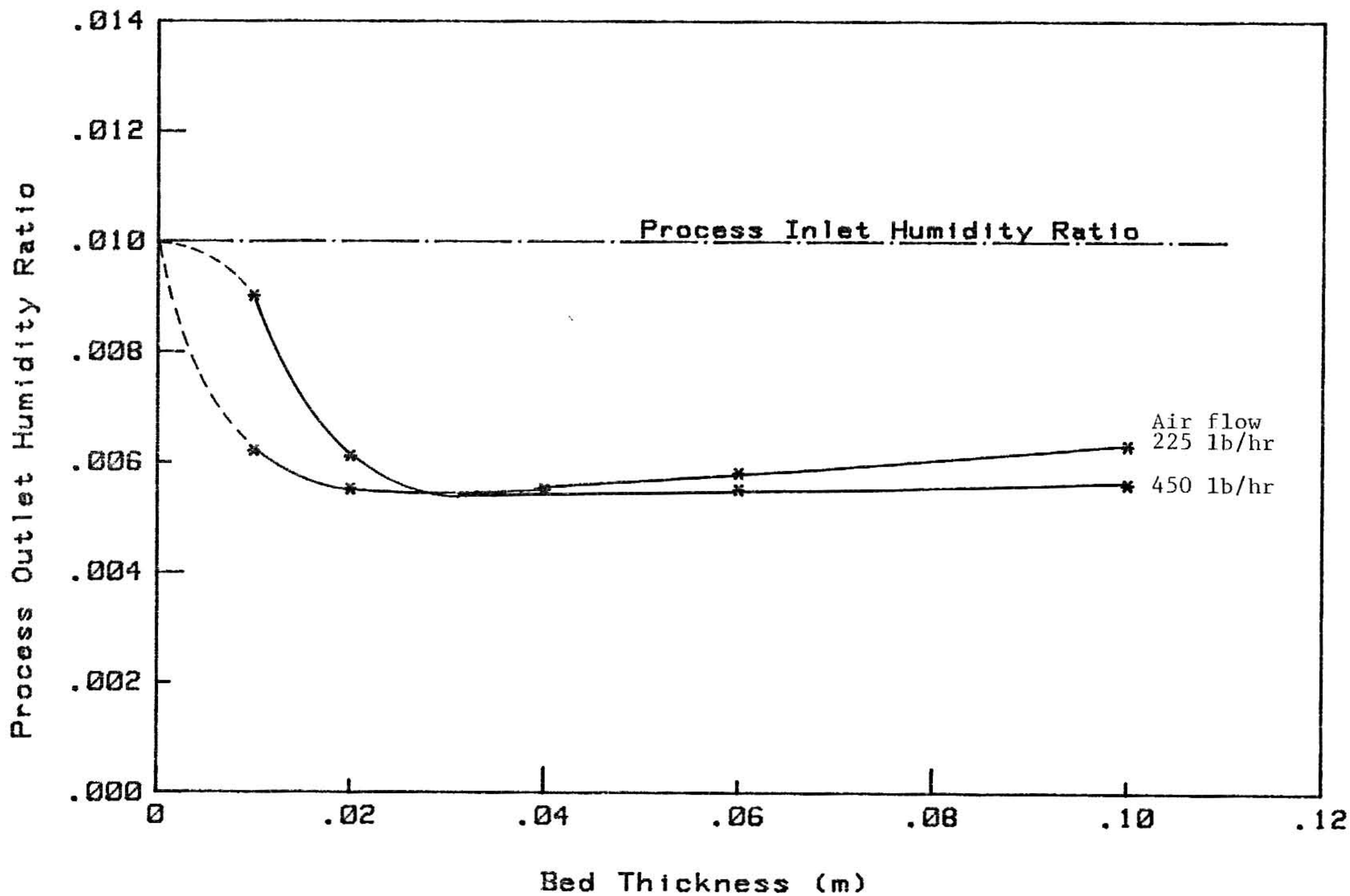

FIGURE 5-8 EFFECT OF VARTATION OF BED THICKNESS

CHAPTER VI

AIR CONDITIONING SYSTEM ANALYSIS

The fuel used for much of the air conditioning is using electricity generated by natural gas. And natural gas being the cleanest and most desirable fuel, a solar powered air conditioning system will go far to conserve it. Solar energy could be utilized in a mechanical refrigeration air conditioning system too. For instance, a conventional vapor compression refrigeration system can use a Rankine cycle to convert solar energy to mechanical power to drive the compressor. Or, in a vapor absorption system, refrigerant vapor can be driven off from a strong solution of the solvent in a generator with the help of solar energy, as the generation temperatures are within the temperature range attainable with flat plate collectors.

Although the technical feasibility of solar energy utilized refrigeration air conditioning systems has been established, their economic viabilities have not quite yet been established.

An all solar-powered desiccant air conditioning system can be contrived, if the desiccant is used to remove more moisture than is necessary to dehumidify a conditioned space, and the dry air is cooled by rehumidification. The layout of such a system, operating in vent and recirculation modes, was presented in Chapter II.

The purpose of this chapter is to study the attractiveness of using a solid sorption system with solar regeneration for air conditioning, as compared to a mechanical refrigeration system.

The present analysis assumes that the components that are contained in the system are ideal, i.e., the sensible heat exchangers, spray/

evaporative coolers, evaporator, condenser, etc., are 100% efficient. If, for such an ideal case, an analysis of the sorption system yields good performance evaluation parameters, it would definitely be worthwhile to introduce actual operative efficiencies in the system and analyze further.

The proposed air conditioning system is for a small residential application of about 4 tons (48,000 Btu/hr.) capacity, with an assumed sensible heat ratio of 70%. First, a sorption system would be analyzed and then a vapor compression refrigeration system using Freon 12 as the refrigerant is analyzed.

Nelson (8) has studied an adsorption air conditioning system for both vent and recirculation modes, and his conclusion was that the vent mode is slightly superior in performance than the recirculation mode. Although the layout of the components in the presently proposed system is different from his system, the current analysis is for vent mode of operation.

A study of Table 4-2 shows that, the temperature of air stream, after regeneration of the silica gel bed is well above ambient temperature. Hence, the present model is designed to utilize this waste heat to preheat the regeneration stream itself. This is expected to result in significant saving of energy.

With the above considerations, a model has been contrived. The schematic representation of the layout of various components is shown in Figure 6-1.

Manhattan, KS, is chosen as the location for analysis. The summer design conditions for this place are (10)

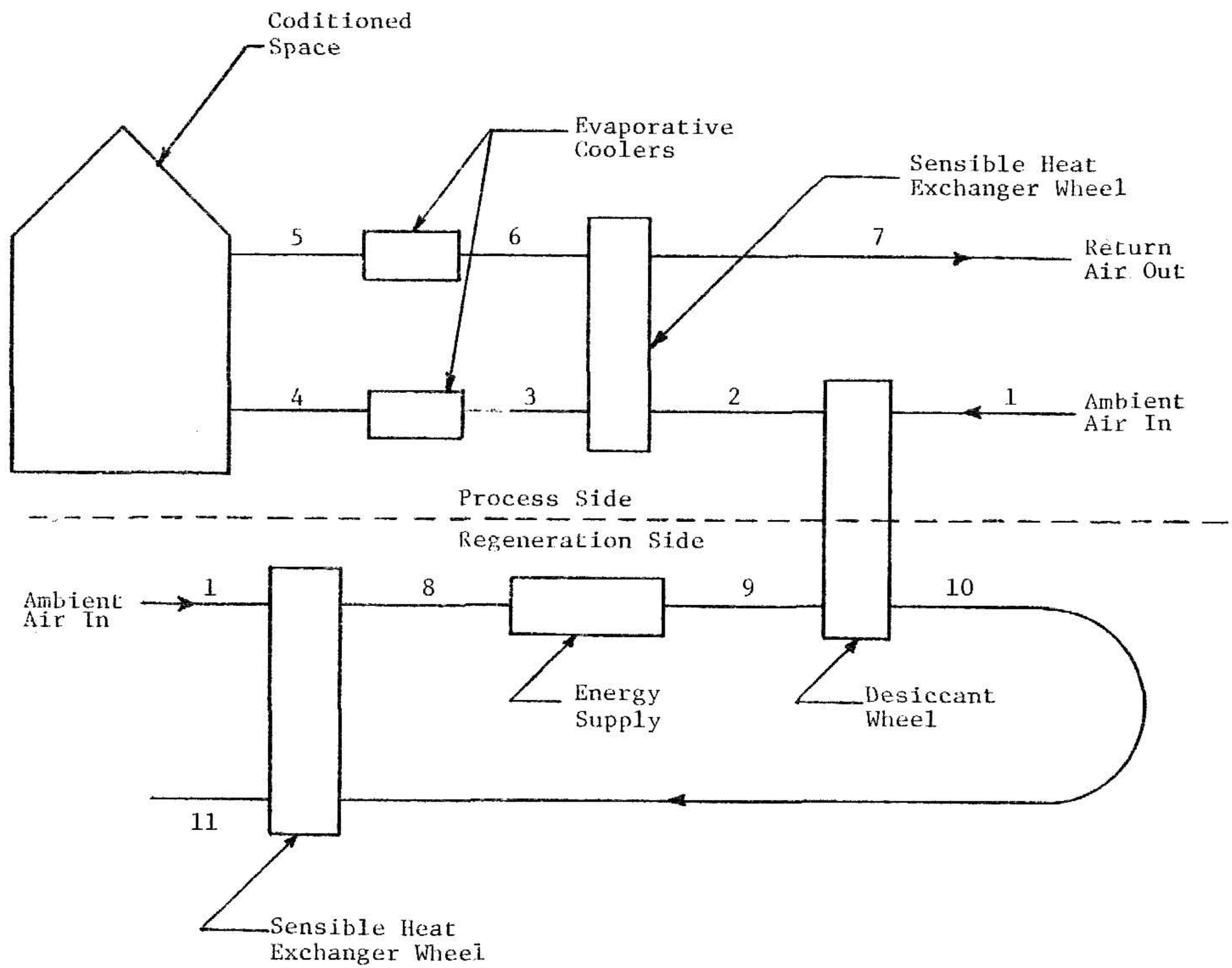

FIGURE 6-1 A SORPTION AIR CONDITIONING SYSTEM

$$T_{dry\ bulb} = 95^oF$$

$$T_{wet\ bulb} = 75^oF$$

The air conditioned space is to be maintained at 75°FDB, 50% RH. A preliminary analysis indicated that a bed of 5' diameter and 1" thick, would be of reasonable size. Allowing 5% of the face area of the bed for mounting and supporting structures, the net frontal flow area exposed to each stream is calculated to be 7.35 ft^2 (0.87 m^2). Leakages across and around the desiccant bed are assumed to be negligible.

The performance charts presented in Chapter 5 show that the regeneration temperature plays an important role in the dehumidification. A higher regeneration temperature could result in a better dehumidification of the process air. But beyond a certain limit, the auxiliary energy required to add that extra heat may not justify the extra dehumidification achieved. The equilibrium curves for silica gel (1) signify that at temperatures over 180°F, the decrease in equilibrium partial pressure of water vapor in silica gel for increase in temperatures is very marginal. Hence it was decided to set the upper limit of regeneration air temperature at 180°F.

In general, cooling load more or less parallels solar insolation; hence more solar energy is available for the regeneration stream when the load on the dehumidifier is high. However, if for example, the ambient temperature decreases at constant wet bulb temperature, a more severe latent load removal requirement is imposed on the system. Fortunately, the desiccant's capacity to remove moisture increases as the process air inlet temperature decreases, as may be observed from Figure 5-4. Also, the desiccant performance doesn't appear to get significantly altered with varying moisture content of the process air, Figure 5-5.

Mass flow rates on the process and regeneration streams are determined by confirming that the face velocity of air on the desiccant bed is such that the desiccant's performance is not impaired. One other factor that needs to be taken into account, is that the process stream flow rate confirms to ASHRAE recommendations based on noise/ comfort criteria (10). If the flow rate chosen is small, then, for the process air to pick up the load in the room, it needs to be at a low temperature. This would mean larger evaporative cooling. However, there appears to be no disadvantage in chosing large flow rate within the above mentioned limitations.

Based on the above, a flow rate of 2500 cfm was decided upon, leading to a mass flow rate of 173.2 lbm/min. This produced a face velocity of 1.81 m/sec. on the bed, which is in the optimum range, as seen from the performance curve (Figure 5-7).

The evaporative cooler in the return air line saturates the return air at 75°F DB 50% relative humidity to 62.5°F dry and wet bulb temperatures, and then led into the sensible heat regenerator.

The ambient air is sensibly heated to 180°F for regeneration of the silica gel. To reduce the problem of leakages, the flow rates are assumed to be balanced, and hence, we now know the inlet conditions of air to the silica gel on both process and regeneration sides. The computer program was run to obtain the outlet conditions of air of both streams. A copy of this output is presented in Table 6-1.

The ambient air, after dehumidification, is at 140°F and having a moisture content of 0.0065 lb_w/lb_a. This air is cooled in the

TABLE 6-1 COMPUTER OUTPUT FOR SORPTION AIR CONDITIONING SYSTEM ANALYSIS

Process side flow rate	: 1.3094 kg/s (10390 lb/hr)
Regeneration side flow rate	: 1.3094 kg/s (10390 lb/hr)
Matrix rotational speed	: .0157 rad/s (400 s/rev)
Thickness of bed/Frontal area	: .025 m / .87 m2
Pr. drop, proc./regn. sides	: 5.231 / 5.231 inches of water
Recirculation/Bypass leaks	: 0.000 / 0.000
Peripheral leak, proc./regn.	: 0.000 / 0.000

NOTE: Temperatures in Deg F

PROCESS SIDE PARAMETERS

	INLET CONDITIONS		OUTLET CONDITIONS	
HOUR ENDING	Temperature (Deg F)	Humidity Ratio (lb/lb)	Temperature (Deg F)	Humidity Ratio (lb/lb)
0	95.0	0.0142	140.1	0.0065

REGENERATION SIDE PARAMETERS

	INLET CONDITIONS		OUTLET CONDITIONS	
HOUR ENDING	Temperature (Deg F)	Humidity Ratio (lb/lb)	Temperature (Deg F)	Humidity Ratio (lb/lb)
0	180.0	0.0142	134.9	0.0234

sensible heat exchanger (effectiveness = 1.0) with the evaporatively cooled return air, to a dry bulb temperature of 62.5^{o}F.

The latent load removal rate by the air is computed to be 1.386 Btu/hr per pound of air circulated. From the psychrometric plot, it is found that the evaporative cooling that needs to be done to pick up this latent load is by cooling the air up to 56^{o}F, along the constant wet bulb temperature line, to result in a humidity ratio of 0.0082 lb_w/lb_a. The capacity of air to meet the sensible load is determined from the psychrometric plot, and it is found that the cooling air at this condition (56^{o}F dry bulb temperature and 0.0082 lb_w/lb_a humidity ratio) is capable of meeting sensible load of 4.4 Btu/hr per pound of air. Hence this system, operating with 173.2 lbm/min of air is calculated to be adequate to meet a total load of 60,125 Btu/hr with a sensible heat ratio of 79%. This overdesign will be offset towards the required duty of 48,000 Btu/hr when actual operating efficiencies are taken into consideration.

In the regeneration stream, the solar collector and auxiliary heater heat the ambient air, that has been preheated in the sensible heat exchanger, to 180^{o}F. With a sensible heat exchanger effectiveness of 1.0, the solar and auxiliary heat supplied will be to heat the air from 135^{o}F to 180^{o}F. Knowing the mass flow rate of air and its specific heat, the heating energy supplied by solar and/or auxiliary source is 112,234 Btu/hr.

Figure 6-2 is a psychrometric plot showing the states of air at various locations in the process.

With an overall collector efficiency of 50%, and with a solar insolation value obtained from (10) for 21st July, for a place

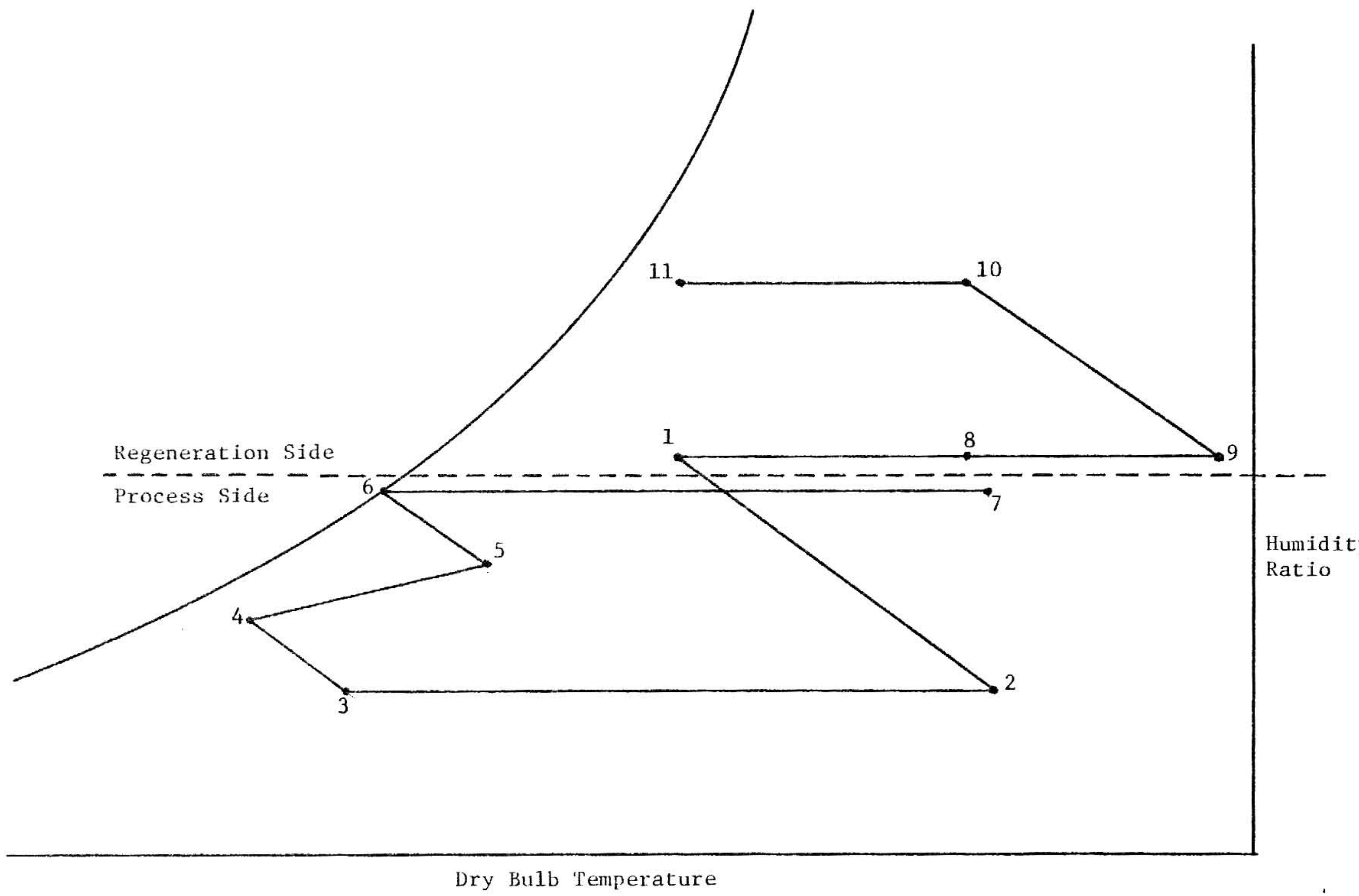

FIGURE 6-2 PSYCHROMETRIC PLOT FOR THE SORPTION AIR CONDITIONING SYSTEM

located on 40^{o} N latitude (Manhattan being at 39.2^{o} N latitude), the collector area required to meet the entire regeneration air heating is about 813 ft^2.

Turning to the mechanical refrigeration system now, we require it to cool ambient air at $95^{o}F$ dry bulb and $75^{o}F$ wet bulb temperatures to the saturation temperature corresponding to a humidity ratio of 0.0082 (which is the same moisture content as in the sorption case). This corresponds to a final dew point temperature of $52^{o}F$.

Basically, the vapor compression system has a compressor, condenser, an expansion valve and an evaporator. As mentioned earlier, in the present analysis, all the components are considered to be ideal.

In this case, a recirculation mode proved to be undoubtedly superior over the vent mode. Hence the return air from the air conditioned space at $75^{o}F$ and 50% relative humidity is cooled to the saturated condition at $52^{o}F$ in the evaporator and led back to the space. The cooling load is maintained as 60,125 Btu/hr, as was in the sorption case. But since air is supplied at saturated condition, unlike in the previous case where it was at about 90% relative humidity, the sensible heat ratio in the present case is about 83%.

The coolant fluid in the condenser is assumed to be the ambient air. And to enhance the performance of the system, an ideal spray cooler is used to cool the ambient air to its saturation temperature, adiabatically, to $75^{o}F$, before it is fed to the condenser. Such an arrangement of the vapor compression air conditioning system can be seen in Figure 6-3.

Since supply air is required at $52^{o}F$, and an ideal evaporator is considered, the refrigerant exits the evaporator as a saturated vapor at 52^{o}.

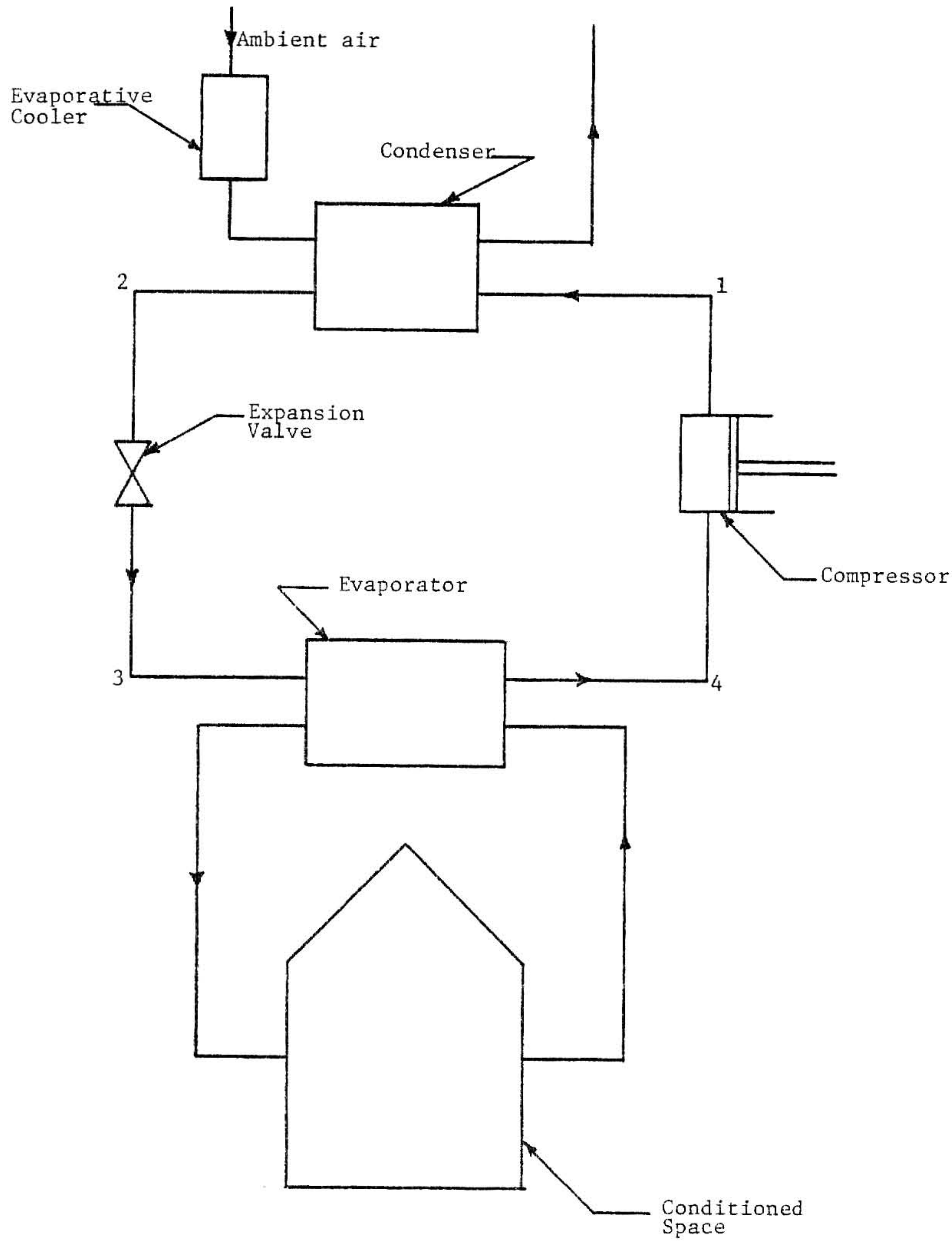

FIGURE 6-3 LAYOUT OF A MECHANICAL REFRIGERATION AIR CONDITIONING SYSTEM

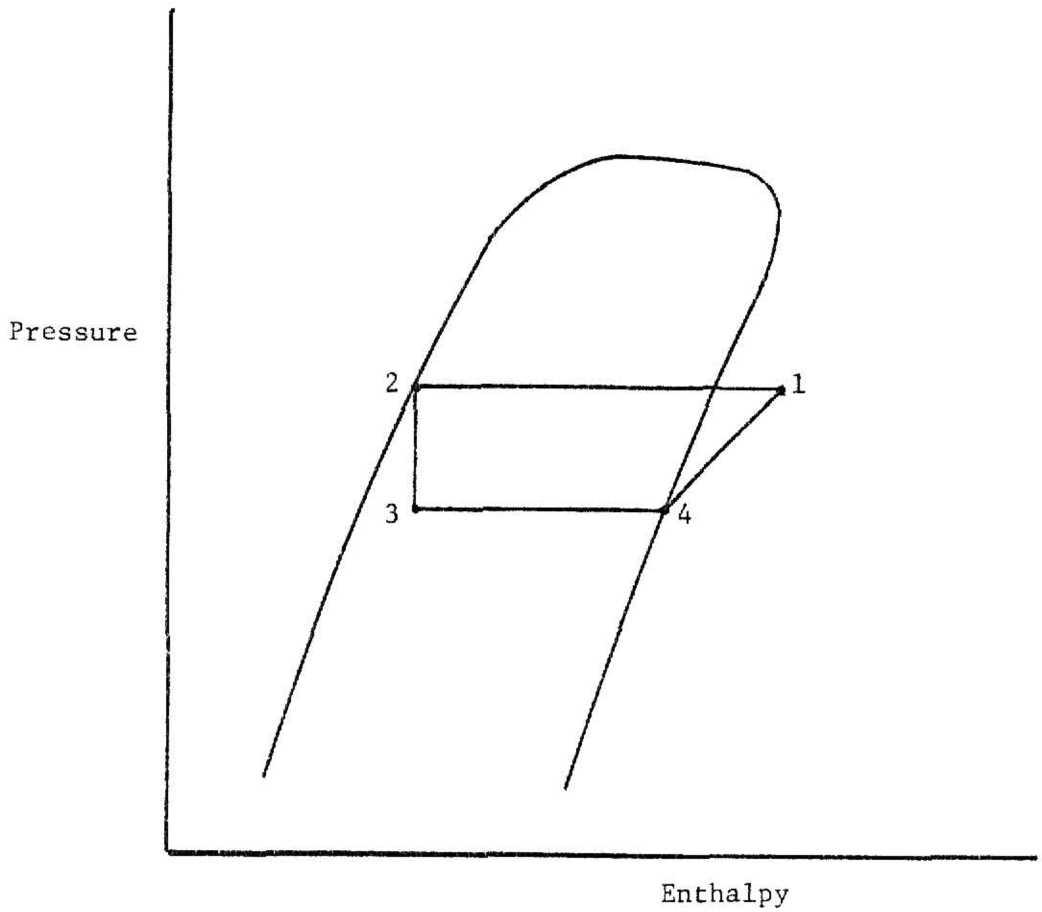

FIGURE 6-4 PRESSURE - ENTHALPY PLOT FOR THE MECHANICAL REFRIGERATION AIR CONDITIONING SYSTEM

TABLE 6-2 MECHANICAL REFRIGERATION AIR CONDITIONING SYSTEM ANALYSIS, COOLING AIR TEMPERATURE OF 75°F

Air Flow Rate (lbm/min)	Air Flow Rate (cfm)	R-12 Flow (lbm/min)	T_{cond}' (°F)	$T_{x,air}$" (°F)	T_{out}# (°F)	$T_{in,R12}$ (°F)	Cond. pres. (psia)	$H_{in,R-12}$ (Btu/min)	Compressor work (Btu/min)	COP
550	7640	18.11	84.0	83.0	83.2	93.0	104.9	87.0	79.1	12.1
500	6940	18.11	84.0	83.8	84.0	93.0	104.9	87.0	79.1	12.7
450	6250	18.19	85.0	84.8	85.1	96.0	106.5	87.3	85.0	11.8
400	5550	18.34	87.0	86.1	86.5	99.0	109.6	87.8	94.8	10.6
350	4860	18.42	88.0	87.7	88.1	100.0	111.2	87.9	97.1	10.3
300	4170	18.59	90.0	89.8	90.3	101.0	114.5	88.0	99.8	10.0
250	3470	18.83	93.0	92.9	93.5	103.0	119.5	88.2	109.6	9.6
200	2780	19.27	98.0	97.6	98.3	108.0	128.2	88.7	116.9	8.6
150	2090	20.01	106.0	105.6	106.9	120.0	143.1	89.8	143.5	7.0

'T_{cond} is the condensation temperature of the refrigerant.

"$T_{x,air}$ is the temperature of air after it has picked up the heat of condensation of R-12.

#T_{out} is the temperature of the air at the exit of the condenser.

The required process air flow rate can be easily found to be 145.2 lbm/min, as the cooling load as well as the conditions of air at inlet to and outlet from the conditioned space are known.

Enough parameters are not known to analyze the refrigerant cycle now. By a trial and error method, however, refrigerant flow rate and condenser pressure could be determined for a given cooling air flow rate. The procedure is to assume the condensation temperature of the refrigerant which results in a particular flow rate of the refrigerant to meet the evaporator load, as state 2 in Figure 6-4 gets fixed. Then the total heat of condensation and hence the temperature raise of the assumed flow of air can be calculated. For condensation of R-12 to take place, this temperature of air has to be below the condensation temperature of R-12. Once this is confirmed, the compressor work and the coefficient of performance of the cycle (defined as the ratio of refrigeration effect to the compressor work input, i.e. $(h_4 - h_3)/(h_1 - h_4)$, Figure 6-4) can be determined.

Table 6-2 lists the performance of the system for differing cooling air flow rates.

These figures may be compared with Table 6-3 which lists the same parameters for a condenser cooled with ambient air at 95°F without spray or evaporative cooling. Obviously, the former system is significantly superior due to a lower condenser operating temperature. A study of the above tables show that the performance of the system increases with the increasing cooling air flow rate. However, a large blower would be required to force a greater volume of air through the condenser. The above table does not bring this fact to light.

TABLE 6-3 MECHANICAL REFRIGERATION AIR CONDITIONING SYSTEM ANALYSIS, COOLING AIR TEMPERATURE OF 95^oF

Air Flow Rate (lbm/min)	Air Flow Rate (cfm)	R-12 Flow (lbm/min)	T_{cond}' (^oF)	$T_{x,air}$" (^oF)	T_{out} (^oF)	$T_{in,R12}$ (^oF)	Cond. pres. (psia)	$H_{in,R-12}$ (Btu/min)	Compressor work (Btu/min)	COP
500	7150	19.91	105.0	104.2	104.5	116.0	141.3	89.5	136.8	7.3
400	5720	20.11	107.0	106.5	106.1	118.0	145.1	89.8	144.2	7.0
300	4290	20.51	111.0	110.4	111.1	123.0	153.1	90.3	157.3	6.4
200	2860	21.38	119.0	118.5	119.7	130.0	170.1	91.2	183.2	5.5

'T_{cond} is the condensation temperature of the refrigerant.

"$T_{x,air}$ is the temperature of air after it has picked up the heat of condensation of R-12.

#T_{out} is the temperature of air at the exit of the condenser.

Now that ideal air conditioning systems operating on adsorption cycle and vapor compression cycle have been presented, the relative merits and de-merits of each could be studied. Due to lack of time, a detailed energy and economic analysis could not be carried out.

The following points are worth mentioning:

1. In the sorption system, the entire regeneration energy could be met with solar collectors of adequate area. In the present case, as mentioned earlier, a collector area of about 813 ft^2 could do the needful. Then, the only energy requirement would be for the process and regeneration air blowers and the drive motors for the sensible heat exchangers and dehumidifier beds.
2. The dehumidifier performance considered in the above analysis is the actual performance and not an ideal one. Further, the evaporative/spray coolers and the sensible heat exchangers used in an actual sorption system could have effectiveness of up to 0.99 (11) and 0.90 (8) respectively. Hence the values realized in actual practice are not expected to be much different from those presented earlier.
3. The vapor compression system involves compressor whose actual work input could be much greater than what has been specified above, because its efficiency seldom exceeds 0.6. This has a significant consequence, in that the refrigerant's inlet temperature to the condenser would increase requiring larger coolant flow rate, etc. Other factors that deteriorates the COP values specified above are due to the irreversibilities in the expansion valve, the slight amount of super heat desired at the exit of the evaporator and the operating efficiencies of the condenser and evaporator.

The intent of the above analysis was not to conclusively prove the superiority of one system over the other, but to demonstrate that a solar powered sorption system could still be a topic worth further study. As the need to conserve the high grade energy increases, and as more and more efficient components are designed, it appears that a solar powered sorption air conditioning system could prove to be a promising alternative.

CHAPTER VII

CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER STUDY

From this research, the following conclusions may be drawn:

1) The experimental system has been run with differing flow rates, temperatures, and humidity ratios on process and regeneration temperatures. The data gathered from the above experimental runs are compared with the results from the modified computer program and shown that the computer model predictions are reasonably accurate.
2) The performance curves generated with the help of the computer program allow the behavior of silica gel be observed, for variations in values of certain parameters.
3) The air conditioning system analysis has shown that a sorption dehumidification system with solar power for regeneration merits further study.

Following recommendations are made at the conclusion of this research, for further study:

1) An analytical method to arrive at the optimum values of desiccant bed parameters and flow rates of process and regeneration air streams to obtain a given set of process outlet conditions should be developed.
2) The air conditioning system needs to be analyzed with components that operate with realistic efficiencies.
3) The performance of the sorption air conditioning system for part load operations needs to be studied.
4) The operation of this system over a long period of time, say over a period of month or year need to be analyzed taking

into consideration the changing of the load with time of the day and of the year.

5) A sorption air conditioning system can be combined with mechanical refrigeration system and an analysis for optimum load shared by each system can be determined.

6) Possibilities of using the sorption air conditioning systems having banks of solar collectors, for winter heating can be investigated.

7) Performance of sorption systems for large scale applications can be studied.

8) An economic analysis of the dehumidification systems need to be performed.

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A P P E N D I X

Computer Model

```
1000  ! ***********************************************************************
1010  ! ########### NOMENCLATURE FOR PROGRAM INPUT PARAMETERS ##################
1020  !
1030  !
1040  ! Ac      - AREA OF THE COLLECTOR, m2
1050  ! Af      - TOTAL FRONTAL AREA EXPOSED TO FLOW, m2
1060  ! Along   - LONGITUDE AT COLLECTOR LOCATION, Deg
1070  ! Amassp  - MASS FLOW RATE, PROCESS SIDE, kg/s
1080  ! Amassr  - MASS FLOW RATE, REGENERATION SIDE, kg/s
1090  ! Av      - INTERNAL SURFACE AREA OF DESICCANT PER UNIT VOLUME, m2/m3
1100  ! Bleak   - BYPASS LEAKAGE RATE
1110  ! Ctype   - COLLECTOR TYPE
1120  ! Dd      - CHARACTERISTIC DIMENSION OF THE DESICCANT, m
1130  ! Dtcd    - TEMPERATURE DROP, COLLECTOR TO DESICCANT, DegC
1140  ! Dtdc    - TEMPERATURE DROP, DESICCANT TO COLLECTOR, DegC
1150  ! D$      - DATE
1160  ! Fl      - FLOW LENGTH IN THE DESICCANT (THICKNESS OF THE MATRIX), m
1170  ! Fric    - FRICTION FACTOR OF THE BED
1180  ! H       - INSOLATION ON A HORIZONTAL PLANE, kJ/hr-m2
1190  ! Hmu     - AIR TO DESICCANT MASS RATIO IN DESICCANT WHEEL
1200  ! Idate   - NUMBER OF DATE IN THE YEAR
1210  ! Iflag   - FLAG TO FIX REGENERATION TEMPERATURES
1220  ! Ihr     - HOURS OF ANALYSIS
1230  ! Nd      - NUMBER OF DAYS OF ANALYSIS
1240  ! Nh      - NUMBER OF HOURS OF ANALYSIS
1250  ! Perb    - PERCENTAGE BEAM RADIATION
1260  ! Perd    - PERCENTAGE DIFFUSE RADIATION
1270  ! Pleakp  - PERIPHERAL LEAKAGE, PROCESS SIDE
1280  ! Pleakr  - PERIPHERAL LEAKAGE, REGENERATION SIDE
1290  ! Phi     - LATITUDE OF LOCATION, DegC
1300  ! Reflec  - PERCENTAGE REFLECTED RADIATION
1310  ! Rhof    - DENSITY OF AIR STREAM, kg/m3
1320  ! Rleak   - RECIRCULATION LEAKAGE
1330  ! Rps     - ROTATIONAL SPEED OF MATRIX, rev/s
1340  ! Sl      - SLOPE OF THE COLLECTOR, Deg
1350  ! T       - AMBIENT TEMPERATURE, DegC
1360  ! Tidp    - PROCESS STREAM INLET TEMPERATURE, DegC
1370  ! Tidr    - REGENERATION STREAM INLET TEMPERATURE, DegC  (Iflag #1)
1380  ! Tmin    - MINIMUM REGENERATION TEMPERATURE, DegC    (Iflag #2)
1390  ! U       - EFFECTIVE CONDUCTANCE BETWEEN AIR STREAM AND DESICCANT
1400  ! V       - VELOCITY OF AMBIENT AIR, m/s
1410  ! Widp    - PROCESS STREAM INLET HUMIDITY RATIO
1420  ! Widr    - REGENERATION STREAM INLET HUMIDITY RATIO
1430  ! Xe      - EFFECTIVENESS OF REGENERATOR
1440  ! ***********************************************************************
1441  !
1450  PRINTER IS 0
1460  ! ***********************************************************************
1470  ! DATA INPUT HERE
1480  ! ***********************************************************************
1490  DATA 0,1,6,1                                  ! Ctype,Iflag,Nh,Nd
1500  DATA 0.01778,.02836,3.172,39.42,39.42,96.6 ! Amassp,Amassr,Ac,Sl,Phi,Along
1510  DATA 0,1,.8,.2,.2                             ! Dtdc,Dtcd,Perb,Perd,Reflec
1520  DATA 1962,.017,.0364,.75,.0016667,1.11,65   ! Av,Fl,Af,Void,Rps,Rhof,Tmin
1530  DATA 1281,.0031,.1463,592,.40,.20,.20,.40,0! U,Dd,Fric,Hmu,Rleak,Pleakp,Pl
eakr,Bleak,Xe
1540  DATA "JUNE 19, 1982 "    ! D$
1550  DATA 170                 ! Idate(I)
1560  DATA 12,13,14,15,16,17   ! Ihr(I,J)
1570  DATA 2391,3143,3089,2931,2402,1856   ! H(I,J)
1580  DATA 24.9,27.4,28.9,28.3,27.5,26.9   ! T(I,J)
1590  DATA 2.23,1.79,2.23,1.95,1.84,2.06   ! V(I,J)
1600  DATA .0159,.0161,.0156,.0154,.0146,.0135   ! Widp(I,J)
1610  DATA .0078,.0087,.0081,.0074,.0079,.0077   ! Widr(I,J)
1620  DATA 32.3,34.7,35.6,36.9,35.8,35.8         ! Tidp(I,J)
1630  DATA 75.1,80.6,79.8,80.8,72.8,67.0         ! Tidr(I,J)
```

```
1640  ! ************************************************************************
1650  !
1660  ! MAIN PROGRAM BEGINS
1670  !
1680  ! ************************************************************************
1690  STANDARD
1700  DIM Tidp(1,24),Widp(1,24),Todp(1,24),Wodp(1,24),Toxp(1,24),Toxr(1,24),Tidr
(1,24),Tord(1,24),Word(1,24)
1710  DIM Fp(1,24),Tqhex(1,24),Tqduct(1,24),Tqu(1,24),Tqaux(1,24),Tqsurp(1,24),T
qdehu(1,24)
1720  DIM Tfd(1,24),Trd(1,24),Fr(1,24),Todr(1,24),Wodr(1,24),Widr(1,24)
1730  DIM Woxr(1,24),Woxp(1,24),D$(1)[25]
1740  COM Wnolkp(1,24),Tnolkp(1,24),Wnolkr(1,24),Tnolkr(1,24)
1750  COM Cp,Ac,Len,Tau,Alpa,Ncovs,Epp,Epg,D,Width,Amu,Ak,Del,W,Cb,Id,Hfilmi,Kfi
n,Ub,Qduct(1,24),Ctype,Iflag,Amassp,Amassr,Wxe,Phi,Along,E,Sl,Dtdc,Dtcd,Od,Aflow
1760  COM Shr,Squ,Seff,Sqtl,Sqaux,Sqsurp,R(1,24),Qaux(1,24),Qtl(1,24),Qsurp(1,24
),Al,Fl,Void,Rps,Rhof,U,Dd,Fric,Delpp,Delpr
1770  COM Alfa(4,2),Sig(2,2),Gam(2,2),Bleak,Pleakp,Pleakr,Rleak,Ctau,Cal,Pa,Aff,
Alf(4,2),Gama(4,2),Kount1,Kount2,Qdehu(1,24),Qhex(1,24)
1780  COM Perb,Perd,Reflec,Omega(1,24),Af,Av,Hmu,Enu,Wmass
1790  COM T(1,24),V(1,24),H(1,24),Tmin(1,24),Ihr(1,24),Tic(1,24),Toc(1,24),Eff(1
,24),Qu(1,24)
1800  Cp=1005
1810  READ Ctype,Iflag,Nh,Nd
1820  READ Amassp,Amassr,Ac,Sl,Phi,Along
1830  READ Dtdc,Dtcd,Perb,Perd,Reflec
1840  READ Av,Fl,Af,Void,Rps,Rhof,Tmin
1850  READ U,Dd,Fric,Hmu,Rleak,Pleakp,Pleakr,Bleak,Xe
1860  FOR I=1 TO Nd
1870       READ D$(I)
1880       READ Idate(I)
1890       FOR J=1 TO Nh
1900            READ Ihr(I,J)
1910       NEXT J
1920       FOR J=1 TO Nh
1930            READ H(I,J)
1940       NEXT J
1950       FOR J=1 TO Nh
1960            READ T(I,J)
1970       NEXT J
1980       FOR J=1 TO Nh
1990            READ V(I,J)
2000       NEXT J
2010       FOR J=1 TO Nh
2020            READ Widp(I,J)
2030       NEXT J
2040       FOR J=1 TO Nh
2050            READ Widr(I,J)
2060       NEXT J
2070       FOR J=1 TO Nh
2080            READ Tidp(I,J)
2090       NEXT J
2100       IF Iflag>1 THEN 2140
2110       FOR J=1 TO Nh
2120            READ Tidr(I,J)
2130       NEXT J
2140       FOR J=1 TO Nh
2150            IF Iflag=2 THEN Tidr(I,J)=Tmin
2160 CALL Humid(Tidp(I,J),Tidr(I,J),Widp(I,J),Todp(I,J),Todr(I,J),Wodr(I,J),Wodp
(I,J),Widr(I,J),Xe)
2170 CALL Colper(J,I,T(I,J),V(I,J),H(I,J),Tidp(I,J),Tih(I,J),Hr(I,J),Eff,Qu(I,J)
,Ul(I,J),Ut(I,J),Fp(I,J),Fr(I,J),Tic(I,J),Toc(I,J),Ihr(I,J),Idate(I),Tidr(I,J))
2180            Todp(I,J)=Todp(I,J)+273.16
2190            Todr(I,J)=Todr(I,J)+273.16
2200            Haidp=Cp*Tidp(I,J)+1061*1005.7*2.205*Widp(I,J)
2210            Haodp=Cp*Todp(I,J)+1061*1005.7*2.205*Wodp(I,J)
```

```
2220           Qdehu(I,J)=Amassp*(Haodp-Haidp)
2230           Tqdehu(I,J)=3.6*Qdehu(I,J)
2240           Qhex(I,J)=0
2250           Tqhex(I,J)=3.6*Qhex(I,J)
2260           Tqduct(I,J)=3.6*Qduct(I,J)
2270           Tqu(I,J)=3.6*Qu(I,J)
2280           Tqaux(I,J)=3.6*Qaux(I,J)
2290           Tqsurp(I,J)=3.6*Qsurp(I,J)
2300        NEXT J
2310   NEXT I
2320   ! ******************************************************************
2330   !
2340   ! OUTPUT
2350   !
2360   ! ******************************************************************
2370   FOR I=1 TO Nd
2380       PRINT PAGE
2390       PRINT LIN(5),"                        Analysis of data taken on ";D$(I)
2400       PRINT
2410       PRINT "          Process side flow rate            :";Amassp;"Kg/sec ("
;DROUND(Amassp*2.2046*3600,4);"Lbs/hr)"
2420       PRINT
2430       PRINT "          Regeneration side flow rate       :";Amassr;"Kg/sec ("
;DROUND(Amassr*2.2046*3600,4);"Lbs/hr)"
2440       PRINT
2450       PRINT USING 2460;"Matrix rotational speed            :";Rps*2*PI;"Rad/se
c (";DROUND(1/Rps,4);"Sec/rev)"
2460       IMAGE 10X,34A,X,.DDDD,X,9A,DDD,X,8A
2470       PRINT
2480       PRINT "          Thickness of bed                  :";F1;"Mts"
2490       PRINT
2500       FIXED 3
2510       PRINT "          Pr. drop, proc./regn. sides       :";Delpp/248.84;"/";
Delpr/248.84;" inches of water"
2520       PRINT
2530       PRINT "          Recirculation/Bypass leaks        :";Rleak;"/";Bleak
2540       PRINT
2550       PRINT "          Peripheral leak, proc./regn.      :";Pleakp;"/";Pleakr
2560       PRINT LIN(1);"          ---------------------------------------------
---------------------"
2570        PRINT "                                   PROCESS SIDE PARAMET
ERS"
2580        PRINT "                          INLET CONDITIONS             OUTLET
 CONDITIONS"
2590        PRINT "          HOUR  Temperature     Humidity Ratio    Temperature
 Humidity Ratio"
2600        PRINT "          ENDING   (Deg C)          (Kg/Kg)          (Deg C)
   (Kg/Kg)"
2610        PRINT "          ---------------------------------------------------
---------------"
2620        FOR J=1 TO Nh
2630             PRINT
2640             PRINT USING 2650;Ihr(I,J),Tidp(I,J),Widp(I,J),Todp(I,J),Wodp(I,J
)
2650             IMAGE 11X,DD,6X,DDD.D,11X,Z.DDDD,11X,DDD.D,9X,Z.DDDD
2660        NEXT J
2670        PRINT "          ---------------------------------------------------
---------------"
2680        PRINT LIN(1);"          ---------------------------------------------
---------------------"
2690        PRINT "                                   REGENERATION SIDE PAR
AMETERS"
2700        PRINT "                          INLET CONDITIONS             OUTLET
 CONDITIONS"
2710        PRINT "          HOUR  Temperature     Humidity Ratio    Temperature
 Humidity Ratio"
```

```
2720        PRINT "          ENDING   (Deg C)          (Kg/Kg)          (Deg C)
   (Kg/Kg)"
2730        PRINT "          ---------------------------------------------------
---------------"
2740        FOR J=1 TO Nh
2750              PRINT
2760              PRINT USING 2650;Ihr(I,J),Tidr(I,J),Widr(I,J),Todr(I,J),Wodr(I,
J)
2770        NEXT J
2780        PRINT "          ---------------------------------------------------
---------------"
2790        PRINT PAGE
2800        PRINT LIN(2);"TOTAL HOURLY HEAT BALANCE MAP                    DATA AN
ALYSIS FOR ";D$(I)
2810        PRINT LIN(1);"Rates in Watts, Totals in KJ                        Heat
 Exchanger Efficiency=0"
2820        PRINT LIN(1);"----------------------------------------------------
-------------------------"
2830        PRINT "        Heat trans.      Heat lost    Collector       Aux. hea
t   Surplus solar"
2840        PRINT " Hour   in Dehumid.      in Ducting    Heat gain        Supplie
d   Heat Available"
2850        PRINT "Ending Rate    Total    Rate   Total  Rate    Total    Rate  Tot
al   Rate   Total "
2860        PRINT "----------------------------------------------------------------
------------------"
2870        PRINT
2880        FOR J=1 TO Nh
2890             PRINT USING 2900;Ihr(I,J),Qdehu(I,J),Tqdehu(I,J),Qduct(I,J),Tqdu
ct(I,J),Qu(I,J),Tqu(I,J),Qaux(I,J),Tqaux(I,J),Qsurp(I,J),Tqsurp(I,J)
2900             IMAGE X,DD,3X,DDD.D,3X,DDDD.D,2X,DDD.D,2X,DDD.D,2X,DDDD.D,2X,DDD
D.D,X,DDDD.D,2X,DDDD.D,2X,DDD.D,2X,DDD.D
2910        NEXT J
2920        PRINT "----------------------------------------------------------------
------------------"
2930        PRINT LIN(4);"HOURLY PERFORMANCE OF THE COLLECTOR"
2940        PRINT LIN(1);"Collector Type #0 - Manufacturer's Performance Curves I
nput"
2950        PRINT LIN(1);"----------------------------------------------------
-------------------------"
2960        PRINT "       Inlet Temp.    Outlet Temp.    Useful Rate
  Factor  Factor"
2970        PRINT " Hour  to Collector   from Collector  of Heat Gain  Efficiency
  FrU)    FrTa"
2980        PRINT "Ending   (Deg C)          (Deg C)          (Watts)"
2990        PRINT "----------------------------------------------------------------
------------------"
3000        PRINT
3010        FOR J=1 TO Nh
3020             PRINT USING 3030;Ihr(I,J),Tic(I,J),Toc(I,J),Qu(I,J),Eff(I,J),Fp(
I,J),Fr(I,J)
3030             IMAGE 2X,DD,7X,DD.D,11X,DDD.D,10X,DDDD.D,8X,Z.DDD,4X,Z.DDDD,3X,Z
.DDDD
3040        NEXT J
3050        PRINT "----------------------------------------------------------------
------------------"
3060        PRINT LIN(4);"HOURLY SUMMARY OF AMBIENT CONDITIONS"
3070        PRINT LIN(1);"----------------------------------------------------
-------------------------"
3080        PRINT " Hour    Ambient        Wind         Omega      Radiation
   R       HR"
3090        PRINT "Ending  Temp.(DegC)    Speed(M/s)     (Deg)        (W/M2)
       (W/M2)"
3100        PRINT "----------------------------------------------------------------
-----------------"
3110        PRINT
```

```
3120       FOR J=1 TO Nh
3130             PRINT USING 3140;Ihr(I,J),T(I,J),V(I,J),Omega(I,J),H(I,J),R(I,J
),H(I,J)*R(I,J)
3140             IMAGE 2X,DD,7X,DD.D,10X,DD.D,9X,MDD.D,7X,DDDD.D,8X,D.DDD,3X,DDD
D.D
3150       NEXT J
3160       PRINT "-------------------------------------------------------------
-----------------"
3170  NEXT I
3180  END
3190  ! ********************************************************************
3200  !
3210  ! SUBROUTINE HUMID BEGINS HERE
3220  !
3230  ! ********************************************************************
3240  SUB Humid(T1,T2,W1,Tout,Tex,Wex,Wout,W2,Xe)
3250  COM Wnolkp(*),Tnolkp(*),Wnolkr(*),Tnolkr(*)
3260  COM Cp,Ac,Len,Tau,Alpa,Ncovs,Epp,Epg,D,Width,Amu,Ak,Del,W,Cb,Id,Hfilmi,Kfi
n,Ub,Qduct(*),Ctype,Iflag,Amassp,Amassr,Wxe,Phi,Along,E,Sl,Dtdc,Dtcd,Od,Aflow
3270  COM Shr,Squ,Seff,Sqtl,Sqaux,Sqsurp,R(*),Qaux(*),Qtl(*),Qsurp(*),Al,Fl,Void
,Rps,Rhof,U,Dd,Fric,Delpp,Delpr
3280  COM Alfa(*),Sig(*),Gam(*),Bleak,Pleakp,Pleakr,Rleak,Ctau,Cal,Pa,Aff,Alf(*)
,Gama(*),Kount1,Kount2,Qdehu(*),Qhex(*)
3290  COM Perb,Perd,Reflec,Omega(*),Af,Av,Hmu,Enu,Wmass
3300  COM T(*),V(*),H(*),Tmin(*),Ihr(*),Tic(*),Toc(*),Eff(*),Qu(*)
3310  Itnum=0
3320  FOR I=1 TO 2
3330      FOR J=1 TO 2
3340         Sig(I,J)=0
3350      NEXT J
3360  NEXT I
3370  Tau=1/(2*Rps)
3380  Velp=Amassp/(Rhof*Af*Void)
3390  Velr=Amassr/(Rhof*Af*Void)
3400  Stdf=.06
3410  Inc=20
3420  Nn=1
3430  !
3440  ! !!!!!PROPERTIES AT STATE 1
3450  !
3460  N=1
3470  FOR I=1 TO 2
3480       Inci=1
3490       CALL Alfav(T1,W1,I,N,Inci,Alfa(N,I))
3500  NEXT I
3510  !
3520  ! !!!!! PROPERTIES AT STATE 2
3530  !
3540  N=2
3550  FOR I=1 TO 2
3560       CALL Alfav(T2,W2,I,N,Inci,Alfa(N,I))
3570  NEXT I
3580  !
3590  ! !!!!! ESTIMATE POINT 3
3600  !
3610  W3=(Alfa(1,2)*W1-Alfa(2,1)*W2+T1-T2)/(Alfa(1,2)-Alfa(2,1))
3620  T3=-Alfa(2,1)*(W3-W2)+T2
3630  !
3640  ! !!!!!ESTIMATE POINT 4
3650  !
3660  W4=(Alfa(1,1)*W1-Alfa(2,2)*W2+T1-T2)/(Alfa(1,1)-Alfa(2,2))
3670  T4=-Alfa(2,2)*(W4-W2)+T2
3680  !
3690  ! !!!!! DETERMINE TRUE POINTS 3 AND 4
3700  !
3710  Delt23=(T3-T2)/Inc
```

```
3720  Delt13=(T3-T1)/Inc
3730  T31=T1
3740  T32=T2
3750  W31=W1
3760  W32=W2
3770  Ts1=T1
3780  Ws1=W1
3790  Ts2=T2
3800  Ws2=W2
3810  Inc=3*Inc
3820  Nnn=3
3830  Nn=2
3840  Indic=0
3850  FOR L1=1 TO Inc
3860        IF Indic=2 THEN 3900
3870        N=1
3880        I=2
3890        CALL Egv(Ts1,Ws1,I,L1,N,Nnn,Delt13,Nn,T31,W31)
3900        N=2
3910        I=1
3920        IF Indic=1 THEN 3940
3930        CALL Egv(Ts2,Ws2,I,L1,N,Nnn,Delt23,Nn,T32,W32)
3940        Indic=1
3950        !
3960        ! !!!!! CHECK TO ASSURE EQUAL PROGRESSION FROM BOTH SIDES
3970        !
3980        IF W31>W32 THEN Indic=2
3990        IF T32<T31 THEN 4040
4000  NEXT L1
4010  !
4020  ! !!!!! NOTE: POINT 3 VALUES DO NOT CONSIDER ROTATING BED
4030  !
4040  W3=(Alfa(3,2)*W32-Alfa(3,1)*W31+T32-T31)/(Alfa(3,2)-Alfa(3,1))
4050  T3=-Alfa(3,1)*(W3-W32)+T32
4060  Gam(2,1)=Sig(2,1)/ABS(W31-W1)
4070  Gam(2,2)=Sig(2,2)/ABS(W32-W2)
4080  !
4090  ! !!!!! DETERMINE POINT 4 BY SIMILAR METHOD
4100  !
4110  Delt14=-Delt23
4120  Delt24=-Delt13
4130  T41=T1
4140  T42=T2
4150  W41=W1
4160  W42=W2
4170  Ts1=T1
4180  Ts2=T2
4190  Ws1=W1
4200  Ws2=W2
4210  Nnn=4
4220  Nn=1
4230  Indic=0
4240  FOR L1=1 TO Inc
4250        I=1
4260        N=1
4270        IF Indic=2 THEN 4290
4280        CALL Egv(Ts1,Ws1,I,L1,N,Nnn,Delt14,Nn,T41,W41)
4290        I=2
4300        N=2
4310        IF Indic=1 THEN 4330
4320        CALL Egv(Ts1,Ws1,I,L1,N,Nnn,Delt24,Nn,T42,W42)
4330        Indic=1
4340        !
4350        ! !!!!! CHECK TO ASSURE EQUAL PROGRESSION FROM BOTH POINTS
4360        !
4370        IF W41<W42 THEN Indic=2
```

```
4380          IF T42<T41 THEN 4430
4390  NEXT L1
4400  !
4410  ! !!!!! NOTE: POINT 4 VALUES DO NOT CONSIDER ROTATING BED
4420  !
4430  W4=(Alfa(4,2)*W42-Alfa(4,1)*W41+T42-T41)/(Alfa(4,2)-Alfa(4,1))
4440  T4=-Alfa(4,2)*(W4-W42)+T42
4450  Gam(1,1)=Sig(1,1)/ABS(W42-W2)
4460  Gam(1,2)=Sig(1,2)/ABS(W41-W1)
4470  !
4480  ! !!!!! NOW DETERMINE TRUE STATES OF POINTS 3 AND 4
4490  !
4500  Fle=37.244*Fl
4510  Aup=196.85*(Amassp/Rhof)/Af
4520  Aur=196.85*(Amassr/Rhof)/Af
4530  Delpep=(.0053*Aup+5.9E-5*Aup^2)*Fle
4540  Delper=(.0053*Aur+5.9E-5*Aur^2)*Fle
4550  Delpp=Delpep*248.84
4560  Delpr=Delper*248.84
4570  !
4580  ! !!!!! DETERMINE EFFECTIVENESS OF ROTATING DESICCANT WHEEL
4590  !
4600  E11=Stdf*Delpp*2*(1+Hmu*Gam(1,1))/(Rhof*Velp^2*Hmu*(Gam(1,1)+Gam(2,1)))
4610  E12=(1+Hmu*Gam(2,1))/(1+Hmu*Gam(1,1))
4620  E13=(1+Hmu*Gam(1,1))*Fl/(Tau*Velp)
4630  E21=Stdf*Delpr*2*(1+Hmu*Gam(1,2))/(Rhof*Velr^2*Hmu*(Gam(1,2)+Gam(2,2)))
4640  E22=(1+Hmu*Gam(2,2))/(1+Hmu*Gam(1,2))
4650  E23=(1+Hmu*Gam(1,2))*Fl/(Tau*Velr)
4660  Etacf2=(1-EXP(-E21*(1-E22)))/(1-E22*EXP(-E21*(1-E22)))
4670  Eta11=E13
4680  Eta12=E12*Eta11
4690  Eta21=Etacf2*(1-1/(9*E23^1.93))
4700  Eta22=Eta21*E22
4710  !
4720  ! !!!!! DETERMINE TRUE PROCESS OUTLET CONDITIONS
4730  !
4740  Tout=T1+Eta11*(T2-T41)+Eta21*(T41-T1)
4750  Wout=W1+Eta11*(W2-W41)+Eta21*(W41-W1)
4760  IF Wout<0 THEN Wout=0
4770  !
4780  ! !!!!! DETERMINE TRUE REGENERATION OUTLET CONDITIONS
4790  !
4800  Tex=T2+Eta12*(T1-T31)+Eta22*(T31-T2)
4810  Wex=W2+Eta12*(W1-W31)+Eta22*(W31-W2)
4820  Tnolkp(Kount1,Kount2)=Tout
4830  Wnolkp(Kount1,Kount2)=Wout
4840  Tnolkr(Kount1,Kount2)=Tex
4850  Wnolkr(Kount1,Kount2)=Wex
4860  !
4870  ! !!!!! CORRECT FOR LEAKAGE
4880  !
4890  CALL Leak(W1,W2,T1,T2,Wout,Wex,Tout,Tex,Wout,Wex,Tout,Tex,Xe)
4900  Time=2*Tau
4910  SUBEND
4920  ! ********************************************************************
4930  !
4940  !         SUBROUTINE EGV BEGINS HERE
4950  !
4960  ! ********************************************************************
4970  SUB Egv(Tc,Wc,I,L1,N,Nnn,Delt,Nn,Tcn,Wcn)
4980  COM Wnolkp(*),Tnolkp(*),Wnolkr(*),Tnolkr(*)
4990  COM Cp,Ac,Len,Tau,Alpa,Ncovs,Epp,Epg,D,Width,Amu,Ak,Del,W,Cb,Id,Hfilmi,Kfi
n,Ub,Qduct(*),Ctype,Iflag,Amassp,Amassr,Wxe,Phi,Along,E,Sl,Dtdc,Dtcd,Od,Aflow
5000  COM Shr,Squ,Seff,Sqtl,Sqaux,Sqsurp,R(*),Qaux(*),Qtl(*),Qsurp(*),Al,Fl,Void
,Rps,Rhof,U,Dd,Fric,Delpp,Delpr
5010  COM Alfa(*),Sig(*),Gam(*),Bleak,Pleakp,Pleakr,Pleak,Ctau,Cal,Pa,Aff,Alf(*)
```

```
,Gama(*),Kount1,Kount2,Qdehu(*),Qhex(*)
5020  COM Perb,Perd,Reflec,Omega(*),Af,Av,Hmu,Enu,Wmass
5030  COM T(*),V(*),H(*),Tmin(*),Ihr(*),Tic(*),Toc(*),Eff(*),Qu(*)
5040  !
5050  ! THIS PROGRAM DETERMINES INCRIMENTAL TEMP. & HUMIDITY VALUES FOR HUMID
5060  !
5070  Alfc=Alfa(N,I)
5080  IF Ll=1 THEN 5100
5090  Alfc=Alfa(Nnn,I)
5100  Tc=Tcn
5110  Wc=Wcn
5120  Tcn=Tc+Delt
5130  FOR Mm=1 TO 10
5140        IF Mm=1 THEN Alfcb=Alfc
5150        Wcn=Wc-(Tcn-Tc)/Alfcb
5160        IF Mm=1 THEN 5190
5170        Deltw=Wcn-Wco
5180        IF ABS(Deltw)/Wcn<=.05 THEN 5250
5190        Inci=1
5200        CALL Alfav(Tcn,Wcn,I,Nnn,Inci,Alfa(Nnn,I))
5210        Alfcn=Alfa(Nnn,I)
5220        Alfcb=(Alfcn+Alfc)/2
5230        Wco=Wcn
5240  NEXT Mm
5250  Alfa(Nnn,I)=Alfcb
5260  Tmean=(Tc+Tcn)/2
5270  Wmean=(Wc+Wcn)/2
5280  IF I=1 THEN Ii=2
5290  IF I=2 THEN Ii=1
5300  Inci=2
5310  CALL Alfav(Tmean,Wmean,Ii,Nn,Inci,Gam(Nn,Ii))
5320  Sig(Nn,Ii)=Gam(Nn,Ii)*ABS(Wcn-Wc)+Sig(Nn,Ii)
5330  SUBEND
5340  ! ********************************************************************
5350  !
5360  !  SUBROUTINE ALFAV BEGINS HERE
5370  !
5380  ! ********************************************************************
5390  SUB Alfav(T,W,I,Nn,Indi,Dummy)
5400  !
5410  ! !!!!! THIS SUPPLIES VALUES OF ALPHA FOR EGV
5420  !
5430  Tf=1.8*T+32
5440  Hfg=1095.4-.5877*Tf
5450  Tk=273.16+T
5460  ! DETERMINE SATURATION PRESSURE USING ASHRAE FORMULA
5470  Beta=647.27-Tk
5480  A1=3.24378
5490  A2=5.86826E-3
5500  A3=1.170238E-8
5510  A4=2.13785E-3
5520  Xpws=-(Beta/Tk)*((A1+A2*Beta+A3*Beta^3)/(1+A4*Beta))
5530  Pws=218.167*10^Xpws
5540  Tkn=Tk+.01
5550  Betan=Beta-.01
5560  Xpwsn=-(Betan/Tkn)*((A1+A2*Betan+A3*Betan^3)/(1+A4*Betan))
5570  Pwsn=218.167*10^Xpwsn
5580  Dpws=(Pwsn-Pws)/((Tkn-Tk)*1.8)
5590  Itnum=1
5600  IF W>0 THEN 5700
5610  Alf(Nn,1)=5E10
5620  Alf(Nn,2)=-1E10
5630  Gama(Nn,1)=.5
5640  Gama(Nn,2)=1
5650  GOTO 6340
5660  Rh=1
```

```
5670  Pw=Pws
5680  W=.62198*Pws/(1-Pws)
5690  GOTO 5760
5700  Pw=W/(.622+W)
5710  Rh=Pw/Pws
5720  IF Rh>1 THEN 5660
5730  !
5740  ! "X" IS THE SILICAGEL WATER CONTENT
5750  !
5760  X=Pw/(2.009*Pws)
5770  FOR K=1 TO 20
5780        !
5790        ! DETERMINE Rhh - RATIO OF HEAT OF SORPTION TO HEAT OF VAPORISATION
5800        !
5810        IF X<=.01 THEN Rhh=22.79*X+1
5820        IF (X>.01) AND (X<=.04) THEN Rhh=8533*X^3-542*X^2+11.79*X+1.1557
5830        IF (X>.04) AND (X<=.1) THEN Rhh=45.4*X^2-8.97*X+1.5922
5840        IF (X>.1) AND (X<=.35) THEN Rhh=-9.1553*X^3+7.048*X^2-1.8966*X+1.2789
5850        IF X>.35 THEN Rhh=-.58315*X+1.2894
5860        Pw=(29.91*2.009*X*Pws)^Rhh/29.91
5870        Wn=.622*Pw/(1-Pw)
5880        IF Itnum=1 THEN 5910
5890        IF ABS((Wn-W)/W)<=.05 THEN 6130
5900        !
5910        ! Drhh IS THE DERIVATIVE OF HS/HV WITH RESPECT TO X
5920        !
5930        IF X<=.01 THEN Drhh=22.79
5940        IF (X>.01) AND (X<=.04) THEN Drhh=25600*X^2-1084*X+11.79
5950        IF (X>.04) AND (X<=.1) THEN Drhh=90.8*X-8.97
5960        IF (X>.1) AND (X<=.35) THEN Drhh=-27.466*X^2+14.096*X-1.8966
5970        IF X>.35 THEN Drhh=-.58135
5980        Itnum=Itnum+1
5990        Zeta=LOG(29.91*2.009*Pws*X)*Drhh+Rhh/X
6000        Zeta=1/Zeta
6010        Dxdw=Zeta*(1-Pw)/Wn
6020        Xsave=X/2
6030        !
6040        ! !!!!! DETERMINE NEW VALUE OF SILICAGEL BY SIMPSON'S RULE
6050        !
6060        X=Dxdw*(W-Wn)+X
6070        IF X<0 THEN X=Xsave
6080  NEXT K
6090  PRINT "**********ERROR*******  NOT CONVERGING"
6100  !
6110  ! !!!!! DETERMINE (dt/dw) AT CONST X AND H
6120  !
6130  Alph=(Tf-32+Hfg)/(.24+W*.4123)
6140  Alpw=(Pw-1)*Pws/(W*Dpws*Rhh)
6150  Dhdx=Tf-32+Hfg*(1-Rhh)
6160  Dhadw=Tf-32+Hfg
6170  Alamda=1-Dhdx/Dhadw
6180  !
6190  ! !!!!! CALCULATE INTEGRAL OF (1-HS/HV)
6200  !
6210  IF X<=.01 THEN Sumr=-11.395*X^2
6220  IF (X>.01) AND (X<=.04) THEN Sumr=-2133.3*X^4+180.7*X^3-5.895*X^2-1.1577*X
+8.475E-4
6230  IF (X>.04) AND (X<=.1) THEN Sumr=-15.13*X^3+4.484*X^2-.5922*X+8.7731E-3
6240  IF (X>.1) AND (X<=.35) THEN Sumr=2.2889*X^4-2.3493*X^3+.9483*X^2-.2789*X-2
.0959E-4
6250  IF X>.35 THEN Sumr=.29158*X^2-1.2894*X
6260  Sigma=(.22+X-.5877*Sumr)/(.24+W*(1-.5877))
6270  Anu=Pws/(Zeta*Rhh*Dpws)
6280  Term1=Alph*Alamda+Sigma*Anu+Alpw
6290  Term2=(Term1^2-4*Alpw*Alamda*Alph)^.5
6300  Alf(Nn,1)=(Term1+Term2)/(2*1.8)
```

```
6310  Alf(Nn,2)=(Term1-Term2)/(2*1.8)
6320  Gama(Nn,1)=Sigma*Alpw/(Alpw-Alf(Nn,1)*1.8)
6330  Gama(Nn,2)=Sigma*Alpw/(Alpw-Alf(Nn,2)*1.8)
6340  IF Indi=1 THEN Dummy=Alf(Nn,I)
6350  IF Indi=2 THEN Dummy=Gama(Nn,I)
6360  SUBEND
6370  ! ************************************************************************
6380  !
6390  !       SUBROUTINE LEAK BEGINS HERE
6400  !
6410  ! ************************************************************************
6420  SUB Leak(W1i,W2i,T1i,T2i,W1op,W2op,T1op,T2op,W1o,W2o,T1o,T2o,Xeff)
6430  COM Wnolkp(*),Tnolkp(*),Wnolkr(*),Tnolkr(*)
6440  COM Cp,Ac,Len,Tau,Alpa,Ncovs,Epp,Epg,D,Width,Amu,Ak,Del,W,Cb,Id,Hfilmi,Kfi
n,Ub,Qduct(*),Ctype,Iflag,Amassp,Amassr,Wxe,Phi,Along,E,Sl,Dtdc,Dtcd,Od,Aflow
6450  COM Shr,Squ,Seff,Sqtl,Sqaux,Sqsurp,R(*),Qaux(*),Qtl(*),Qsurp(*),Al,Fl,Void
,Rps,Rhof,U,Dd,Fric,Delop,Delor
6460  COM Alfa(*),Sig(*),Gam(*),Bleak,Pleakp,Pleakr,Rleak,Ctau,Cal,Pa,Aff,Alf(*)
,Gama(*),Kount1,Kount2,Qdehu(*),Qhex(*)
6470  COM Perb,Perd,Reflec,Omega(*),Af,Av,Hmu,Enu,Wmass
6480  COM T(*),V(*),H(*),Tmin(*),Ihr(*),Tic(*),Toc(*),Eff(*),Qu(*)
6490  T1i=T1i+273.16
6500  T2i=T2i+273.16
6510  T1op=T1op+273.16
6520  T2op=T2op+273.16
6530  T1o=(Pleakp*T1i+Rleak*T2i+(1-Bleak-Pleakp)*T1op)/(1-Bleak+Rleak)
6540  W1o=(Pleakp*W1i+Rleak*W2i+(1-Bleak-Pleakp)*W1op)/(1-Bleak+Rleak)
6550  !
6560  ! !!!!! THESE EQUATIONS ARE USED IF NO HEAT EXCHANGER IS IN USE
6570  !
6580  T2o=(Pleakr*T2i+Bleak*T1i+(1-Pleakr-Rleak)*T2op)/(1+Bleak-Rleak)
6590  W2o=(Pleakr*W2i+Bleak*W1i+(1-Pleakr-Rleak)*W2op)/(1+Bleak-Rleak)
6600  T1o=T1o-273.16
6610  T2o=T2o-273.16
6620  T1i=T1i-273.16
6630  T2i=T2i-273.16
6640  T1op=T1op-273.16
6650  T2op=T2op-273.16
6660  SUBEND
6670  ! ************************************************************************
6680  !
6690  ! SUBROUTINE COLPER BEGINS HERE
6700  !
6710  ! ************************************************************************
6720  SUB Colper(J,I,T,V,H,Tfd,Tnd,Hn,Eff,Qu,Ul,Ut,Fp,Fr,Tic,Toc,Ihr,Idate,Tmin)
6730  COM Wnolkp(*),Tnolkp(*),Wnolkr(*),Tnolkr(*)
6740  COM Cp,Ac,Len,Tau,Alpa,Ncovs,Epp,Epg,D,Width,Amu,Ak,Del,W,Cb,Id,Hfilmi,Kfi
n,Ub,Qduct(*),Ctype,Iflag,Amassp,Amassr,Wxe,Phi,Along,E,Sl,Dtdc,Dtcd,Od,Aflow
6750  COM Shr,Squ,Seff,Sqtl,Sqaux,Sqsurp,R(*),Qaux(*),Qtl(*),Qsurp(*),Al,Fl,Void
,Rps,Rhof,U,Dd,Fric,Delpp,Delpr
6760  COM Alfa(*),Sig(*),Gam(*),Bleak,Pleakp,Pleakr,Rleak,Ctau,Cal,Pa,Aff,Alf(*)
,Gama(*),Kount1,Kount2,Qdehu(*),Qhex(*)
6770  COM Perb,Perd,Reflec,Omega(*),Af,Av,Hmu,Enu,Wmass
6780  COM T(*),V(*),H(*),Tmin(*),Ihr(*),Tic(*),Toc(*),Eff(*),Qu(*)
6790  K=I
6800  L=J
6810  Sigma=5.669E-8
6820  IF J>1 THEN 6920
6830  !
6840  !  INITIALIZE DAILY VALUES
6850  !
6860  Shr(I)=0
6870  Squ(I)=0
6880  Seff(I)=0
6890  Sqtl(I)=0
6900  Sqaux(I)=0
```

```
6910  Sqsurp(I)=0
6920  T(I,J)=T(I,J)+273.16
6930  Tmin=Tmin+273.16
6940  Tfd=Tfd+273.16
6950  Qaux(I,J)=0
6960  Qsurp(I,J)=0
6970  !
6980  ! DETERMINE INSOLATION ON TILTED COLLECTOR SURFACE
6990  !
7000  Bb=360*(Idate-31)*PI/(364*180)
7010  E=9.87*SIN(2*Bb)-7.53*COS(Bb)-1.5*SIN(Bb)
7020  Del=23.45*SIN(2*PI*((284+Idate)/365))
7030  Rs=Sl*PI/180
7040  Rphi=Phi*PI/180
7050  Rdel=Del*PI/180
7060  Romegs=ACS(-(TAN(Rdel)*TAN(Rphi)))
7070  Omegas=Romegs*180/PI
7080  Stime=Ihr-(-E-4*(90-Along))/60
7090  Omega(K,L)=15*(12-Stime)
7100  IF ABS(Omega(K,L))>=ABS(Omegas) THEN 7210
7110  Romega=Omega(K,L)*PI/180
7120  !
7130  ! DETERMINE RB
7140  !
7150  Thetat(K,L)=180/PI*ACS(COS(Rs)*COS(Rdel)*COS(Romega)+SIN(Rs)*SIN(Rdel))
7160  Thetaz(K,L)=180/PI*ACS(SIN(Rdel)*SIN(Rphi)+COS(Rdel)*COS(Rphi)*COS(Romega)
)
7170  Rb(K,L)=COS(Thetat(K,L)*PI/180)/COS(Thetaz(K,L)*PI/180)
7180  R(K,L)=Perb*Rb(K,L)+Perd*(1+COS(Rs))/2+(1-COS(Rs))*Reflec/2
7190  Hr(K,L)=R(K,L)*H(K,L)
7200  GOTO 7270
7210  Omega(K,L)=0
7220  Rb(K,L)=0
7230  R(K,L)=0
7240  Hr(K,L)=0
7250  Thetat(K,L)=0
7260  Thetaz(K,L)=0
7270  Ut=0
7280  Ul=0
7290  Fo=0
7300  Fr=0
7310  !
7320  ! DETERMINE COLLECTOR USEFUL GAIN Qu, CHANGE Hr UNITS TO W/M2
7330  !
7340  Hr(I,J)=Hr(I,J)/3.6
7350  H(I,J)=H(I,J)/3.6
7360  Tic=T(I,J)+3
7370  IF Hr(I,J)<=0 THEN 7400
7380  CALL Coleff(Hr(I,J),T(I,J),V(I,J),Eff(I,J),Qu(I,J),Tic,Ut,Ul,Fo,Fr,Thetat(
I,J))
7390  GOTO 7420
7400  Eff(I,J)=0
7410  Qu(I,J)=0
7420  Toc=Qu(I,J)/(Amassr*1005)+Tic
7430  Trd=Toc-Dtcd
7440  !
7450  ! DETRMINE DUCT LOSS AND AUXILIARY & TOTAL HEATING REQUIREMENTS
7460  !
7470  Qduct(I,J)=Amassr*Cp*(Dtcd+Dtdc)
7480  IF Trd>=Tmin THEN 7520
7490  Qaux(I,J)=Amassr*1005*(Tmin-Trd)
7500  Qtl(I,J)=Qaux(I,J)+Qu(I,J)
7510  GOTO 7530
7520  Qsurp(I,J)=Amassr*1005*(Trd-Tmin)
7530  !
7540  ! CHANGE TEMPERATURES BACK TO DegC
```

```
7550  !
7560  Tfd=Tfd-273.16
7570  Trd=Trd-273.16
7580  T(I,J)=T(I,J)-273.16
7590  Tic=Tic-273.16
7600  Toc=Toc-273.16
7610  Tmin=Tmin-273.16
7620  !
7630  ! CALCULATE DAILY VALUES
7640  !
7650  Shr(I)=Hr(I,J)+Shr(I)
7660  Squ(I)=Qu(I,J)+Squ(I)
7670  Seff(I)=Eff(I,J)+Seff(I)
7680  Sqaux(I)=Qaux(I,J)+Sqaux(I)
7690  Sqsurp(I)=Qsurp(I,J)+Sqsurp(I)
7700  SUBEND
7710  ! ***********************************************************************
7720  !
7730  ! SUBROUTINE COLEFF BEGINS HERE
7740  !
7750  ! ***********************************************************************
7760  SUB Coleff(Hr,Ta,Vwind,Eff,Qu,Tfin,Ut,Ul,Fp,Fr,Theta)
7770  COM Wnolkp(*),Tnolkp(*),Wnolkr(*),Tnolkr(*)
7780  COM Cp,Ac,Len,Tau,Alpa,Ncovs,Epp,Epg,D,Width,Amu,Ak,Del,W,Cb,Id,Hfilmi,Kfi
n,Ub,Gduct(*),Ctype,Iflag,Amassp,Amassr,Wxe,Phi,Along,E,Sl,Dtdc,Dtcd,Dd,Aflow
7790  COM Shr,Squ,Seff,Sqtl,Sqaux,Sqsurp,R(*),Qaux(*),Qtl(*),Qsurp(*),Al,Fl,Void
,Rps,Rhof,U,Dd,Fric,Delpp,Delpr
7800  COM Alfa(*),Sig(*),Gam(*),Bleak,Pleakp,Pleakr,Rleak,Ctau,Cal,Pa,Aff,Alf(*)
,Gama(*),Kount1,Kount2,Qdehu(*),Qhex(*)
7810  COM Perb,Perd,Reflec,Omega(*),Af,Av,Hmu,Enu,Wmass
7820  Sigma=5.669E-8
7830  Ul=0
7840  Qu=0
7850  I=0
7860  G=Amassr/Ac
7870  Svol=.2868*Tfin/101.3
7880  Vdot=Amassr*Svol
7890  !
7900  ! UNITS OF Vac IN FPM, FOLLOWING EQUATIONS ONLY FOR SOLARON SERIES 3000
7910  !
7920  Vac=Vdot/Ac*(60/.3048)
7930  Frta=.1470+.2485*Vac-.035*Vac^2
7940  Frul=.098+.52*Vac-.073*Vac^2
7950  Cosin=1/COS(Theta*PI/180)
7960  Gm=Cosin-1
7970  Amod=1.00636+.25793*Gm-.38517*Gm^2
7980  Eff=Amod*(Frta-Frul*5.6785*(Tfin-Ta)/Hr)
7990  Qu=Eff*Ac*Hr
8000  Ut=0
8010  Ul=0
8020  Fp=Frul
8030  Fr=Frta
8040  IF Qu>=0 THEN 8070
8050  Qu=0
8060  Eff=0
8070  SUBEND
```

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PERFORMANCE OF SOLAR REGENERATED ROTATING BEDS OF SILICA GEL

by

Gopal P. Ananth

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ABSTRACT

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MASTER OF SCIENCE

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ABSTRACT

An existing computer program that predicts the dehumidification-regeneration performance of a rotary bed of silica gel is modified to enable specification of differing air flow rates, temperatures, and humidity ratios on process and regeneration streams. To facilitate execution of the program, it is written in BASIC language that enables an interactive desk top computer available in the Department, be used.

The accuracy of the revised computer model has been checked against actual experimental data. After establishing the validity of the computer model, it is used to generate performance curves over a wide range of operating parameters of the silica gel dehumidifier. Finally, an air conditioning system that uses sorption dehumidification principle with solar regeneration is proposed and analyzed for an ideal situation, and comparison is made with an ideal vapor compression refrigeration air conditioning system.