

THE DEHUMIDIFICATION OF AIR USING
SOLAR REGENERATED ROTATING
BEDS OF SILICA GEL

by

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B.S., University of Missouri-Rolla, 1975

A THESIS

submitted in partial fulfillment of the
requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1981

Approved by:



Major Professor

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NOMENCLATURE

<u>Symbol</u>	<u>Definition</u>
<u>English Letter</u>	
BLEAK	Bypass leakage rate
C_p	Specific heat of air, J/kg °K
F'	Collector efficiency factor
FR	Collector heat removal factor
$FR\tau\alpha$	Product of collector heat removal factor, transmittance and absorptance
FRUL	Product of collector heat removal factor, and heat loss coefficient, W/m^2 °K
HR	Solar radiation per unit area in plane of the collector, W/m^2 °K
MOD	Collector efficiency modifier
PLEAK	Periphery leakage rate
RLEAK	Recirculation leakage rate
T_a	Ambient temperature, °K
T_{cin}	Temperature cold side inlet, °K
T_{cout}	Temperature cold side outlet, °K
T_{hin}	Temperature hot side inlet, °K
T_{hout}	Temperature hot side outlet, °K
T_{in}	Temperature collector inlet, °K
T_{1I}, T_{2I}	Inlet temperature of side 1 and side 2, respectively, °K

<u>Symbol</u>	<u>Definition</u>
T_{10}, T_{20}	Outlet temperature of sides 1 and 2, respectively, including effects of leakage, °K
T_{10}', T_{20}'	Outlet temperature of sides 1 and 2, respectively, excluding effects of leakage, °K
W_{1I}, W_{2I}	Inlet humidity ratio sides 1 and 2, respectively, kg water/kg air
W_{10}, W_{20}	Outlet humidity ratio sides 1 and 2, respectively, including the effects of leakage, kg water/kg air
W_{10}', W_{20}'	Outlet humidity ratio sides 1 and 2, respectively, excluding the effects of leakage, kg water/kg air

Greek Letters

ϵ	Heat exchanger effectiveness
η	Collector efficiency
λ	Uncertainty

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CHAPTER I

INTRODUCTION

The use of dehumidified air has, for decades, been employed to numerous industrial applications. In the production of foods such as potato chips, dry breakfast foods, crackers, dried fruits, and powdery foods, air humidity control is a necessity for quality control. In the pharmaceutical and chemical industries dehumidified air has been used to reduce lumping and caking of powdered substances due to the presence of moisture. Air humidity control in storage areas is a must to reduce the possibilities of rot, mildew, rust and corrosion, especially when storage locations are underground or in marine areas. Humidity control of operating equipment rooms, and packaging rooms, in many instances reduces the maintenance requirements of electrical and mechanical systems. Also many pneumatic control systems require the use of very low humidity air to prevent the possibility of water condensation in instrument air lines.

More recently the applications of dehumidified air to the lumber, grain drying and human comfort areas have been investigated. In the grain and lumber industry product drying is normally achieved by blowing ambient air heated from 65°C (150°F) to 85°C (185°F) through the grain or lumber held in a drying chamber. Since the partial pressure of the water inside the grain or wood is higher than the partial pressure of the water in the heated air, water is transferred from the product to the airstream. The advantages of this conventional system are the simplicity of operation, low drying times and proven technology.

Drying with the use of dehumidified air is accomplished by reducing the wet bulb temperature of the air until the partial pressure of water in the grain or lumber is greater than the partial pressure of the water in the dehumidified air.

Check and Pfaff (6) have indicated the dehumidification drying of spruce studs would result in an energy savings of 46 to 68 percent compared to conventional drying techniques. Dehumidification drying also has the added advantages of being able to dry softwoods and hardwoods simultaneously in the same kiln. Kiln construction may be of any air tight, moisture resistant, insulated material, not restricted to high temperature applications only, as in the conventional systems. Also the spruce studs which are dried by dehumidified air exhibited less shrinkage, warp and grade drop. The major disadvantage of dehumidification drying is the increased drying time.

The use of dehumidified air in the drying of grain has been investigated by Rodda (37) and Danziger et al. (11). Dehumidification drying was shown to be a very effective way of grain drying when compared to conventional methods. Less kernel cracking and other grain degradation was found in the dehumidification dried grain. Also, dehumidification dried corn at 21°C (70°F) maintained germination above 90% while corn dried with conventional heated air at 60°C (140°F) exhibited zero percent germination.

Until recently, humidity control of conditioned air for human comfort has been accomplished by employing a conventional air conditioner. In this conventional system the cooling coil is required to remove the moisture from the circulated room air as well as reduce the dry bulb

temperature. In some instances, to achieve proper moisture control the air is over-cooled to remove the excess moisture content and then must be reheated to achieve the required dry bulb temperature resulting in obvious energy waste. Felski et al. (16) has indicated that an alternative method would be to dehumidify outside makeup air and introduce it into the return ducts of the conventional air conditioner. Ideally, the returning air mixture would then have the proper moisture content and the cooling coils would only be required to remove the sensible cooling load. Lunde (28) has indicated this could reduce the air conditioner's load requirement by 20 to 30 percent. Also since the cooling coil temperature of the air conditioner could possibly be increased to 20°C (68°F), the air conditioner could be operated more efficiently.

Coincident with these increased requirements for dehumidified air, there has been a decrease in the availability of fossil energy sources. This has led to an increased interest in the use of solar energy to produce dehumidified air. In most research utilizing solar energy the dehumidification apparatus is of similar construction to a conventional system with the exception that the heat required to regenerate the desiccant material is supplied from solar collectors.

The objectives of this research were to of been three fold;

1. Because the selection of the optimum solar dehumidification technique has been a topic of much investigation, an evaluation of the numerous dehumidification types and arrangements was to be conducted with regard to their application to solar energy.

2. Once this preferred method had been selected , a computer model simulating this solar dehumidifier was to be developed.
3. A test apparatus of the preferred technique was to be constructed so a comparative evaluation of the computer model could be performed.

CHAPTER II

DEHUMIDIFICATION PROCESS DESCRIPTION

Methods of Dehumidification

The dehumidification of air can be accomplished by three economically feasible methods; refrigeration, liquid sorption, solid sorption, or a combination of these systems.

The refrigeration method is the most widely utilized, efficient, compact, reliable and technically understood method of dehumidification. This system, composed of a common mechanical air conditioner, employs a refrigerant, compressor, expansion valve and evaporation coils. As humid air comes into contact with the evaporating coils the air's dew point and dry bulb temperature are reduced near the evaporating or cooling coil temperature, thereby condensing excess moisture from the air.

In liquid sorption dehumidification systems, air is passed through an atomized spray of hygroscopic liquid sorbent such as solutions of lithium chloride, calcium chloride, lithium bromide, ethylene glycol, ammonia, or sulfuric acid. Sorption using these sorbents is termed absorption since the sorbent changes physically or chemically during the sorption process. The sorbent in the concentrated state has a vapor pressure of water less than that of the air, therefore absorbing moisture from the air. When the sorbent solution becomes diluted with the absorbed moisture it is regenerated by heating and spraying the solution in an ambient air stream. To provide a continuous process only a portion of the total sorbent solution is bled-off and regenerated while the rest remains in the contactor section.

If this process is conducted without the loss of heat, it will be adiabatic causing the outlet dry bulb temperature to increase significantly. If, however, the sorbent solution is sprayed over cooling coils in the contact chamber the process becomes nearly isothermal.

Solid sorption can be accomplished by absorption or adsorption utilizing either fixed or rotating beds. The dehumidification is accomplished by passing the unconditioned air through the hygroscopic sorbent or desiccant. Bullock and Therelkeld (5) describe the solid sorption process as the migration of the water vapor molecules to the desiccant surface caused by the unbalanced molecular forces occurring at the surface of the solid desiccant. This spontaneous process continues until a concentrated layer of the vapor is collected on the desiccant surface. The process is exothermic. Regeneration of the desiccant is accomplished by passing a heated air stream through the bed causing the absorbed layer to evaporate from the surface of the desiccant.

This process is termed adsorption or absorption dependent upon the physical or chemical changes of the desiccant. In the absorption of water on a desiccant, such as lithium chloride, the desiccant undergoes a chemical change to a hydrate state. Additional moisture absorption would cause the desiccant to dissolve into a solution.

The desiccant used in adsorption processes will not react physically or chemically with the condensed vapor which is termed sorbate. Numerous substances may be used for adsorption desiccants including: activated alumina, silica gel, activated bauxites and activated charcoals.

If fixed beds are utilized in the solid sorption process, two beds must be installed to maintain the system in continuous operation (i.e. one

is regenerated while the other is in use). A rotating bed requires the use of only one bed for a continuous operating system. A portion of the bed is rotated into the regeneration chamber while the rest remains in the operation mode.

Solar Dehumidification

The particular attractiveness of solar dehumidification lies in the fact that dehumidification need may, in many cases, parallel solar radiation intensity. This may be especially true in the area of human comfort.

Several papers have summarized the recent research being conducted to utilize solar energy to regenerate the desiccant material so it can be reintroduced into the moist air stream. Satcunanathan and So'Brien (43) have published an excellent summary of solar dehumidification techniques. Similar papers have been written by Shelpuk (44), Thornton (49), Shelpuk and Hooker (45), and Olsen (35).

Solar Dehumidification Utilizing Vapor Compression Cycles

Although dehumidification by compression refrigeration is the most efficient, compact, and reliable, the adaptation of this system to solar energy does not appear attractive in the near future. Either the compressor is electrically driven from photovoltaic solar cells or directly driven using a solar heat engine similar to those discussed by Eckard (14), Freeman (17) and Barber (2). Neither of these two methods is as attractive as other comparable methods of solar dehumidification currently available.

Solar Dehumidification Utilizing Liquid Sorbents

The investigation of solar liquid desiccant systems was pioneered by Löff (25) in 1955 as the critical part of a solar air conditioning system. Others who have studied or reported in this area included Mallick and Gupta (33), Olsen (35), Johannsen (20), Kababaer and Khandurdyev (21), Boldrin et al. (4) and Valentine (50). These systems utilize either lithium bromide, calcium chloride, glycol or ammonia as the liquid desiccant. The liquid system's major advantages over the solid systems are its lower air pressure drop, and the ease in which it can become an isothermal process.

The disadvantages of the liquid systems are the solution leakage possibilities, solution entrainment in the processed air, cooling tower requirements, maintenance requirements of pumps, valves, and heat exchangers as well as their bulky size. These disadvantages become more pronounced when small domestic dehumidification systems are considered.

Solar Dehumidification Utilizing Solid Sorbents

The solar dehumidification of air using solid desiccants was originally developed in the early 1960's by Dunkle (13) in Australia, again with regard to solar air conditioning applications. The major advantages of these systems are simplicity and the fact that they may be constructed from inexpensive materials. Piping and pumps are not necessary with the solid desiccant. The only moving parts are the fans, and possible rotating desiccant beds and heat exchangers.

While the major advantage of a solid desiccant system is its simplicity, it has several disadvantages when compared to a liquid dehumidification

system. If the solid desiccant system has a rotating bed, there will be inherent air leakage past the seals separating the two airstreams. If it is a fixed bed system, the apparatus is made more complex with the valves and associated control mechanism required to provide continuous regeneration. The solid desiccant system will have a greater air stream pressure drop through the dehumidifier requiring increased fan power. Also since it is difficult to provide cooling during the solid desiccant dehumidification process, it will be a more nearly adiabatic process than the isothermal liquid system. This results in a higher final dry bulb air temperature.

After analyzing the advantages and disadvantages of the liquid and solid desiccant systems, it was decided that the solid system may provide superior performance for solar dehumidification applications.

A schematic of a continuous solar air dehumidification system is shown on Figure 1. To make the system shown continuous, the desiccant wheel is rotated allowing concurrent air dehumidification through one half of the bed and desiccant regeneration in the remaining portion of the bed. The flow of air is directed through the system by way of two isolated flow paths: the process flow stream, in which air will be dehumidified, and the regeneration flow stream which is used to regenerate the desiccant.

The rotating bed system shown is considered superior to the fixed bed system because of its increased simplicity. No dew point sensing devices or timers are required to initiate the regeneration cycle characteristic of the fixed bed system. Also no mechanically modulated directional dampers and significantly less ductwork are required by the rotating system. The major disadvantage of the rotating bed technique is

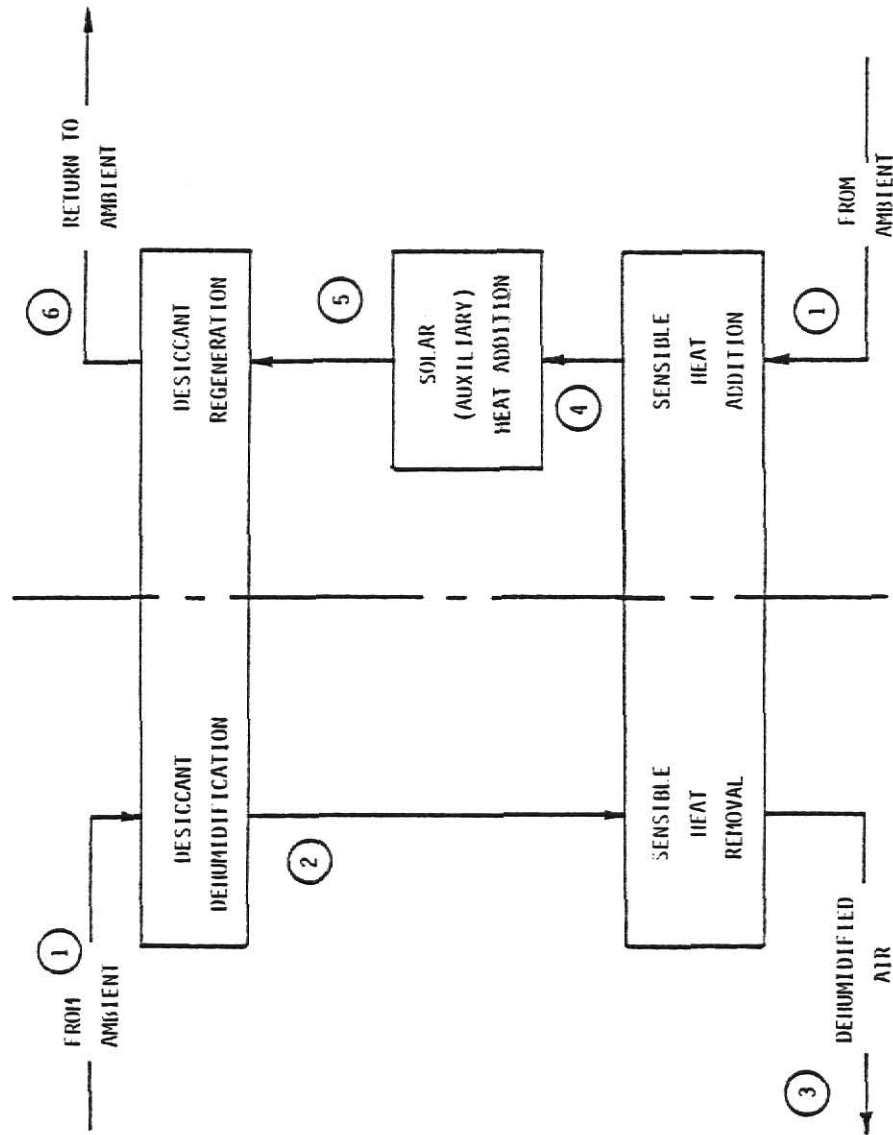


FIGURE 1 SCHEMATIC OF SOLAR DESICCANT DEHUMIDIFICATION APPARATUS

the air stream leakage associated with the wheel seals.

Figure 2 shows a psychrometric plot of the flow streams at each state during the dehumidification process. Air to be dehumidified (ambient air) enters the process flow stream at condition 1 and is adiabatically dehumidified to state 2. The dried air has a higher dry bulb temperature resulting from the heat of condensation being converted to sensible heat in the process airstream. This flow stream then passes through an optional sensible heat exchanger to reduce the final dry bulb temperature to point 3.

The regeneration flow stream begins with ambient air at state 1. The air is heated to state 4 by passing it through the sensible heat exchanger adding the heat extracted from the process flow stream. Because this temperature is insufficient for desiccant bed regeneration, the flow stream is then directed through the solar collectors. If the temperature is still too low for desiccant regeneration, it is increased to the predetermined regeneration air temperature by the use of auxiliary heaters to point 5. As the regeneration air stream passes through the desiccant bed, the bed is regenerated and the air stream is humidified to state 6 where it is released to the atmosphere.

It should be noted that in some drying application the dehumidified air at point 2 may be at more beneficial conditions than point 3. In this instance, the heat exchanger is removed from the system resulting in the regeneration flow stream being heated from point 1 to point 5 by solar and auxiliary heating.

Comparison of Solid Desiccant Bed Materials

From Dunkles original work on solar dehumidification systems, subsequent investigators have developed solid desiccant systems differing most

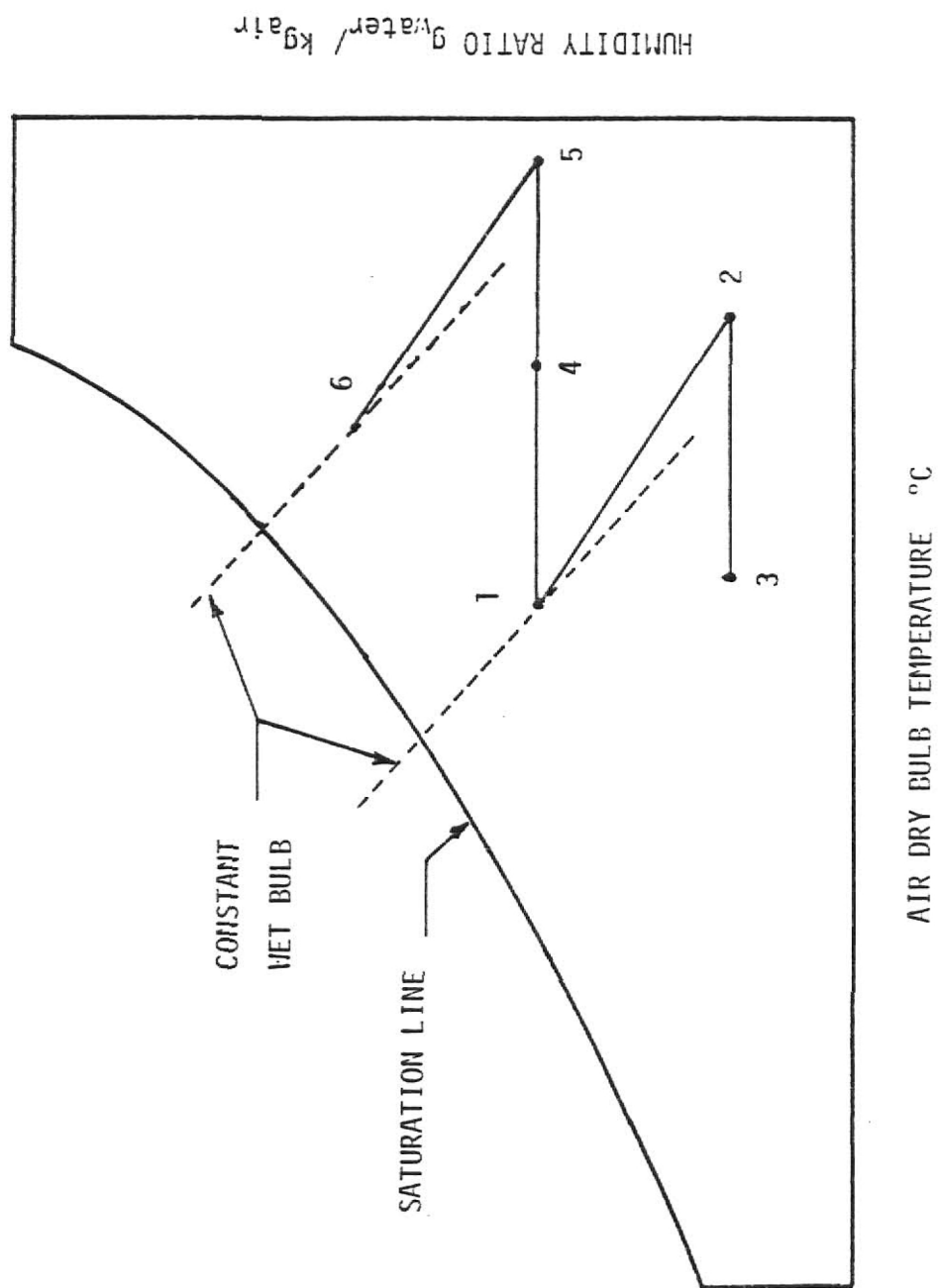


FIGURE 2 PSYCHROMETRIC PLOT OF DEHUMIDIFICATION PROCESS

significantly in the desiccant material utilized. Lunde (26) (27) (28) employed a rotating desiccant bed containing silica gel with interspaced heat exchangers to produce a more nearly isothermal process. Rush (40) (41) (42) developed a system using a molecular sieve as the desiccant bed. This molecular sieve was originally constructed from sodium aluminum silicate deposited on an asbestos matrix which was later replaced by a refractory material to improve long term stability. Mei et al. (30) have developed a unique fixed desiccant bed constructed by depositing silica gel upon a Teflon matrix web. Robinson et al. (39) have investigated the use of a commercially available lithium chloride rotating desiccant wheel as the heart of a solar air conditioning apparatus. Tchernev (48) has investigated the use of zeolite for air dehumidification.

Because the selection of the optimum solar desiccant material is a topic of much discussion, Satcunanathan and So'Brien (43) have developed a rating system based upon the desiccant efficiency, capacity, and speed to dehumidify. However, the overriding parameter for solar desiccant selection is the ease with which it can be regenerated. This implies that the desiccant material should have a regeneration temperature obtainable with common flat plate collectors, if minimum auxiliary fossil fuels are to be required. When considering this criterion silica gel and activated alumina desiccant beds become the most attractive. However silica gel proves to be superior to activated alumina in the capacity to adsorb water from the air stream. This results in silica gel being the common choice for solar dehumidification. The principle

disadvantage of silica gel is the air pressure drop associated with a packed bed. This is one of the primary reasons other desiccant beds have been developed such as the molecular sieve (Rush), lithium chloride on a honeycomb matrix (Robinson et al), and silica gel on Teflon (Mei). Each of these, except the silica gel on Teflon would require an additional "topping" air heater to completely regenerate the bed. Other important parameters to consider in desiccant bed selection include: low cost, long life, and stability for all gases found in air such as NO_x , CO_2 and SO_2 .

Other Associated Equipment

The other equipment required for solar dehumidification utilizing a solid adsorbent include the heat exchanger wheel, drive motors to rotate the desiccant and heat exchanger wheels, fans, auxiliary heaters, and the airtight enclosure to house these components.

The most significant of these components is the sensible heat exchanger. Because the adsorption process is nearly adiabatic, it produces a high final air dry bulb temperature; this heat is then transferred to the inlet regeneration air in the sensible heat exchanger. This air pre-heating permits a higher collector temperature which in turn allows more complete regeneration of the desiccant bed. Properties of the sensible heat exchanger are that it must have a high heat exchanger effectiveness, a low air pressure drop, a low thermal conductivity in the direction of air flow, the appropriate heat capacity and a long life. The heat exchanger must also be hydrophobic since any absorbed water will result in loss of performance.

CHAPTER III

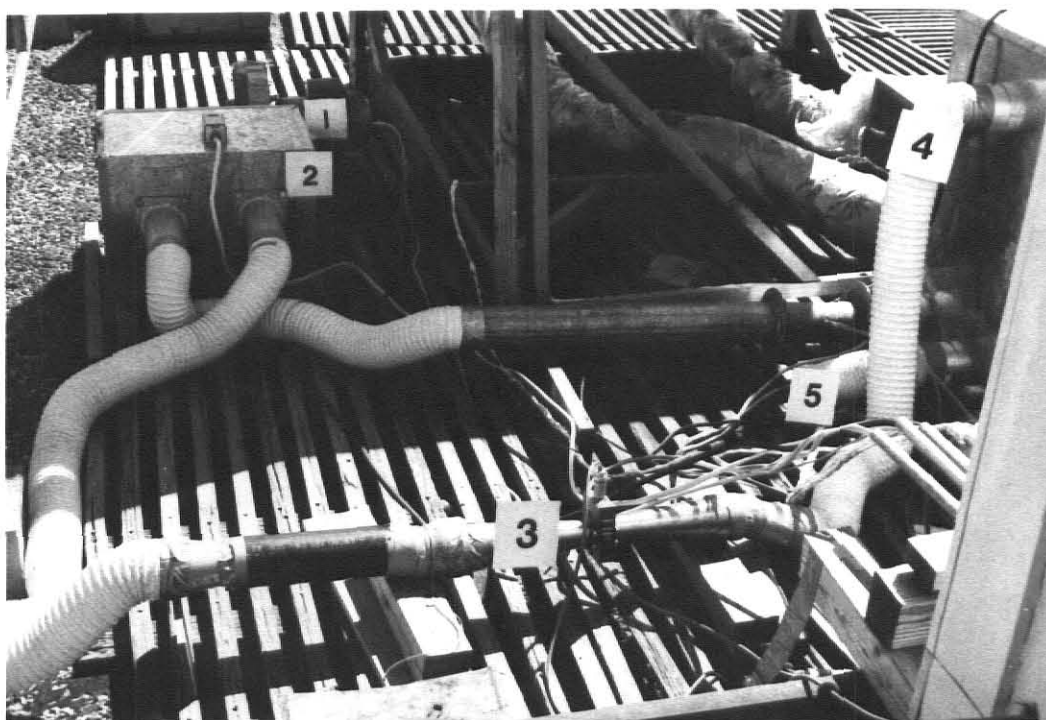
SOLAR DEHUMIDIFICATION APPARATUS

The experimental apparatus used in this investigation is shown in Figure 3. The process side of the apparatus is shown in photograph (a). Process air enters the system through a fan (1) and flows into a flow divider (2) where the flowstream is modulated for the proper flowrate. The flowrate is measured by an electronic flowmeter (3). The flow stream then enters the dehumidifier (4) and exits to the atmosphere (5).

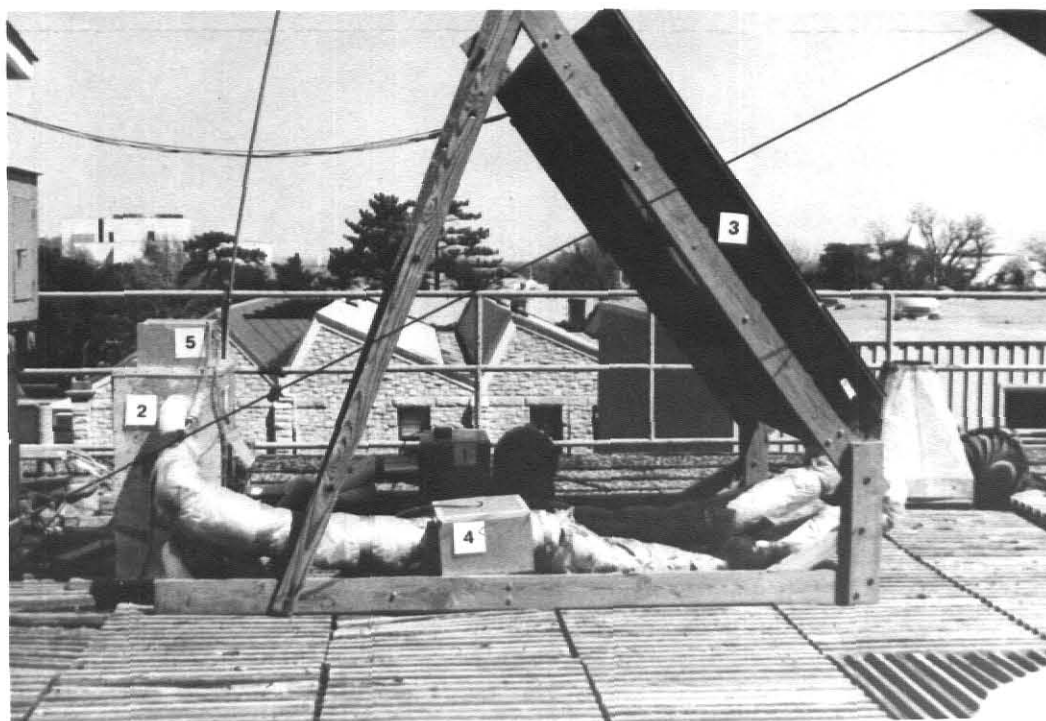
The regeneration side is shown in photograph (b). The regeneration flow stream enters through the fan (1) and is modulated and measured similar to the process flow stream before it enters the dehumidifier (2). Heat is supplied to the system through the solar collectors (3). However, if this air temperature is insufficient for regeneration of the silica gel, the auxiliary heater (4) will provide sufficient heat input. This is accomplished by utilizing a thermostat (5) to maintain minimum regeneration temperature.

The complete dehumidification system can be divided into nine major components. They are: desiccant wheel, sensible heat exchanger, air seals, motor drives, dehumidifier housing and ducting, solar collectors, blowers, auxiliary heaters, and instrumentation.

Desiccant Wheel: The desiccant wheel is the principal component of the dehumidification system. The selected material for the desiccant wheel is commercial silica gel. Silica gel is an amorphous form of silica consisting mainly of partially hydrated silicon dioxide.



(a)



(b)

FIGURE 3 LAYOUT OF PROCESS (a) AND REGENERATION (b) FLOW STREAMS

The reasons for selecting silica gel as the desiccant material are numerous. First the adsorption of water vapor by silica gel has been studied for many years (31) (32) and is common practice. Therefore silica gel is readily available commercially, and is accepted by the dehumidification industry. Secondly there is published material available regarding the physical characteristics, heat of wetting, and equilibrium data of silica gel and water vapor (10) (18) (19). Finally, there have been numerous investigations into the numerical modeling of the silica gel dehumidification process utilizing fixed and rotating beds.

The silica gel used in the desiccant bed was manufactured by Davison Chemical Division of WR Grace and Co. Inc., Grade 01 (3-8 mesh). The estimated particle diameter is 3.23 mm (.0106 ft), the surface to volume ratio is $714.6 \text{ m}^2/\text{m}^3$ ($218 \text{ ft}^2/\text{ft}^3$) and the volume fraction is 75 percent (18).

The physical dimensions of the desiccant bed are shown in Figure 4 and its location in the dehumidifier in Figure 5. The bed is constructed from a plastic rim 10 mm (.39 in) in thickness. Attached to the bottom of the rim along its perimeter is a brass/felt seal to decrease leakage past the desiccant wheel into the next chamber. The silica gel is sandwiched between two layers of perforated aluminum sheet. Also located between these aluminum sheets is a plastic lattice which connects the outer rim with a small pulley located in the center and provides overall bed rigidity. The small pulley allows mounting of the desiccant wheel on a drive shaft. All the materials except the silica gel are assumed to be hydrophobic.

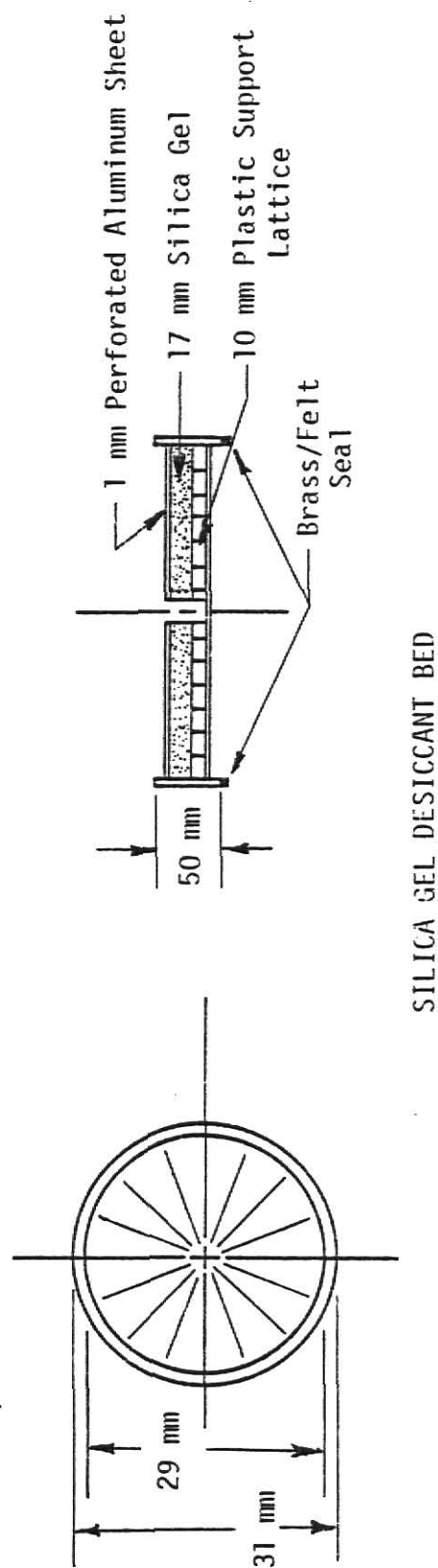
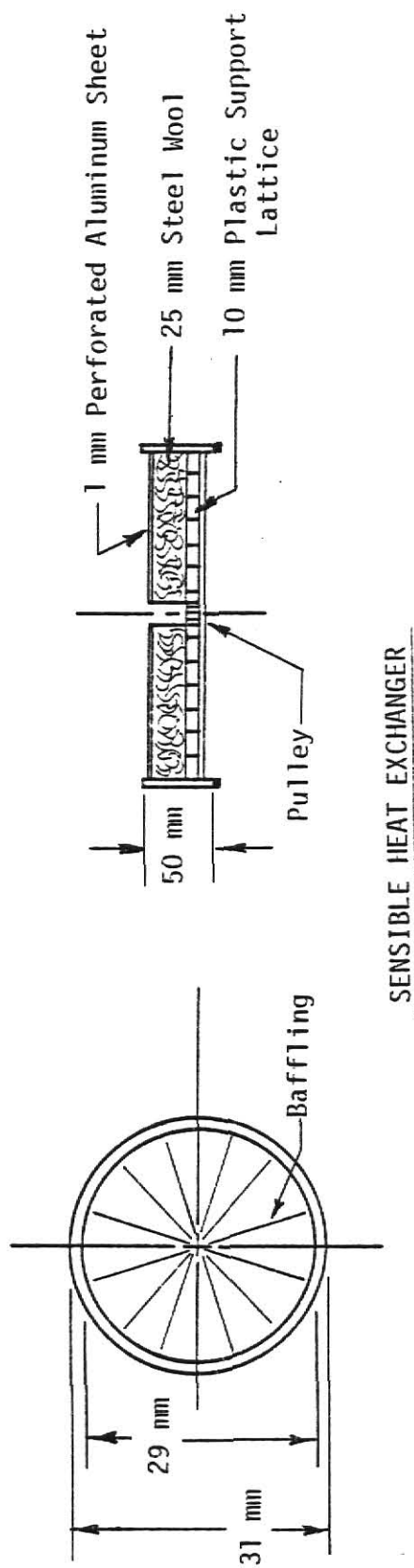


FIGURE 4 SENSIBLE HEAT EXCHANGER AND DESICCANT WHEELS

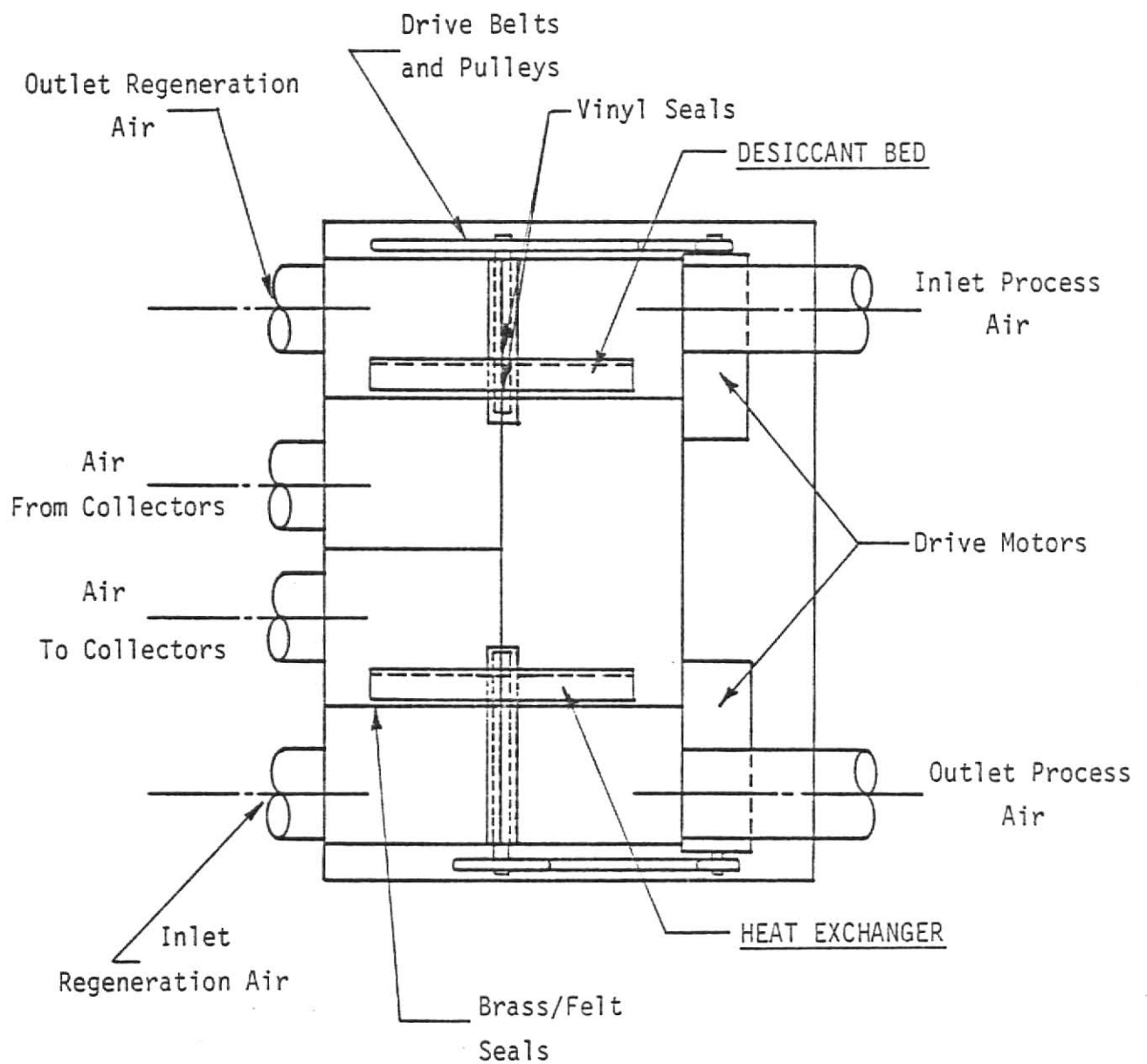


FIGURE 5 DIAGRAM OF DEHUMIDIFICATION APPARATUS

The total mass of the desiccant bed is 1.831 kg, of which 0.816 kg is contributed by the silica gel and adsorbed water vapor. Based upon formation by Hubbard (19) the estimated water vapor content of silica gel under the weighing conditions is approximately 21% by weight of silica gel. Therefore the actual estimated mass of the silica gel, only, is 0.645 kg.

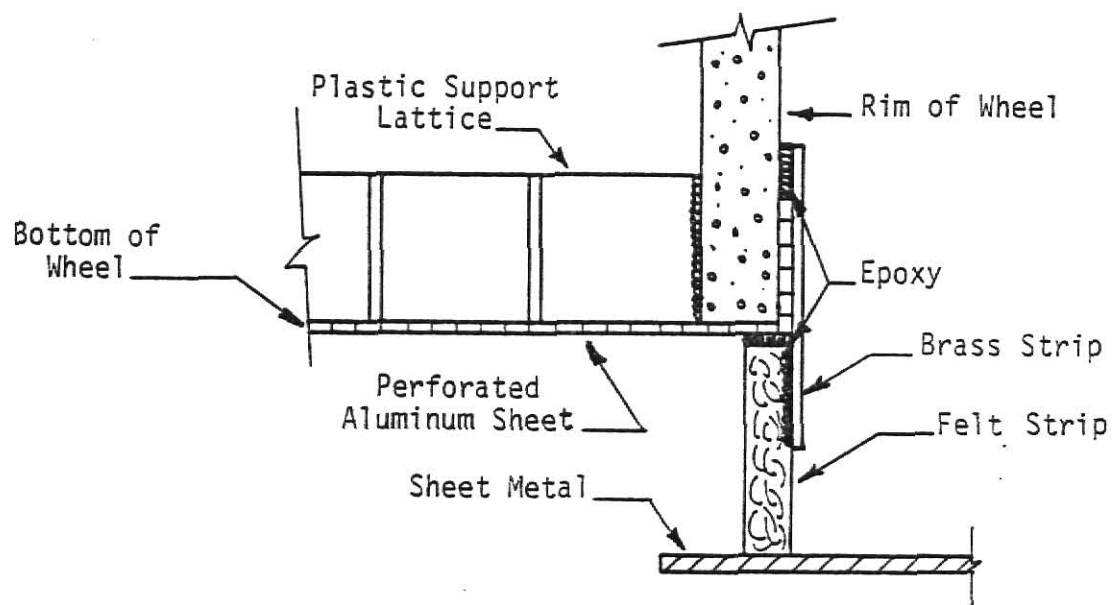
Sensible Heat Exchanger Wheel: The sensible heat exchanger wheel is constructed identically to the desiccant wheel and is located as shown in Figure 5.

The fill material is NAPA Balkamp No. 3 steel wool. The calculated fill density is 191 kg/m^3 (12 lbm/ft^3) and void fraction 98 percent.

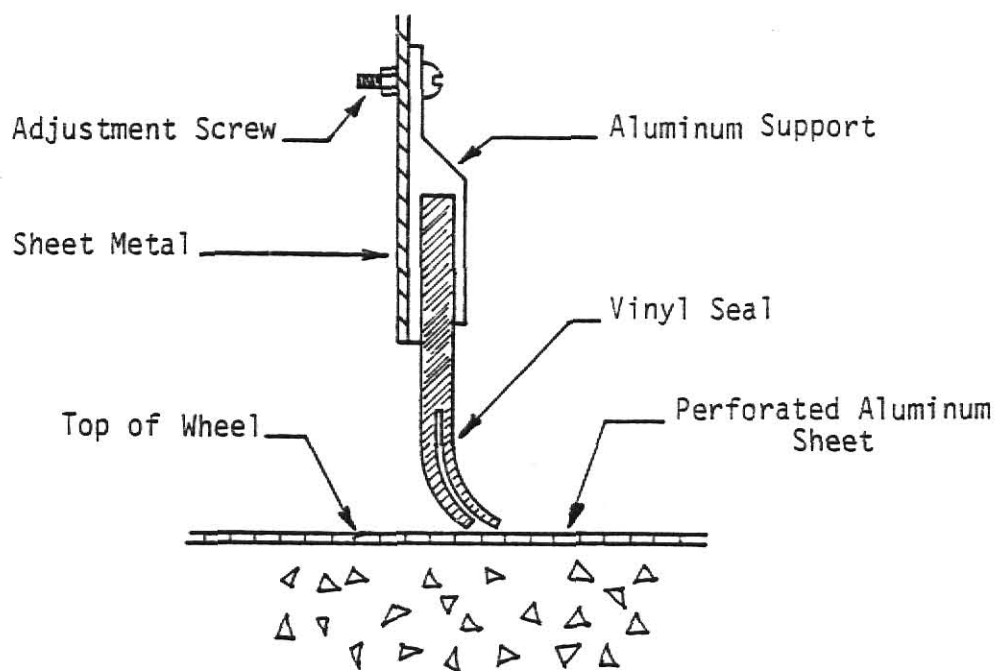
The physical dimensions of the sensible heat exchanger are shown on Figure 4. The total mass of the wheel is 1.279 kg. Of this, 0.481 kg, representing the fill and perforated aluminum, is considered to have any heat transfer ability. All materials composing the heat exchanger wheel are considered hydrophobic.

Air Seals Air seals required to reduce the leakage of air around the desiccant and heat exchanger wheels are shown in Figure 6. Seals were required in two areas: first to prohibit the flow of air around the desiccant and heat exchanger wheels while remaining in the same flow path and secondly to reduce air leakage from one flow path to the other where the rotating wheels pass through the compartment dividers.

To reduce the air flow around the rotating wheels in the same flow path a periphery seal was utilized. The periphery seal is constructed from a 25.4 mm (1 in.) 25 guage brass strip and a 15.9 mm (.625 in) by



CROSSECTION OF PERIPHERY SEAL



CROSSECTION OF CROSSFLOW SEAL

FIGURE 6 PERIPHERY AND CROSSFLOW SEALS

1.6 mm (.063 in) felt strip epoxied to the periphery of the bottom of the rotating wheels.

Leakage of air from one flow path to the other is reduced using a crossflow seal. These seals are constructed from a 25.4 mm (1 in.) vinyl strip held in a 25.4 mm (1 in) aluminum support. The seals are held to the sheet metal compartment partitions by screws which may be adjusted for various bed depths.

Motor Drives: Two motor drives and control systems are utilized to rotate the desiccant and heat exchanger wheels independently. The motors are Dayton Model 7Z574 1/8 hp, 90V DC, permanent magnet, TENV gear motors with Dayton SCR Model 6X165A 115 VAC input, 90 VDC output speed controllers. Full load speed of the motors is 2.3 rad/s (22 rpm) at a rated full load torque of 31.6 Nm (280 in lbf).

To provide proper rotating speeds of each wheel various belts and pulleys are required. The pulley ratio of the desiccant wheel is 22:1 while the heat exchanger pulley ratio is approximately 1.25:1. These pulley ratios will permit operating speeds of 0.0034 rad/s (2 rph) to 0.105 rad/s (60 rph) for the desiccant bed and 0.105 rad/s (1 rpm) to 1.885 rad/s (18 rpm) for the heat exchanger wheel.

Enclosures and Ducting: The desiccant wheel, heat exchanger wheel, drive motors, and associated pulleys and belts are enclosed in a sealed metal enclosure. The enclosure shown in Figure 5 is constructed from commercial 24 gauge sheet metal fitted over a 19.1 mm (.75 in) aluminum "L" framing. All internal compartments are constructed using sheet metal and insulated with 25.4 mm (1 in.) polystyrene insulation.

The ductwork located inside the enclosure is 102 mm (4.0 in) round aluminum duct with .5 mm (.0187 in) wall thickness. All external ducting is accomplished using 102 mm (4.0 in) flexible plastic duct insulated with a 102 mm (4 in) mineral fiber batt insulation manufactured by Certainteed Corp. The batt insulation is wrapped around the flexible ductwork and then covered with a plastic sheet to protect it from the environment.

Solar Collectors: The solar collectors employed to provide solar heated air to the dehumidification apparatus are two Solaron Series 3000 Air Collectors manufactured by Solaron Corporation in Denver Co. The collector aperture area is 1.586 m^2 (17.07 ft^2) for each collector. The single glazing consists of 3.2 mm (0.125 in) thick tempered, low iron glass with a weather resistant, synthetic rubber (EPDM) edge gasket. Transmittance of the glazing is 0.89. The absorber plate consists of electroplated black chrome on 24 gauge steel with a 0.5 mil undercoating of nickel. 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 l/s (37.4 scfm) to 26.5 l/s (56.2 scfm).

Blowers. One Dayton type 2C573 blower is used to provide air flow to the process and regeneration flow paths. The rated air flow at 3 in wg static pressure is 322 l/s (682 scfm) at a rotating speed of 361.2 rad/s (3450 rpm). The wheel diameter and width are 270 mm (10.625 in) and 64 mm (2.5 in), respectively.

The blower discharges into a supply plenum constructed of 24 gauge sheet metal. From this plenum the flow rates for the process air and the regeneration air may be controlled individually by modulating the guillotine

type dampers located in the plenum. An air filter is located inside the plenum to minimize dust entrainment into the system.

Auxiliary Heater: In the event the solar collectors are unable to maintain the regeneration air temperature to the desiccant bed at the required set point, an auxiliary heater will automatically be energized to maintain the regeneration set temperature. The auxiliary heater consists of a Eagle Model 415A, 660W cone glo coil contained in a 457 mm (18 in) by 457 mm (18 in) by 610 mm (24 in) sheet metal enclosure. The auxiliary heater enclosure is equipped with a series of baffles to promote mixing of the air flow stream.

The auxiliary heater is controlled by a RFL Model 760K-1 92945 Temperature Controller. The regeneration air temperature is measured at the inlet to the desiccant wheel by an encased chrome-alumel thermocouple. Should this temperature be below the set point of the temperature controller the auxiliary heater is energized. For safety considerations, a Honeywell S688A sail switch is installed in the fan discharge plenum to insure sufficient air flow prior to auxiliary heater excitation. Should air flow be insufficient, the sail switch deenergizes a Dayton Model 5X846A power relay cutting off power to the auxiliary heater.

Instrumentation: Instrumentation was required to collect and record data of the following types: temperature, dew point, solar radiation, wind speed, time of day, and air flow rate.

A Monitor Labs Model 9302 data acquisition system is the focal point for most data recording. This data recording system provides an alpha-numeric display and printer coupled with a 365 day clock allowing

the automatic acquisition of hard copy data and concurrent time of day. This data recording system allows direct input and units conversion of thermocouples and permits units conversion from any instrument with output voltage from ± 30.0 mV to ± 10.0 V. This data recording system has capacity of 40 independent channels which may be scanned at any predetermined time interval of less than 999 hours. Averaging of up to 999 samples in each channel is also permitted. All programming is accomplished through the keyboard located on the front of the console.

Redundant temperature measurement is provided in each compartment of the dehumidifier, at the inlet and outlet to the solar collectors and auxiliary heater, and at the ambient conditions. This is achieved by the use of type T-copper constantan thermocouples inserted into the air flow streams at these locations. The thermocouples were constructed from 24 gauge copper constantan wire manufactured by Omega Corp. The ends of the thermocouple wires were stripped and soldered and then tested for uniformity by immersing them into a constant temperature bath. The thermocouples are terminated and grounded at the data acquisition system input connections. Aluminum foil is wrapped around the thermocouples as shielding to reduce radio frequency interference.

Dew point temperatures are measured in each flow stream at the inlet and outlet desiccant wheel compartments. These four dew points are measured by the insertion of a 6.4 mm (0.25 in) OD soft drawn copper tube into each of the four compartments. The copper tubing is routed to a five way electrically actuated valve permitting the systematic sampling from any of the four locations. A Hills-McCanna Ramcon Mode 832-3 rotary electro-

mechanical actuator is used to rotate the five way valve in $\pi/2$ radian (90 degree) increments when a signal is received from the timer. A Dayton Model 2E213 24 hour control timer is employed to provide an input signal to the valve actuator in 15 minute intervals.

Once the sample location is selected, a single copper tube delivers the air sample from the five way valve to the dew point analyzer. The dew point analyzer, a EG&G Model 911 Dew-All digital humidity analyzer, determines the sample dew point and transmits the reading to the data recorder. Intermittent wet and dry bulb temperatures are manually taken using a DigiTec HT Series 5810 thermometer to provide cross checking of the dew point analyzer.

To insure proper operation of this system a 160 watt heating tape and Delta-T Model ATT-70 differential temperature thermostat is utilized to insure that at no time does the tube wall temperature fall below the air sample dew point. All tubing located outdoors is also insulated with a fiberglass wrap and covered with plastic sheeting.

Solar radiation is measured by two independent methods. The first employs an Epply Radiometer Model PSP-16275F3 which provides direct millivolt input into the data acquisition system. A Hollis Observatory Model MR-5A-2092 pyranometer is the measuring device for the secondary system. Output from this pyranometer is directed to a Hollis Observatory Model LM-3000 Solar Integrator for output on pressure sensitive chart paper. The recorder provides instantaneous as well as integrated results.

Wind direction and speed is measured by a Climatronics Mark III Wind Measuring System with a wind velocity range of 0 to 22 m/s (50 mph).

Wind speed is determined using a 3 cup anemometer and wind direction with the use of a wind vane. Direction and speed are transmitted electrically to a Gulton Model 3 CS recorder where they are output on a pressure sensitive chart paper.

Air flow is measured in the process and regeneration flow path after the air exits the fan discharge plenum. The process flow stream air flow rate is measured by a Gould Datametrics Series 100-3F Mass Flow Transducer. The electrical transducer output is wired to a Datometrics Series 810C digital indicator giving instantaneous air flow rates. The regeneration volumetric flow rate is measured using a 2 inch ASME long radius flow nozzle and an oil filled manometer. The two flow-meters were calibrated prior to use.

A visual layout of the instrumentation is shown in Figure 7. The photograph shows the incident solar radiation recorder (1), digital electronic flowmeter indicator (2), dew point meter (3), wind speed and direction recorder (4), and the data logger (5).



FIGURE 7 LAYOUT OF INSTRUMENTATION USED FOR DATA AQUISITION

CHAPTER IV

COMPUTER MODEL

A principal objective of this research was the development of a computer model for a solar dehumidification process. The model constructed consists of a main program, MAIN, and seven subroutines; HUMID, EGV, ALFAV, LEAK, COLPER, COLEFF, and HEATX. A listing of the program and associated nomenclature are contained in Appendix B. All computer simulation and results were performed on the KSU IBM 370 digital computer, utilizing WATFIV FORTRAN language.

MAIN: This portion of the program performs three major tasks. First, most data input and output is accomplished through the MAIN subprogram. Required input parameters are hourly ambient temperature, humidity ratio, wind speed, and solar radiation. Parameters describing the dehumidification wheel, collectors, and heat exchangers are also required. Time of year and time of day must also be input for each hour so that the position of the sun may be determined relative to the earth. Collector orientation relative to the earth (i.e. slope, latitude, and longitude) must be input so actual radiation striking the collector may be determined. The program output consists of temperature and humidity ratio profiles and a heat balance of the dehumidifier. Input parameters, hourly solar data and collector performance are also output for verification.

Secondly, the MAIN subprogram specifies which program option and collector type will be utilized. There are two program options available which involve the specification of dehumidifier regeneration temperature. This parameter may be specified as a constant most useful during preliminary design procedures utilizing an auxiliary heater with a downstream

thermostat. The regeneration temperature may also be specified as an hourly value which will be the instance where no thermostat is installed and the outlet collector temperature is known prior to dehumidifier design.

Four collector types may be specified. A type zero collector indicates manufacturers performance data for collector efficiency have been input into the program. These equations provide values of $FR\tau\alpha$, $FRUL$, and any efficiency modifiers required by the manufacture.

Three collector types may be specified in which the efficiency is theoretically determined using techniques outlined by Beckman and Duffie (3) (12). A type 1 collector is a water collector with bonded tubes and specified air to water heat exchanger. Type 2 and 3 collectors are air type collectors with metal/glass and metal/metal flow channels, respectively.

The third task accomplished by the MAIN subprogram is it serves as a data base and a source for calling the HUMID, COLPER, and HEATX subroutines representing the three major subcomponents of the dehumidifier.

HUMID: The HUMID subroutine is an analytical model of the desiccant wheel component of the dehumidification system.

The methods for calculating the performance of a silica gel desiccant bed have lagged significantly behind the commercial use of silica gel dehumidifiers. The modeling of the dehumidification process involves the simultaneous transfer of heat and mass between the desiccant and air stream. Systems of equations modeling this process are difficult to describe and solve theoretically owing to their non-linear and coupled characteristics. This non-linearity is a direct result of the complex thermal and sorptive

properties of the air/water mixture and desiccant. For this reason initial analyses of the dehumidification process were either empirical in nature or involved linearization by considering only isothermal dehumidification.

One of the first attempts to determine an analytical solution to this problem was Hougen and Marshall (18) in 1949. This research provided a method of graphically calculating the time-position-temperature-concentration conditions in both the gas and solid. The process was assumed to be isothermal, the bed stationary, and a linear equilibrium relation was assumed to exist between the adsorbate contained in the gas the solid absorbent.

A method of analysis of an adsorption system was developed based upon empirical relationships by Ross and McLaughlin (38) in 1955. Taking experimental data from a stationary silica gel bed, Ross and McLaughlin were able to predict bed parameters such as break point time, absorption efficiency, temperature rise, etc. based upon inlet air temperature, relative humidity, air flow, and bed dimensions. Adsorption efficiency is defined as the ratio of the amount of water adsorbed by the desiccant in a given time to the amount of water vapor in the air entering the desiccant bed during that time. The adsorption efficiency of a desiccant bed remains relatively constant from the beginning of the adsorption cycle until some later point when the desiccant bed becomes nearly completely saturated resulting in a sharp decrease in adsorption efficiency. This point is termed breakpoint or breakpoint time.

Since these initial attempts at modeling the dehumidification of air by silica gel, the majority of theoretical effort has been concentrated

on two techniques. The first technique was originally developed by Bullock and Threlkeld (5) for isothermal dehumidification. A series of partial differential equations resulting from an energy balance, water mass balance, and heat and mass transfer relationships were solved numerically using a given set of boundary conditions. The results were compared to experimental data taken for a stationary silica gel bed and were found to be very accurate.

Chi and Wasan (7) expanded Bullock and Threlkeld's model to include adiabatic dehumidification for stationary beds of hygroscopic salts on a support matrix. Again the results were verified by comparing the calculated values to Chi's experimental work for lithium chloride on Torrex. Pla Barby (36) further refined Bullocks and Chi's work by expanding it to include rotating beds.

The greatest advantage of this numerical analysis method is that temperature and moisture content of the desiccant and air/water mixture may be determined at any time step and at any bed thickness. Other advantages of this method are the proven numerical techniques used (i.e. predictor-corrector and modified Euler) and the excellent correlation between measured and calculated values.

The second major method utilized to predict dehumidifier performance was initially developed by Banks (1) in Australia in 1971. This approximate method transforms the mass and energy conservation equations into two potential kinematic wave equations. These equations are linearized by assuming the potentials are linear functions of temperature and humidity. This allows constant potential lines to be drawn on a psychrometric chart from known physical properties of the desiccant. These potentials are analogous to temperature if heat transfer alone is considered.

By incorporating rate equations of heat and mass transfer for silica gel and an air-water mixture, Close and Banks (9) (10) were able to theoretically determine the outlet conditions of a stationary silica gel desiccant bed. Assuming an initially steady state desiccant bed disrupted by a step change in inlet condition, the Bank and Close model predicts a wave front will pass through the bed resulting in new outlet conditions. These conditions will remain constant until the second wave passes through the bed leaving the outlet conditions identical to the inlet conditions.

This model was further developed by MacLaine-Cross and Banks (29) to include rotating desiccant wheels. Since the heat and mass transfer potentials are analogous to temperature when heat transfer alone is considered, they were able to apply these newly developed potentials to sensible rotating heat exchanger theory. Using correlations similar to parameters developed by Kays and London (22) (23) for rotating sensible heat exchanger effectiveness, MacLaine-Cross and Banks were able to predict the performance of a rotating desiccant wheel.

Nelson (34) computerized the MacLaine-Cross and Banks method to provide outlet process and regeneration temperatures similar to the Pla Barby model. The MacLaine-Cross and Banks model does not provide time-temperature-position results as does the Pla Barby model, however this information is not necessary for use in the solar dehumidification model. Since the Pla Barby computer model requires significantly greater storage capacity and calculations, as the result of the time-temperature-position information, this model requires significantly

more computer cost per run. The Pla Barby model requires some 200 to 1000 seconds of computation time for one desiccant bed analysis, while the MacLaine-Cross and Banks model only requires 1 to 2 seconds of computation time.

As a result of the savings in computation time with no significant difference in required output information, the MacLaine-Cross and Banks model and Nelson computer model are used as the bases of the HUMID subprogram with only minor revisions. The HUMID subroutine is supported by two subprograms ALFAV and EGV which provide parameter information based upon properties of silica gel.

A comparison of the output results for the Pla Barby, MacLaine-Cross and Banks, and HUMID subroutine is shown in Table 1. The silica gel desiccant bed modeled is 50 mm (2.0 in.) in thickness. The inlet temperature and humidity ratio for the process side are 26.2°C (79.2°F) and 10.1 g/kg, respectively. The regeneration air flow temperature and humidity ratio are 61.1°C (141.9°F) and 18.2 g/kg, respectively. Rotating speed of the desiccant wheel is 18.1 minutes per revolution. Between all three models the maximum variation between temperatures is 3°C or about 7 per cent and 1 g/kg or 4 per cent for absolute humidity.

LEAK: Since the desiccant bed and the sensible heat exchanger are packed beds the pressure drop through each is significant when compared to the total fan static pressure. This high static pressure (3 to 5 in wg) through each wheel at 23.6 l/s (50 cfm) results in a strong tendency for the air flow streams to leak around the beds through the seals in lieu of passing through the beds. The LEAK subroutine models the leakage process of both the desiccant bed and heat exchanger wheel.

Table 1. Comparison Between HUMID Subprogram, Pla Barby Model, and MacLaine-Cross and Banks Model (36) (29).

Property	Mac Laine-Cross and Banks	Pla-Barby	HUMID
Outlet Process Air Temp - °C	41	38	41
Outlet Regeneration Air Temp - °C	47	49	46
Outlet Process Air Humidity - g/kg	6	7	6
Outlet Regeneration Air Humidity - g/kg	22	21	23

Leakage is expected to occur by three methods. First bypass leakage may occur when inlet air bypasses the complete dehumidification process by flowing across one of the crossflow seals rather than passing through the bed. Secondly, recirculation leakage may result when the air water-vapor mixture, after passing through one bed, would prefer to recirculate to the other flow stream by passing through a crossover seal rather than passing through the second bed. Finally, periphery leakage is a result of leakage around a bed by passing through the periphery seals. It should be noted that recirculation and bypass leakage result in the passage of some of the air water mixture from one flow stream to the other while periphery leakage does not.

A theoretical derivation of the equations modeling the leakage processes is included in Appendix C. If only a desiccant wheel is used in the dehumidification process, the LEAK subprogram will select the proper leakage equations by inputting a zero value for the sensible heat exchanger effectiveness.

COLPER: The COLPER subprogram determines the performance or useful heat gain of the solar collector. After initialization of parameters, the COLPER subroutine determines the amount of radiation striking the collector,utilizing ambient conditions, collector orientation, time of day, and time of year. The computer model follows closely the equations developed by Beckman and Duffie (3).

Knowing the incident radiation on the surface of the collector, the COLPER subprogram calls the COLEFF subroutine for the appropriate collector efficiency given ambient conditions, inlet collector conditions

and loss coefficients. Given this efficiency, the collector useful heat gain and outlet temperature are determined.

The COLPER subprogram also determines the amount of heat lost through the ductwork from the collector to the dehumidifier given the estimated temperature drops. If the temperature returning to the dehumidifier is below the required inlet regeneration temperature, the COLPER subprogram determines the amount of auxiliary heat required to assure proper regeneration temperature. If the return temperature is greater than the required regeneration temperature, the COLPER subprogram determines the amount of heat available for storage.

The final purpose of the COLPER subprogram is to assure proper collector parameter input by descriptively outputting each input collector parameter.

COLEFF: The COLEFF serves as a support program for the COLPER subprogram. This subprogram has the capacity to determine the collector efficiency for any of four given collector types specified in the MAIN subprogram.

The zero type collector indicates that empirical correlations between FR_{UL} , $FR_{T\alpha}$, efficiency modifiers, and volumetric air flow per collector area developed from test data have been input. Once the values of $FR_{T\alpha}$ and FR_{UL} have been determined the collector efficiency is calculated by the following equation.

$$\eta = MOD \left[FR_{T\alpha} - FR_{UL} \left(\frac{T_{in} - T_a}{HR} \right) \right]$$

The remaining three collector types permit the collector efficiency to be determined theoretically. One of the collector types is a water

collector coupled to a air to water heat exchanger. The remaining two collector types are air collectors. For all these collector types ambient temperature, solar radiation, and wind speed are required input parameters. Also collector air flow rate, collector area and length, collector transmittance, emittance, and absorptance are all required input. The number of collector covers and bottom loss coefficient must be input for all theoretically calculated collector efficiencies.

In all collector geometries, the top heat loss coefficient is initially calculated assuming a collector absorber plate temperature. Values of the collector heat removal factor, FR , and the collector efficiency factor, F' , are then determined for each collector geometry using the unique correlations for that particular geometry (3). Knowing the values of FR and F' the collector efficiency and useful heat gain are determined. Once the useful heat gain is calculated a new value of the average absorber plate temperature may be determined and compared to the assumed value. If the difference is greater than 0.5°C the entire process is reiterated using the new absorber plate temperature.

In the special water collector geometry, an air to water counterflow heat exchanger effectiveness and water flow rate must be specified. This results in the regeneration flow stream from the dehumidifier entering the air side of the heat exchanger and leaving the heat exchanger to return to the dehumidifier.

The final step of the COLEFF subroutine is to assure the collector heat gain is greater than zero. If the heat gain is less than zero the collector is considered not operable and efficiency and heat gain are set equal to zero.

HEATX: The final program subroutine in the program models the sensible heat exchanger of the dehumidification system. Given the heat exchanger effectiveness, the program determines the corresponding outlet temperatures using the following equations

$$\epsilon = \frac{T_{cout} - T_{cin}}{T_{hin} - T_{cin}}$$

$$\text{and } \epsilon = \frac{T_{hin} - T_{hout}}{T_{hin} - T_{cin}} .$$

Knowing these temperatures the LEAK subprogram is called to determine the actual outlet temperature and humidity ratios accounting for leakage. The heat transfer by the sensible heat exchange may then be determined from these values by calculating the change of enthalpy for one of the flow paths. It should be noted that dehumidification systems not utilizing a sensible heat exchanger may be modeled by setting the heat exchanger effectiveness equal to zero.

CHAPTER V

EXPERIMENTAL RESULTS AND DISCUSSION

The experimental apparatus was constructed to permit concurrent operation of the desiccant and heat exchanger wheels. During preliminary testing however, the operation of both wheels produced such large leakage rates and low air stream flow rates that it was decided to concentrate the data collection solely around the desiccant wheel. Since the operation of the sensible heat exchanger is more widely understood and predictable, the computer modeling of the heat exchanger is considered acceptable and its inclusion into the dehumidification system not critical for proper operation.

Data Taking Procedures.

Because the operation of the experimental apparatus is dependent upon the presence of sunshine and other ambient conditions, the everyday operation of the experimental apparatus was not possible. When ambient conditions in the early morning hours did appear promising the following data taking procedures were followed.

Approximately two hours prior to actual data acquisition, the desiccant wheel drive motor, blower, and all instrumentation were turned on to allow proper warm up of all equipment and to promote steady state operation of the entire dehumidification system.

After proper warm up, the desiccant wheel drive motor was set at the proper rotating speed by adjusting the motor controller. The guillotine dampers, located in the fan discharge plenum, were then modulated to provide balanced air flow to the process and regeneration flow streams.

Prior to any data recording, the data logger was checked for accurate time of day and all instrument readings were examined for reasonableness and consistency. The dew point meter was zeroed, balanced, and placed into operation, measuring the ambient dew point. This indication was checked against values obtained by a sling psychrometer. If the two readings did not agree within approximately 1°C , the dew point meter was rebalanced or disassembled and cleaned. Should ambient conditions be such that vapor condensation could occur in the dew point sampling lines, a heater tape was placed into service to maintain the local sampling line temperature above the dew point. The timer controlling the five way valve used to systematically sample each dew point line was synchronized with the data logger. This assured that the dew point reading was recorded approximately 1 to 2 minutes before the valve was rotated to a new sampling line. Strip chart recorders for solar radiation and wind speed and direction were also synchronized with the data logger.

The actual recording of data information was begun on the hour, permitting 120 readings of all thermocouples and pyranometer inputs to be taken each hour. The average value of these 120 readings was computed and printed by the data logger. A dew point value was printed out every fifteen minutes, starting 30 minutes after the beginning of the test. Thus the four dew point measurements were printed at 30 and 45 minutes after the hour, on the hour, and 15 minutes after the average thermocouple values were printed. The last dew point was taken 15 minutes after the thermocouple averages were printed because it better represented the effect of the preceding hour due to the lagging dehumidification characteristics of a rotating desiccant bed. Continuously recorded on

strip charts were values of wind speed and direction, and solar radiation in the plane of the collector.

All data recorded by the data logger for May 1, 5, 6, and 10, 1981 are shown in Appendix A. The errors associated with their measurement are discussed in Appendix D. Because the temperatures at each sampling point were determined by at least two independent thermocouples, each thermocouple reading is given in the appendix and the average value computed. Very little discrepancy between thermocouples located at the same sampling point were noted, with one exception. As the process flow stream exits the desiccant wheel a variation of $\pm 10^{\circ}\text{C}$ is indicated. It is expected that this may be the effect of stratification in this flow chamber. As the regenerated desiccant wheel enters the process flow stream, heating of the process air is not only due to the effects of dehumidification but also the cooling of the hot desiccant wheel. If this air remains stratified the local thermocouple will indicate high temperature reading. The opposite effect may occur as the desiccant wheel is about to reenter the regeneration flowstream. Since the desiccant bed may be nearing its dehumidification capacity the process air passing through will be only partially dehumidified resulting in a lower outlet temperature.

The only other significant variation in measured values is between the inlet dew points of the regeneration and process flow streams. Both of these points were ambient conditions so they could have had the same value. However, these two dew point readings were taken 30 minutes apart and ambient conditions could have changed in this time span. Also the dew point meter itself was observed to oscillate as much as $\pm .5^{\circ}\text{C}$ prior

to most readings. This could also result in significantly different dew points being recorded for the same conditions.

Comparison of Experimental Results with Computed Values.

From the experimental data shown in Appendix A, the validity of the computer model can be analyzed. The computer program requires the following inputs to determine dehumidification system performance. These inputs include, collector and desiccant bed parameters, time of day, leakage rates, ambient temperature and dew point (humidity ratio), wind speed, and solar radiation. The results of these computer simulations are compared to the experimental data in Table 2 through Table 11. Experimental and calculated desiccant bed inlet and outlet conditions are shown in Tables 2 through 9. Experimental and calculated solar collector inlet and outlet conditions are compared in Tables 10 and 11.

It should be noted that the three leakage factors: BLEAK, PLEAK, and RLEAK can make a significant change in the predicted outlet temperatures and humidity ratios. The actual effect of these three leakages is unknown, because time did not permit their experimental determination. However, the three leakage rates may be varied independently or together in the computer model. This was done and the results for a typical hour, 14:00 CDT May 5, 1981, are shown in Table 12. The base case is evaluated for the following leakage rates; BLEAK and RLEAK equal to 40 per cent, and PLEAK equal to 20 per cent. It is obvious from these results that each leakage rate is coupled to all four outlet conditions and changing one leakage rate may improve the correlation between one temperature and humidity ratio while decreasing the correlation

Table 2. Comparison of Experimental and Calculated Values of the Dehumidifier Process Outlet
Conditions for May 1, 1981.

<u>Process Side</u>	<u>Outlet Temperature</u>				<u>Outlet Humidity Ratio</u>			
	<u>T_{in}</u> (°C)	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)	<u>W_{in}</u> (g/kg)*	<u>W_{exp}</u> (g/kg)	<u>W_{cal}</u> (g/kg)	<u>ΔW</u> (g/kg)
<u>TIME-CDT</u>								
14:00	27.4	43.2	45.8	2.6	3.1	2.4	2.1	-0.3
15:00	27.5	42.0	46.4	4.4	3.0	2.2	2.0	-0.2
16:00	26.4	41.0	45.5	4.5	2.9	2.3	2.0	-0.3
17:00	25.4	37.9	42.5	4.6	2.7	2.4	1.9	-0.5
18:00	24.6	36.7	38.2	1.5	2.7	2.3	1.9	-0.4
19:00	23.7	30.1	32.0	1.9	2.4	2.3	1.9	-0.4

* Units of humidity ratio are grams of water per kilogram of air.

Table 3. Comparison of Experimental and Calculated Values of the Dehumidifier Regeneration Outlet
Conditions for May 1, 1981.

<u>Regeneration Side</u>	<u>Outlet Temperature</u>			<u>Outlet Humidity Ratio</u>				
	<u>T_{in}</u> (°C)	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)	<u>W_{in}</u> (g/kg) [*]	<u>W_{exp}</u> (g/kg)	<u>W_{cal}</u> (g/kg)	<u>ΔW</u> (g/kg)
<u>Time-CDT</u>								
14:00	69.6	40.8	48.1	7.3	3.2	3.5	4.1	0.6
15:00	70.5	41.8	49.0	7.2	3.1	3.5	4.0	0.5
16:00	68.3	40.6	48.0	7.4	2.9	3.3	3.8	0.5
17:00	61.4	38.1	44.3	6.2	2.6	3.1	3.4	0.3
18:00	52.3	31.5	38.9	7.4	2.4	2.6	3.3	0.7
19:00	39.6	30.1	32.0	1.9	2.4	2.6	2.8	0.2

* Units of humidity ratio are grams of water per kilogram of air.

Table 4. Comparison of Experimental and Calculated Values of the Dehumidifier Process Outlet Conditions for May 5, 1981.

<u>Process Side</u>	<u>Outlet Temperature</u>				<u>Outlet Humidity Ratio</u>			
	<u>T_{in}</u> (°C)	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)	<u>W_{in}</u> (g/kg)*	<u>W_{exp}</u> (g/kg)	<u>W_{cal}</u> (g/kg)	<u>ΔW</u> (g/kg)
<u>Time-CDT</u>								
13:00	23.1	45.3	50.3	5.0	8.2	6.2	5.5	-0.7
14:00	23.7	46.5	50.3	3.8	7.3	5.4	4.9	-0.5
15:00	23.4	44.9	49.2	4.3	7.2	5.1	4.9	-0.2
16:00	22.7	42.4	47.8	5.4	7.5	5.2	5.2	0.0
17:00	22.0	38.7	44.5	5.8	7.3	5.2	5.2	0.0

* Units of humidity ratio are grams of water per kilogram of air.

Table 5. Comparison of Experimental and Calculated Values of the Dehumidifier Regeneration Outlet Conditions for May 5, 1981.

<u>Regeneration Side</u>	<u>Outlet Temperature</u>				<u>Outlet Humidity Ratio</u>			
	<u>T_{in}</u> (°C)	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)	<u>W_{in}</u> [*] (g/kg)	<u>W_{exp}</u> (g/kg)	<u>W_{cal}</u> (g/kg)	<u>ΔW</u> (g/kg)
<u>Time-CDT</u>								
13:00	73.3	36.1	43.1	7.0	8.0	9.6	11.1	1.5
14:00	74.2	37.7	44.9	7.2	8.0	9.2	9.9	0.7
15:00	70.9	36.7	43.8	7.1	7.7	10.1	9.8	-0.3
16:00	67.2	35.4	41.3	5.9	7.3	8.9	10.2	1.3
17:00	60.9	32.9	38.2	5.3	7.6	8.3	9.7	1.4

* Units of humidity ratio are grams of water per kilogram of air.

Table 6. Comparison of Experimental and Calculated Values of the Dehumidifier Process Outlet
Conditions for May 6, 1981.

<u>Process Side</u>	<u>Outlet Temperature</u>			<u>Outlet Humidity Ratio</u>		
	<u>T_{in}</u> (°C)	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)	<u>W_{in}</u> (g/kg)*	<u>ΔW</u> (g/kg)
<u>Time-CDT</u>						
14:00	19.1	41.8	45.2	3.4	6.8	-0.1
15:00	20.7	48.4	43.7	-4.7	6.3	0.1
16:00	20.1	45.6	46.5	0.9	5.6	-0.2
17:00	20.0	43.8	45.5	1.7	5.4	-0.2
18:00	19.7	41.3	42.3	1.0	4.7	0.0
19:00	21.0	36.6	38.1	1.5	4.2	0.0

*Units of humidity ratio are grams of water per kilogram of air.

Table 7. Comparison of Experimental and Calculated Values of the Dehumidifier Regeneration Outlet Conditions for May 6, 1981.

<u>Time-CDT</u>	<u>Outlet Temperature</u>				<u>Outlet Humidity Ratio</u>			
	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)
14:00	66.8	30.1	40.0	9.9	7.0	8.2	9.4	1.2
15:00	78.8	29.4	39.5	10.1	6.3	8.1	8.2	0.1
16:00	70.0	33.9	43.2	9.3	5.8	6.9	7.7	0.8
17:00	67.8	32.9	42.2	9.3	5.6	6.7	7.4	0.7
18:00	62.6	31.7	40.2	8.5	5.0	6.0	6.4	0.4
19:00	54.4	28.9	36.1	4.2	4.2	5.5	5.5	0.0

* Units of humidity ratio are grams of water per kilogram of air.

Table 8. Comparison of Experimental and Calculated Values of the Dehumidifier Process Outlet Conditions for May 10, 1981.

<u>Process Side</u>	<u>Outlet Temperature</u>				<u>Outlet Humidity Ratio</u>			
	<u>T_{in}</u> (°C)	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)	<u>W_{in}</u> (g/kg)*	<u>W_{exp}</u> (g/kg)	<u>W_{cal}</u> (g/kg)	<u>ΔW</u> (g/kg)
<u>Time-CDT</u>								
12:00	22.1	42.4	45.3	2.9	3.1	2.4	1.9	-0.5
13:00	23.0	42.4	44.3	1.9	3.3	2.0	2.2	0.2
14:00	23.3	40.8	45.3	4.5	3.3	1.9	2.1	0.2
15:00	24.0	41.3	46.7	5.4	3.2	1.8	2.0	0.2
16:00	23.1	40.6	46.4	6.2	3.3	1.8	2.1	0.3
17:00	22.6	40.8	47.5	6.7	3.6	2.1	2.4	0.3
18:00	22.2	38.6	44.0	5.4	3.5	2.2	2.4	0.2

* Units of humidity ratio are grams of water per kilogram of air.

Table 9. Comparison of Experimental and Calculated Values of the Dehumidifier Regeneration Outlet

Conditions for May 10, 1981.

<u>Regeneration Side</u>									
<u>Time-CDT</u>	<u>Outlet Temperature</u>				<u>Outlet Humidity Ratio</u>				
	<u>T_{in}</u> (°C)	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)	<u>W_{in}</u> (g/kg) *	<u>W_{exp}</u> (g/kg)	<u>W_{cal}</u> (g/kg)	<u>ΔW</u> (g/kg)	
12:00	74.3	35.9	48.8	12.9	3.9	4.6	4.2	-0.4	
13:00	71.0	36.5	46.8	10.3	3.9	4.3	4.4	0.1	
14:00	71.7	36.7	47.5	10.8	3.6	3.8	4.5	0.7	
15:00	72.6	37.8	49.2	11.4	3.5	4.0	4.2	0.2	
16:00	71.1	36.8	49.4	12.6	3.5	4.3	4.0	-0.3	
17:00	73.4	37.1	49.3	12.2	3.7	4.3	4.8	0.5	
18:00	65.8	34.8	45.3	10.5	3.8	4.2	4.6	0.4	

* Units of humidity ratio are grams of water per kilogram of air.

Table 10. Comparison of Experimental and Calculated Values of Solar Collector Performance for May 1 and May 5, 1981.

Time (CDT)	Inlet Temperature			Outlet Temperature		
	T_{exp} (°C)	T_{cal} (°C)	ΔT (°C)	T_{exp} (°C)	T_{cal} (°C)	ΔT (°C)
May 1, 1981						
14:00	25.8	24.0	-1.8	72.6	71.3	-1.3
15:00	26.6	24.5	-1.9	72.3	69.1	-3.2
16:00	26.3	25.0	-1.3	68.1	64.8	-3.3
17:00	26.4	25.2	-1.2	59.2	53.7	-5.5
18:00	26.0	24.3	-1.7	52.3	26.1	-26.2
19:00	24.6	24.1	-0.5	33.0	24.1	-8.9
May 5, 1981						
13:00	21.6	20.8	-0.8	67.1	62.8	-4.3
14:00	22.8	21.7	-1.1	68.6	65.6	-3.0
15:00	23.1	22.4	-0.7	61.6	59.0	-2.6
16:00	22.7	22.7	0.0	54.8	50.5	-4.3
17:00	22.4	22.6	+0.2	44.4	39.4	-5.0

Table 11. Comparison of Experimental and Calculated Values of Solar Collector Performance for May 6 and May 10, 1981.

<u>Time</u> (CDT)	<u>Inlet Temperature</u>			<u>Outlet Temperature</u>		
	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)	<u>T_{exp}</u> (°C)	<u>T_{cal}</u> (°C)	<u>ΔT</u> (°C)
May 6, 1981						
14:00	19.1	18.9	-0.2	44.4	43.8	-0.6
15:00	20.7	19.6	-1.1	64.2	69.5	+5.3
16:00	20.4	20.2	-0.2	59.3	51.8	-7.5
17:00	21.2	20.7	-0.5	55.2	50.3	-4.9
18:00	20.8	20.2	-0.6	44.5	21.2	-23.3
19:00	19.6	20.2	+0.6	30.2	20.2	-10.0
May 10, 1981						
12:00	21.2	19.3	-1.9	65.5	62.1	-3.4
13:00	22.6	20.2	-2.4	71.7	66.9	-4.8
14:00	23.2	21.0	-2.2	72.4	68.9	-3.5
15:00	23.9	22.7	-1.2	71.4	67.9	-3.5
16:00	23.2	23.1	-0.1	66.0	62.2	-3.8
17:00	23.6	23.4	-0.2	56.4	50.8	-5.6
18:00	23.0	23.1	+0.1	43.7	23.4	-20.3

Table 12. Comparison of the Effect of Leakage Rates on Typical
Calculated Values of Outlet Temperatures and Humidity
Ratios.*

Leakage Rates

BLEAK	Measured	40	55	40
PLEAK		20	20	20
RLEAK		40	40	55

Process Side

Desiccant Bed

T_{in}	23.7	21.7	21.7	21.7
T_{out}	46.5	50.3	53.4	52.0
W_{in}	7.3	7.3	7.3	7.3
W_{out}	5.4	4.9	5.2	5.5

Regeneration Side

Desiccant Bed

T_{in}	74.2	74.2	74.2	74.2
T_{out}	37.7	44.9	43.4	41.8
W_{in}	8.0	7.3	7.3	7.3
W_{out}	9.2	9.9	9.2	9.6

*Temperature shown in °C, humidity ratios shown in g water per kg of
dry air.

of another property or at another test point. After comparing these results at several test points it was decided that the base case of: 40 per cent leakage for BLEAK and RLEAK, and 20 per cent leakage for PLEAK seems to compare very favorably for several test points. These values are used to evaluate all the test points shown in Tables 2 through 11.

In reviewing Tables 2 through 9, it is noted that the predicted outlet temperatures leaving the desiccant bed are always greater than the experimental values. This is especially true of the regeneration airstream. The reason for this discrepancy is not completely understood. One possible source of this difference may be related to the heat transfer through the insulated compartment walls, not included in the computer program. A theoretical analysis indicates the effect of this heat transfer should be less than 1°C (1.8°F). Another possible cause of this discrepancy may be in silica gel properties used in the program. The silica gel used in the experimental apparatus is 3-8 mesh, however input parameters required by the program are only available for mesh sizes 2-4, 4-6, and 6-8, therefore a weighted average was used for the 3-8 mesh. It is not known if this type of averaging is permissible in evaluating desiccant performance or if the silica gel utilized is equivalent to that used by Hubbard (19) in determining silica gel properties. The heat transfer effects of the top and bottom perforated aluminum sheets are also not included in the program model. The program assumes the heat transfer between the support material and the air stream will be insignificant when compared to the effects of the desiccant material. Should this assumption be grossly incorrect it is expected that the

calculated outlet process air temperature would be significantly lower than the experimental value due to the under estimation of the heat transferred. This is not the case however, shown by the experimental data. A final source of error may be the result of incorrectly estimating the leakage rates as discussed previously.

Despite the differences between the calculated and experimental values for outlet temperatures the computer program is considered to yield an acceptable estimation of these values.

Also included in Tables 2 through 9 is a comparison of experimental and predicted dew points. These values are shown to correlate very well for the conditions tested. This is especially significant considering the oscillation of the dewpoint meter of $\pm .5^{\circ}\text{C}$ may result in a variation in humidity ratios of ± 1.0 g/kg. Also, the calculated humidity ratio may be determined from ambient conditions taken as much as 45 minutes prior to the time the experimental value was taken. Again the effects of leakage rates may also cause some variation between actual and calculated values.

The comparison of experimental and calculated values of solar collector temperatures are shown on Tables 10 and 11. A temperature rise of 3°C (5.4°F) is estimated to occur between the ambient conditions and collector inlet due to fan work input. This approximation is shown to be within about $+ 0.5^{\circ}\text{C}$ and -2.0°C of the experimental data. It should be noted that the 3.0°C estimation becomes a better approximation as the time in operation increases.

In most instances, the outlet air collector temperature is underestimated approximately 3°C (5.4°F) to 5°C (8.0°F) by the program. This approximation is considered very satisfactory since outlet temperatures may be as high as 72°C (162°F). Significant underestimations, 20°C (36°F), of the outlet collector temperature are shown to exist consistently at approximately 18:00. This is a direct result of the empirical relationships supplied by the collector manufacturer and is not a result of program error. The manufacturers relationships for determining collector efficiency are only given for zenith angles less than 1 rad. (60°) or approximately 16:00 solar time. As the sun is setting or rising the determination of collector efficiency requires a severe extrapolation of data and results in a significant underestimation of the collector efficiency.

The misestimation of seal leakage rates may also have an effect on the calculation of outlet collector temperature, because collector efficiency is a direct function of air volume flow rate, therefore if the bypass leakage rate is overestimated or the recirculation leakage underestimated the collector efficiency will be underestimated.

It should be noted that only one type of collector, Solaron Series 3000, was utilized during the entire experimental procedure. This did not permit the experimental verification of each type of collector modeled by the computer program. Computer models of these collectors were however developed and the results were as expected for these collectors. The actual verification of these collector models should be the subject of further investigation.

No comparison between the results reported here and other investigators were made. This was due to the limited amount of previous research done in the area of dehumidification utilizing rotating desiccant beds. Pla Barby (36) was the only investigator found to present experimental data for these types of beds. However, the systems investigated were either of unbalanced flow rates or provided insufficient operating parameters to allow any direct comparison between his results and the predictions made by this program.

CHAPTER VI

CONCLUSIONS

From this research it was found:

- 1) A dehumidification system employing a solid rotating bed of silica gel was found to exhibit several characteristics which may make it a preferred method of dehumidification utilizing solar energy.
- 2) The mathematical model developed by MacLaine-Cross and Banks provides acceptable predictions of the outlet conditions of a rotating desiccant bed.
- 3) The complete computer model produces a satisfactory analysis of a solar, rotating desiccant bed, dehumidification system.
- 4) The use of various wheel leakage rates can significantly change the predicted values of outlet desiccant temperature and humidity ratio.

CHAPTER VII

RECOMMENDATIONS OF FURTHER STUDY

As a result of the experimental and theoretical research reported here the following recommendations are made:

- 1) Further test points need to be analyzed with the current experimental apparatus, especially over a wider range of air flow rates and ambient humidity ratios and temperatures.
- 2) Variation of physical bed parameters such as wheel speed and bed depths and diameters need to be investigated to further verify the computer program.
- 3) A complete theoretical or experimental evaluation of the wheel seal leakage rates needs to be performed to accurately determine them.
- 4) If better wheel seals can be developed, experimental tests employing the sensible heat exchanger should be performed.
- 5) Different types of air-cooled collectors and liquid-cooled collectors should be employed to verify these portions of the computer program.
- 6) The computer program should be expanded such that the modeling of dehumidification systems with differing airstream flow rates and airstreams with differing inlet conditions are permitted.
- 7) Following the completion of these other recommendations, an economic evaluation of the dehumidification system should be performed utilizing the auxiliary heating requirements output by the program.

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APPENDIX A

EXPERIMENTAL DATA

Table A.1 Dehumidifier Parameters, Solar Data, and Dew Points for May 1, 1981

Dehumidifier Parameters

Process Side Air Flow Rate-49.0 cfm

Regeneration Side Air Flow Rate-54.0 cfm

Wheel Speed- 650 sec/rev

Solar Data and Dew Points

Time (CDT)	Radiation (Btu/hr-ft ²)	Wind Speed (mph)	Process Side Dewpoint		Regeneration Side Dewpoint	
			Inlet (°C)	Outlet (°C)	Inlet (°C)	Outlet (°C)
14:00	311.5	8.0	-2.4	-5.7	-2.2	-1.1
15:00	293.5	7.0	-2.8	-5.9	-2.6	-1.3
16:00	252.9	6.0	-3.1	-5.9	-3.3	-1.8
17:00	194.9	5.0	-4.2	-5.3	-4.5	-2.5
18:00	124.1	4.0	-4.0	-5.9	-5.5	-4.5
19:00	53.4	4.0	-5.2	-5.8	-5.2	-4.3

Table A.2 Process Side Temperatures for May 1, 1981

Process Side Temperatures- (°F)		Dehumidifier								
T/C Ref. No.	Ambient	Intet			Outlet					
		0	1	Ave.	2	3	4	5	Ave.	
Time-(CDT)	23									
14:00	69.6	81.5	81.4	81.4	107.1	114.3	99.9	107.7		107.3
15:00	71.1	81.6	81.5	81.5	107.3	114.9	100.3	108.0		107.6
16:00	71.1	79.7	79.6	79.6	105.9	112.5	98.3	106.6		105.8
17:00	72.0	77.9	77.8	77.8	99.8	106.7	93.8	100.4		100.2
18:00	70.7	76.4	76.3	76.3	94.3	99.1	89.1	94.9		94.4
19:00	69.8	74.8	74.6	74.7	85.9	89.1	83.3	86.3		86.2

ILLEGIBLE DOCUMENT

**THE FOLLOWING
DOCUMENT(S) IS OF
POOR LEGIBILITY IN
THE ORIGINAL**

**THIS IS THE BEST
COPY AVAILABLE**

Table A.3 Regeneration Side Temperatures for May 1, 1981

Regeneration Side Temperatures- (°f)												
T/C Ref. No.	Solar Collector			Auxiliary Heater			Dehumidifier					
	Inlet		Outlet	Inlet		Outlet	Inlet		Outlet			
	16	17		20	21		10	11	Ave.	12	13	Ave.
Time-(C01)												
14:00	78.3	78.5	78.4	161.9	162.4	162.2	157.6	156.9	157.3	106.4	104.5	105.5
15:00	79.9	80.0	79.9	161.9	162.4	162.2	159.2	158.9	159.0	103.3	106.3	107.3
16:00	79.4	79.5	79.4	155.0	154.6	154.8	155.2	155.0	155.1	106.0	104.3	105.1
17:00	79.5	79.6	79.5	138.4	138.6	138.5	142.6	142.4	142.5	101.1	99.8	100.5
18:00	78.7	78.9	78.8	117.2	117.4	117.3	126.1	126.1	126.1	94.6	93.8	94.2
19:00	76.2	76.4	76.3	91.4	91.5	91.4	103.4	103.3	103.3	86.2	85.9	86.1

Table A.4 Dehumidifier Parameters, Solar Data, and Dew Points for May 5, 1981

Dehumidifier Parameters

Process Side Air Flow Rate- 48.0 cfm
 Regeneration Side Air Flow Rate- 52.0 cfm
 Wheel Speed- 680 sec/rev

Solar Data and Dew Points

Time (CDT)	Radiation (Btu/hr-ft ²)	Wind Speed (mph)		Process Side Dewpoint		Regeneration Side Dewpoint	
		Inlet (°C)	Outlet (°C)	Inlet (°C)	Outlet (°C)	Inlet (°C)	Outlet (°C)
13:00	279.2	11.0	6.9	10.6	13.3	10.6	13.3
14:00	291.6	9.3	5.0	10.6	12.5	10.6	12.5
15:00	240.2	9.0	4.1	10.1	14.1	10.1	14.1
16:00	182.6	9.8	4.5	9.3	12.3	9.3	12.3
17:00	122.8	9.3	4.5	10.0	11.1	10.0	11.1
18:00	95.2	9.6	4.7	8.8	11.7	8.8	11.7
19:00	46.8	9.8	5.5	9.1	11.0	9.1	11.0

Table A.5 Process Side Temperatures for May 5, 1981

Process Side Temperatures- (°F)		Ambient	Dehumidifier							
			Intet			Outlet				
			0	1	Ave.	2	3	4	5	Ave.
T/C Ref. No.		23								
Time-(CDT)										
13:00		64.0	73.7	73.6	73.6	114.0	122.6	103.2	114.3	113.5
14:00		65.6	74.7	74.6	74.6	116.4	124.6	105.3	116.7	115.7
15:00		66.9	74.2	74.1	74.1	113.4	121.4	103.1	113.7	112.9
16:00		67.6	73.0	72.9	72.9	108.6	116.9	99.2	109.1	108.4
17:00		67.4	71.7	71.5	71.6	101.5	109.3	93.9	102.1	101.7
18:00		67.4	71.4	71.3	71.3	98.2	105.1	91.3	98.8	98.3
19:00		66.8	70.3	70.1	70.2	93.9	100.5	88.3	94.4	94.3

Table A.6 Regeneration Side Temperatures for May 5, 1981

Regeneration Side Temperatures- (°F)

T/C Ref. No. Time - (CET)	Solar Collector				Auxiliary Heater				Dehumidifier									
	Inlet		Outlet		Inlet		Outlet		Inlet			Outlet						
	16	17	Ave.	18	19	Ave.	20	21	Ave.	22	23	Ave.	10	11	Ave.	12	13	Ave.
13:00	71.0	71.1	71.0	152.6	153.0	152.8	150.7	150.9	150.8	162.7	162.6	162.6	164.4	164.0	164.2	98.0	96.1	97.0
14:00	73.0	73.1	73.0	155.2	155.7	155.4	153.1	153.3	153.2	163.8	163.7	163.7	165.8	165.5	165.6	101.1	98.4	99.8
15:00	73.5	73.6	73.5	142.8	143.1	142.9	140.9	141.1	141.0	157.8	157.8	157.8	159.7	159.5	159.6	99.2	96.8	98.0
16:00	72.9	73.0	72.9	130.5	130.7	130.6	129.4	129.6	129.5	151.0	151.1	151.0	153.0	152.8	152.9	96.8	94.8	95.8
17:00	72.3	72.5	72.4	111.8	112.0	111.9	111.5	111.8	111.6	140.0	140.1	140.0	141.8	141.6	141.7	91.9	90.5	91.2
18:00	72.0	72.2	72.1	98.7	98.8	98.7	98.5	98.8	98.6	132.1	132.3	132.2	133.6	133.5	133.5	89.1	88.1	88.6
19:00	70.7	70.9	70.8	84.4	84.6	84.5	84.6	85.0	84.8	124.3	124.5	124.4	125.5	125.4	125.4	86.0	85.4	85.7

Table A.7 Dehumidifier Parameters, Solar Data, and Dew Points for May 6, 1981

Dehumidifier Parameters

Process Side Air Flow Rate- 34.0 cfm

Regeneration Side Air Flow Rate- 35.0 cfm

Wheel Speed- 680 sec/rev

Solar Data and Dew Points

Time (CDT)	Radiation (Btu/hr-ft ²)	Wind Speed (mph)	Process Side Dewpoint		Regeneration Side Dewpoint	
			Inlet (°C)	Outlet (°C)	Inlet (°C)	Outlet (°C)
14:00	142.1	7.0	8.3	2.9	8.8	11.0
15:00	275.6	8.0	6.4	0.2	7.3	10.9
16:00	178.7	7.0	5.6	0.3	5.8	8.4
17:00	185.6	7.0	5.1	0.2	5.5	8.1
18:00	122.6	7.0	2.9	-1.9	4.0	6.6
19:00	59.0	5.0	1.4	-3.1	1.6	5.4

Table A.8 Process Side Temperatures for May 6, 1981

<u>Process Side Temperatures- (°F)</u>		<u>Dehumidifier</u>							
<u>T/C Ref. No.</u>	<u>Ambient</u>	<u>Intet</u>			<u>Outlet</u>				
		0	1	Ave.	2	3	4	5	Ave.
		<u>Time- (CDT)</u>							
14:00	60.7	66.4	66.3	66.3	108.0	114.9	98.1	107.8	107.2
15:00	62.4	69.4	69.3	69.3	120.3	128.7	108.0	119.9	119.2
16:00	63.0	68.2	68.1	68.1	115.2	121.9	104.5	115.0	114.1
17:00	63.9	68.0	67.9	67.9	111.7	118.4	101.5	111.5	110.8
18:00	63.9	67.5	67.3	67.4	107.2	113.8	97.7	107.0	106.4
19:00	63.1	65.9	65.8	65.8	98.4	104.1	91.1	98.3	98.0

Table A.10 Dehumidifier Parameters, Solar Data, and Dew Points for May 10, 1981

Dehumidifier Parameters

Process Side Air Flow Rate-48.0 cfm

Regeneration Side Air Flow Rate- 52.0 cfm

Wheel Speed- 680 sec/rev

Solar Data and Dew Points

Time (CDT)	Radiation (Btu/hr-ft ²)	Wind Speed (mph)	Process Side Dewpoint		Regeneration Side Dewpoint	
			Inlet (°F)	Outlet (°F)	Inlet (°F)	Outlet (°F)
11:00	233.1	5.0	34.6	30.0	41.1	43.7
12:00	281.9	5.0	28.0	21.9	33.2	37.2
13:00	309.4	7.0	29.2	18.7	33.1	35.0
14:00	315.0	7.0	29.0	16.9	30.9	32.0
15:00	293.1	8.0	28.8	16.6	30.0	33.6
16:00	252.6	7.0	29.6	16.1	30.5	35.1
17:00	194.5	7.0	30.9	19.3	31.8	35.3
18:00	122.4	7.0	30.7	20.5	32.7	34.5
19:00	53.7	7.0	32.4	23.0	32.1	34.1

Table A.11 Process Side Temperatures for May 10, 1981

Process Side Temperatures- (°F)

T/C Ref. No. Time-(CDT)	Ambient		Dehumidifier						
	23	Intet			Outlet				Ave.
		0	1	Ave.	2	3	4	5	
11:00	57.1	66.7	66.6	66.6	97.4	106.8	87.0	97.7	97.2
12:00	61.3	71.8	71.7	71.7	108.6	119.2	96.4	109.0	108.3
13:00	63.0	73.5	73.4	73.4	108.1	116.9	96.7	108.5	107.6
14:00	64.4	74.2	74.1	74.1	105.3	115.6	95.0	106.0	105.5
15:00	67.5	75.3	75.2	75.2	105.9	116.8	96.1	106.7	106.4
16:00	68.2	73.6	73.5	73.5	104.8	114.8	94.6	105.6	105.0
17:00	68.7	72.8	72.6	72.7	104.9	116.2	94.9	105.7	105.4
18:00	68.3	72.0	71.9	71.9	101.3	110.7	92.0	101.9	101.5
19:00	67.0	70.0	69.8	69.9	92.4	100.4	86.2	93.0	92.8

Table A.12 Regeneration Side Temperatures for May 10, 1981

Regeneration Side Temperatures- (°F)		May 11, 1981																	
T/C Ref. No.	Time- (Cof)	Solar Collector			Auxiliary Heater			Dehumidifier											
		Inlet		Outlet	Inlet		Outlet	Inlet		Outlet									
		16	17	Ave.	18	19	Ave.	20	21	Ave.	22	23	Ave.	10	11	Ave.	12	13	Ave.
11:00		65.7	65.9	65.8	126.7	127.0	126.8	125.5	125.9	125.7	147.4	147.2	147.3	149.5	149.1	149.3	87.0	85.7	86.3
12:00		70.1	70.3	70.2	149.4	149.9	149.6	146.3	146.6	146.4	162.9	163.0	162.9	166.0	165.6	165.8	98.1	95.3	96.7
13:00		72.5	72.7	72.6	160.9	161.4	161.1	157.7	158.1	157.9	156.3	156.4	156.3	159.9	159.6	159.7	99.2	96.2	97.7
14:00		73.7	73.8	73.7	162.0	162.6	162.3	159.6	159.9	159.7	158.3	158.3	158.3	161.2	160.9	161.0	99.5	96.4	98.0
15:00		75.1	75.2	75.1	160.3	160.8	160.5	158.5	158.9	158.8	160.4	160.8	160.6	162.9	162.6	162.7	101.9	98.2	100.0
16:00		73.8	73.9	73.8	150.6	151.0	150.8	148.3	148.8	148.5	158.0	158.3	158.1	160.0	159.8	159.9	99.8	96.7	98.2
17:00		74.5	74.6	74.5	132.9	134.0	133.5	133.1	133.3	133.2	162.3	162.5	162.4	164.4	164.1	164.2	100.7	96.9	98.3
18:00		73.4	73.5	73.4	110.5	110.8	110.6	110.2	110.5	110.3	148.8	148.9	148.8	150.5	150.4	150.4	96.1	93.1	94.6
19:00		70.7	70.8	70.7	86.3	86.5	86.4	86.5	86.9	86.7	128.2	128.2	128.2	129.4	129.3	129.3	88.4	86.6	87.5

APPENDIX B

COMPUTER PROGRAM AND RESULTS FOR TYPICAL INPUT

A listing of the computer program and typical results are given in this Appendix. The input is the experimental data collected on May 1, 1981, 2:00 pm to 7:00 pm CDT. The program option 1 is employed since the regeneration temperature entering the desiccant wheel is a variable and known value from the recorded data. Since manufacturer's performance curves for the Solaron Series 3000 collectors are input, the program collector type is zero.


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$JUB      RJS,P=40,TIME=(,25),LINES=35
C
C ***** NCMFNCLATURE FOR PROGRAM INPUT PARAMETERS *****
C
C AC- AREA OF COLLECTOR
C AF- TOTAL FRONTAL DESICCANT WHEEL AREA EXPOSED TO FLOW
C AK- THERMAL CONDUCTIVITY OF AIR
C ALNG- COLLECTOR LOCATION LONGITUDE
C AMASS- AIR MASS FLOW RATE THROUGH COLLECTOR
C AMU- VISCOSITY OF AIR
C AV- INTERNAL SURFACE AREA OF DESICCANT PER UNIT VOLUME
C BLEAK- BYPASS LEAKAGE RATE
C CALPHA- COLLECTOR ABSORPTANCE
C CB- CONDUCTANCE FOR TUBE TO ABSORBER PLATE BONDING MATERIAL
C CTAU- TRANSMITTANCE OF COLLECTOR GLAZING
C CTYP- COLLECTOR TYPE SPECIFICATION
C D- HEIGHT OF FLOW CHANNEL FOR AIR TYPE COLLECTORS
C DD- DESICCANT CHARACTERISTIC DIMENSION
C DEL- THICKNESS OF ABSORBER PLATE
C DTDC- TEMPERATURE DROP THROUGH DUCTING, DEHUMIDIFIER TO COLLECTOR
C DTCD- TEMPERATURE DROP THROUGH DUCTING, COLLECTOR TO DEHUMIDIFIER
C E- VALUE FOR EQUATION OF TIME
C EPG- EMITTENCE OF COLLECTOR GLAZING
C EPP- EMITTENCE OF COLLECTOR ABSORBER PLATE
C FL- LENGTH OF DESICCANT WHEEL IN FLOW DIRECTION
C FRIC- DESICCANT FRICTION FACTOR
C H- SOLAR RADIATION ON HORIZONTAL SURFACE, IN KJ/HR M**2
C HFIM- HEAT TRANSFER COEFFICIENT FOR INSIDE OF WATER TUBES
C HMU- AIR TO DESICCANT MASS RATIO IN DESICCANT WHEEL
C ID- INSIDE TUBE DIAMETER FOR WATER TYPE COLLECTOR
C IDATE- NUMBER OF DATE IN YEAR
C IFLAG- PROGRAM OPTION, SPECIFICATION OF REGENERATION TEMPERATURE
C KFIN- CONDUCTANCE OF WATER COLLECTOR TUBE MATERIAL
C LEN- COLLECTOR LENGTH
C MDY- MONTH ,DAY,DATE AND YEAR TO BE ANALYZED

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C NCOVS- NUMBER OF COLLECTOR GLAZINGS
C ND- NUMBER OF DAYS TO BE ANALYZED
C NH- NUMBER OF HOURS TO BE ANALYZED
C GD- OUTSIDE TUBE DIAMETER FOR WATER COLLECTOR
C PERB- PERCENTAGE OF BEAM SOLAR RADIATION
C PERD- PERCENTAGE OF DIFFUSE SOLAR RADIATION
C PLEAK- PERIPHERY LEAKAGE RATE
C PHI- COLLECTOR LOCATION LATITUDE
C REFLEC- REFLECTANCE OF GROUND IN AREA OF COLLECTOR
C RHOF- AVERAGE DENSITY OF AIR FLOW STREAM
C RLEAK- RECIRCULATION LEAKAGE RATE
C RPS- ROTATING SPEED OF DESICCANT WHEEL IN REV. PER. SEC.
C SL- SLOPE OF COLLECTOR MINUS LATITUDE
C T- AMBIENT TEMPERATURE
C TIDR- MINIMUM REGENERATION TEMPERATURE (PROGRAM OPTION #1)
C TMIN- MINIMUM REGENERATION TEMPERATURE OF DESICCANT (OPTION #2)
C U- EFFECTIVE CONDUCTANCE BETWEEN DESICCANT AND AIR STREAM
C UB- COLLECTOR BOTTOM LOSS COEFFICIENT
C V- AMBIENT WIND SPEED
C VOID- VOID FRACTION OF DESICCANT MATERIAL
C XE- SENSIBLE HEAT EXCHANGER EFFECTIVENESS
C W- DISTANCE BETWEEN TUBE CENTER LINES IN WATER COLLECTOR
C WAM- AMBIENT HUMIDITY RATIO
C WIDTH- WIDTH OF FLOW CHANNEL FOR AIR TYPE COLLECTORS
C WMASS- WATER FLOW RATE FOR WATER SOLAR COLLECTORS (CTYPE 1)
C WXE- EFFECTIVENESS FOR WATER TO AIR HEAT EXCHANGER (CTYPE 1)
C

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1 REAL*4 LEN, ID, NCOVS, KFIN, INFIST
2 INTEGER*4 CTYPE
3 CHARACTER*40 MDY
4

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COMMON CP, AC, LEN, TAU, ALPA, NCOVS, EPP, EPG, D, WIDTH, AMU, AK, DEL, W, CB, PI
1, ID, HFILM, KFIN, UB, QDUCT, CTYPE, IFLAG, AMASS, WXE, PHI, ALONG, E, SL, DTDC
2, DTCD, UD, AFLOW, SHR, SQJ, SEFF, SQTL, SQAUX, SQSURP, R, GAUX, CIL, QSURP, AL,
3FL, VOID, RPS, RHUF, U, DD, FRIC, ALFA, SIG, GAM, BLEAK, PLEAK, RLEAK, CTAU, CAL
4PA, AFF, ALF, GAMA, KCUNT, QDEHU, PERB, PERL, REFLEC, CMEGA, AF, AV, HMU, ENU,

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5      5WMASS
      DIMENSION T(1,24),WAM(1,24),TIDP(1,24),WIDP(1,24),TQDP(1,24),WODP(
11,24),TCXP(1,24),TOXR(1,24),QDUCT(1,24),      TIDR(1,24),TORD(1,2
24),WORD(1,24),IDATE(1),V(1,24),H(1,24),HR(1,24),EFF(1,24),FP(1,24)
3,SHR(1),      SQU(1),SEFF(1),SQT(1),SQAUX(1),SQSURP(1),R(1,24
4),UT(1,24),UL(1,24),QU(1,24),CAUX(1,24),QSURP(1,24),QTL(1,24),ALFA
5(4,2),SIG(2,2),GAM(2,2),ALF(4,2),GAMA(4,2),TIC(1,24),TUC(1,24),QD
6EHU(1,24),IHR(1,24),IFD(1,24),TRD(1,24),OMEGA(1,24),FR(1,24),IODR(
71,24),WODR(1,24),QHEX(1,24),WIDR(1,24),WEXR(1,24),WOXP(1,24),TIH(1
8,24)

6      C SUBPROGRAM MAIN I/O
      KOUNT=1

      C COLLECTOR TYPES;      #0 MANUFACTURER'S COLLECTOR PERFORMANCE CURVES INPUT
      C      #1 WATER COLLECTOR WITH BONDED TUBES (NOTE WATER/AIR
      C      HEAT EXCHANGER MUST BE SPECIFIED)
      C      #2 AIR COLLECTOR WITH METAL/GLASS CHANNEL.
      C      #3 AIR COLLECTOR WITH METAL/METAL CHANNEL

      C PROGRAM OPTIONS;      #1 VARIABLE REGENERATION TEMPERATURE SPECIFIED
      C      #2 CONSTANT REGENERATION TEMPERATURE SPECIFIED

7      PI=3.141
8      CP=1005.

      C TEMPERATURE INPUT IN DEG C,      WIND VELOCITY IN M/S,      SOLAR RADIATION IN
      C KJ/HR M**2

      C READ COLLECTOR TYPE, PROGRAM OPTION, NUMBER OF DAYS AND HOURS, DATE
9      READ (5,400) CTYPE,IFLAG,NH,ND,MDY
10     WRITE (6,400) CTYPE,IFLAG,NH,ND,MDY
11     400 FORMAT(4I5,A40)

      C READ COLLECTOR DATA
12     READ (5,401) AMASS,AC,SL,PHI,ALONG
13     WRITE (6,401) AMASS,AC,SL,PHI,ALONG
14     401 FORMAT(8F10.3)
15     IF(CTYPE.EQ.0) GO TO 410

      C INPUT INFORMATION FOR TYPES 1,2,AND 3 COLLECTORS
16     READ (5,401) LEN,CIAU,CALPHA,NCOVS,EPP,EPG,UB
17     WRITE (6,401) LEN,CIAU,CALPHA,NCOVS,EPP,EPG,UB

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18 C DATA FOR WATER TYPE COLLECTORS (TYPE #1)
19 IF (CTYPE.EQ.1) READ(5,401) DEL,W,OD,CB, ID,HFIMI,KFIN,WXE
19 IF (CTYPE.EQ.1) WRITE(6,401) DEL,W,OL,CB, ID,HFIMI,KFIN,WXE
20 C DATA FOR AIR TYPE COLLECTORS (TYPE #2 OR TYPE #3)
21 IF (CTYPE.NE.1) READ(5,401) D,WIDTH,AMU,AK
21 IF (CTYPE.NE.1) WRITE(6,401) D,WIDTH,AMU,AK
22 410 READ(5,401) DTDC,DTCD,PERB,PERD,REFLEC
23 WRITE(6,401) DTCC,DTCD,PERB,PERD,REFLEC
24 C READ DEHUMIDIFIER DATA
24 REAC(5,401) AV,FL,AF,VOID,RPS,RHOF,IMIN
25 WRITE(6,401) AV,FL,AF,VGID,RPS,RHOF,IMIN
26 REAC(5,401) U,DD,FRIC,HMU,KLEAK,PLEAK,BLEAK,XE
27 WRITE(6,401) U,DD,FRIC,HMU,RLEAK,PLEAK,BLEAK,XE
28 C READ DAILY AMBIENT CONDITIONS: RADIATION,TEMPERATURE,WIND SPEED,HUMIDITY RATIO
29 C 0
28 DO 20 I=1,ND
29 READ (5,403) ICATE(I)
30 403 FORMAT (I5)
31 READ (5,405) (IHR(I,J),J=1,NH)
32 405 FORMAT(16I5)
33 READ (5,402)(H(I,J),J=1,NH)
34 WRITE (6,402)(H(I,J),J=1,NH)
35 READ (5,402)(T(I,J),J=1,NH)
36 WRITE (6,402)(T(I,J),J=1,NH)
37 READ (5,402)(V(I,J),J=1,NH)
38 WRITE (6,402)(V(I,J),J=1,NH)
39 READ (5,402)(WAM(I,J),J=1,NH)
40 WRITE (6,402)(WAM(I,J),J=1,NH)
41 C IF PROGRAM OPTION #1 SELECTED INPUT VARIABLE REGENERATION TEMPERATURE
41 IF (IFLAG.EQ.1) READ(5,402) (TIDR(I,J),J=1,NH)
42 IF (IFLAG.EQ.1) WRITE(6,402) (TIDR(I,J),J=1,NH)
43 402 FORMAT(7F10.4)
44 DO 10 J=1,NH
45 C IF OPTION #2 SELECTED SET TIDR EQUAL TO IMIN
45 IF (IFLAG.EQ.2) TIDR(1,J)=IMIN

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46 C ASSUME A 3 DEGREE TEMPERATURE RISE THROUGH FAN.
   TIDP(I,J)=T(I,J)+3.0
47 C **NOTE** HUMID SUBPROGRAM IS ON DEG C BASIS
   WIDP(I,J)=WAM(I,J)
48   WIDR(I,J)=WAM(I,J)
49   CALL HUMID (TIDP(I,J),TIDR(I,J),WAM(I,J),TODP(I,J),TODR(I,J),WODR(
50     11,J),WODP(I,J),WIDR(I,J),AMASS,XE)
   CALL HEATX (TODP(I,J),TIDP(I,J),TOXP(I,J),TOXR(I,J),AMASS,QHEX(I,J
51     1),WODP(I,J),WAM(I,J),WOXP(I,J),WOXR(I,J),XE)
   CALL COLPER (J,I,T,V,H,TOXR,TIH,HR,EFF,QU,UL(I,J),UT(I,J),FP(I,J),
   IFR(I,J),TIC,TCC,IFR(I,J),IDATE,TIDR(I,J))
   C CALCULATE ENTHALPY AND HEAT TRANSFERED IN DEHUMIDIFIER.
   HAIDP=CP*TI(I,J)+1061.*1055.7*2.205*WAM(I,J)
   HAODP=CP*TODP(I,J)+1061.*1055.7*2.205*WODP(I,J)
   QDEHU(I,J)=AMASS*(HAODP-HAIDP)
   C INCREMENT KOUNT SO APPARATUS PARAMETERS WILL ONLY BE PRINTED ONCE
   KOUNT=KCOUNT+1
55   10 CONTINUE
56   20 CONTINUE
57   DO 300 I=1,NC
58     PRINT OUT HOURLY TEMP-HUMIDITY MAP FOR PROCESS SIDE
59     WRITE(6,100)
60     WRITE(6,1001) MDY
61     WRITE(6,1193) XE
62     WRITE(6,119) BLEAK
63     WRITE(6,1191) PLEAK
64     WRITE(6,1192) RLEAK
65     119 FORMAT(/T23,'BYPASS LEAKAGE RATE= ',F6.3)
66     1191 FORMAT(/T23,'PERIPHERY LEAKAGE RATE= ',F6.3)
67     1192 FORMAT(/T23,'RECIRCULATION LEAKAGE RATE= ',F6.3)
68     1193 FORMAT(/T00,'HEAT EXCHANGER EFFECTIVNESS ',F6.3)
69     IF(IFLAG.EQ.1) GO TO 30
70     IF(IFLAG.EQ.2) GO TO 40
71     GO TO 50
72     30 WRITE(6,1003)

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73      GO TO 50
74      WRITE(6,1002)
75      WRITE(6,101)
76      WRITE(6,102)
77      WRITE(6,103)
78      WRITE(6,104)
79      WRITE(6,105)
80      WRITE(6,1051)
81      WRITE(6,106)
82      DO 150 J=1,NH
83      WRITE(6,116) IHR(I,J),T(I,J),WAM(I,J),TIDP(I,J),WIDP(I,J),TODP(I,J)
      1),WODP(I,J),TODP(I,J),WODP(I,J),TCXP(I,J),WCXP(I,J)
84      150 CONTINUE
      C PRINT OUT HOURLY TEMP-HUMIDITY MAP FOR REGENERATION SIDE
85      WRITE(6,100)
86      WRITE(6,1001) MDY
87      WRITE(6,1193) XE
88      WRITE(6,119) BLEAK
89      WRITE(6,1191) PLEAK
90      WRITE(6,1192) FLEAK
91      IF(IFLAG.EQ.1) GO TO 31
92      IF(IFLAG.EQ.2) GO TO 41
93      GO TO 51
94      31 WRITE(6,1003)
95      GO TO 51
96      41 WRITE(6,1002)
97      51 WRITE(6,101)
98      WRITE(6,102)
99      WRITE(6,107)
100     WRITE(6,108)
101     IF(CTYPE.NE.1) WRITE(6,1081)
102     IF(CTYPE.EQ.1) WRITE(6,1083)
103     WRITE(6,1082)
104     WRITE(6,109)
105     WRITE(6,110)

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```

106 DO 151 J=1,NH
107   WRITE(6,117) IHR(I,J),TIDP(I,J),WAM(I,J),TOXR(I,J),WOXR(I,J),IIC(I
      1,J),WOXR(I,J),ICC(I,J),WOXR(I,J),TIH(I,J),TICR(I,J),WIDR(I,J),TODR
      2(I,J),WQDR(I,J)
108   151 CONTINUE
      C PRINT OUT HOURLY FEAT FLOW MAP
109   WRITE(6,112)
110   WRITE(6,1001) MDY
111   WRITE(6,113)
112   WRITE(6,1193) XE
113   WRITE(6,119) BLEAK
114   WRITE(6,1191) PLEAK
115   WRITE(6,1192) RLEAK
116   WRITE(6,114)
117   WRITE(6,1141)
118   WRITE(6,115)
119   WRITE(6,1151)
120   WRITE(6,1152)
121   DO 301 J=1,NH
122     TQDEHU=3.6*QDEHU(I,J)
123     TQHEX=3.6*QHEX(I,J)
124     TQDUCT=3.6*QDUCT(I,J)
125     TQU=QU(I,J)*3.6
126     TQ AUX=Q AUX(I,J)*3.6
127     TQSURP=3.6*QSURP(I,J)
128     WRITE(6,118) IHR(I,J),QDEHU(I,J),TQDEHU,QHEX(I,J),TQHEX,QDUCT(I,J)
      1,TQDUCT,QU(I,J),TQU,Q AUX(I,J),TQ AUX,CSURP(I,J),TQSURP
129   301 CONTINUE
130   300 CONTINUE
131   100 FORMAT('1',/,T7,'***** HOURLY TEMPERATURE AND HUMIDITY MAP *****')
132   1001 FORMAT('T90,A40')
133   1002 FORMAT('/T23,'***** CONSTANT REGENERATION TEMPERATURES *****')
134   1003 FORMAT('/T23,'***** VARIABLE REGENERATION TEMPERATURES *****')
135   101 FORMAT('/T19,'TEMPERATURE IN DEG C')
136   102 FORMAT('/T19,'HUMIDITY RATIO IN KGW/KGA')

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137 103 FORMAT(/// T10,'***** PROCESS SIDE *****')
138 104 FORMAT( T10,'*****')
139 105 FORMAT(//,T43,'DEHUMIDIFIER',T73,'SENSIBLE HEAT EXCHANGER')
140 1051 FORMAT(/T6,'FOUR',T19,'AMBIENT',T39,'INLET',T54,'OUTLET',T73,'INLE
      11',T92,'OUTLET')
141 106 FORMAT(/T16,'TEMP      HUMIDITY      TEMP      HUMIDITY      TEMP      HUMIDITY
      1  TEMP      HUMIDITY      TEMP      HUMIDITY')
142 107 FORMAT(//T10,'***** REGENERATION SIDE *****')
143 108 FORMAT( T10,'*****')
144 1081 FORMAT( /T13,'SENSIBLE HEAT EXCHANGER',T57,'SCLAR COLLECTOR',T83,
      1'AUX. HEATER',T105,'DEHUMIDIFIER')
145 1082 FORMAT( T11,'-----',T50,'-----')
146 1083 FORMAT( /T13,'SENSIBLE HEAT EXCHANGER',T57,'AIR/WATER HEATX',T83,
      1'AUX. HEATER',T105,'DEHUMIDIFIER')
147 109 FORMAT( T15,'INLET',T33,'OUTLET',T52,'INLET',T71,'OUTLET',T86,'INL
      1ET',T100,'INLET',T119,'OUTLET')
148 110 FORMAT(/T3,'HOURLY',T11,'TEMP      HUMIDITY      INLET      HUMIDITY',T48,'TE
      1MP      HUMIDITY      TEMP      HUMIDITY      TEMP      HUMIDITY
      1TEMP      HUMIDITY')
149 112 FORMAT(1,'//***** TOTAL HOURLY HEAT BALANCE MAP *****')
150 113 FORMAT(T13,'RATES IN WATTS',T113,'TOTALS IN KJ')
151 114 FORMAT (/T17,'HEAT TRANSFERED',T35,'HEAT TRANSFERED',T60,'HEAT LOS
      1T',T80,'SCLAR HEAT',T99,'AUXILIARY',T115,'SURPLUS SCLAR')
152 1141 FORMAT( T23,'IN',T44,'IN',T64,'IN',T81,'GAIN IN',T101,'HEAT',T115,
      1'HEAT AVAILABLE')
153 115 FORMAT( T5,'FOUR',T18,'DEHUMIDIFIER',T39,'HEAT EXCHANGER',T61,'DUC
      1TING',T80,'COLLECTOR',T99,'SUPPLIED',T116,'FOR STORAGE')
154 1151 FORMAT( T18,'RATE',T25,'TOTAL',T40,'RATE',T48,'TOTAL',T59,'RATE',T
      165,'TOTAL',T77,'RATE',T86,'TOTAL',T97,'RATE',T104,'TOTAL',T115,'RA
      2TE',T123,'TOTAL')
155 1152 FORMAT( T15,'-',T113('---'))
156 116 FORMAT(/T6,T3,T15,F5.1,T24,F6.4,T32,F5.1,T40,F6.4,T45,F5.1,T59,F6.
      14,T69,F5.1,T78,F6.4,T89,F5.1,T97,F6.4)
157 117 FORMAT(/T3,T2,T10,F5.1,T18,F6.4,T29,F5.1,T37,F6.4,T48,F5.1,T56,F6.

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14, I67, F5.1, I74, F6.4, I85, F5.1, I96, F5.1, I104, F6.4, I115, F5.1, I123, F6.
241
118 FORMAT( /I5, I3, I14, F8.1, I23, F7.1, I36, F8.1, I45, F8.1, I55, F8.1, I64, F7
1.1, I73, F8.1, I83, F8.1, I93, F8.1, I102, F8.1, I111, F8.1, I120, F8.1)
C PRINT OUT HOURLY AMBIENT CONDITIONS
DO 1050 M=1, ND
WRITE(6, 1009) MCY
1009 FORMAT('1', //I23, 'HOURLY SUMMARY OF AMBIENT CONDITIONS FOR ', A40)
WRITE(6, 1011)
WRITE(6, 1012)
WRITE(6, 1013)
1011 FORMAT(//I23, 'AMBIENT', I41, 'WIND')
1012 FORMAT(I21, 'HOUR', I31, 'TEMP.', I41, 'SPEED', I50, 'RADIATION', I62, 'OME
IGA', I72, 'FR', I82, 'R')
1013 FORMAT(I30, '(DEG C)', I41, '(M/S)', I51, ' (W/M**2)', I63, '(DEG)', I70, '
1 (W/M**2)')
DO 1060 K=1, NH
WRITE(6, 1014) IHR(M, K), I(M, K), V(M, K), J(M, K), CMGA(M, K), HR(M, K), R(M
1, K)
1014 FORMAT(I23, I3, I32, F5.2, I41, F5.2, I52, F6.1, I62, F5.1, I71, F6.1, I81, F6
1.3)
1060 CONTINUE
1050 CONTINUE
C PRINT OUT HOURLY COLLECTOR PERFORMANCE
DO 251 I=1, ND
WRITE(6, 200)
IF(CTYPE.EQ.0) WRITE(6, 2001)
IF(CTYPE.EQ.1) WRITE(6, 201)
IF(CTYPE.EQ.2) WRITE(6, 202)
IF(CTYPE.EQ.3) WRITE(6, 203)
WRITE(6, 204)
IF(CTYPE.NE.0) WRITE(6, 205)
IF(CTYPE.EQ.0) WRITE(6, 2051)
IF(CTYPE.NE.0) WRITE(6, 206)
IF(CTYPE.EQ.0) WRITE(6, 2061)

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184      WRITE(6,2C7)
185      200 FORMAT('1'//T5,'HOURLY PERFORMANCE OF COLLECTOR')
186      2001 FORMAT(/T5,'COLLECTOR TYPE #0- MANUFACTURERS COLLECTOR PERFORMANCE
187      1 CURVES INPUT')
187      201 FORMAT(/T5,'COLLECTOR TYPE #1 -WATER COOLED COLLECTOR')
188      202 FORMAT(/T5,'COLLECTOR TYPE #2 -AIR COOLED COLLECTOR WITH GLASS/MET
189      AL CHANNEL')
189      203 FORMAT(/T5,'COLLECTOR TYPE #3 -AIR COOLED COLLECTOR WITH METAL/MET
190      AL CHANNEL')
190      204 FORMAT(/T5,'INLET TEMP.      OUTLET TEMP.  USEFUL RATE  ',T68,'TU
191      IP LOSS',T83,'TOTAL LOSS',T98,'COLLECTOR',T113,'COLLECTOR')
191      205 FORMAT('T18','TO',T30,'FROM',T44,'OF',T67,'COEFFICIENT',T83,'COEFFI
192      CIENT',T98,'EFFICIENCY',T113,'HEAT REMOVAL')
192      2051 FORMAT('T18','TO',T30,'FROM',T44,'OF',T67,'COEFFICIENT',T83,'COEFFI
193      CIENT',T98,'EFFICIENCY',T113,'EFFICIENCY')
193      206 FORMAT('T5','HOUR',T15,'COLLECTOR',T28,'COLLECTOR',T42,'HEAT GAIN',
194      T55,'EFFICIENCY',T71,'-UT',T87,'UL',T98,'FACTOR-FP',T113,'FACTOR-F
195      2R')
194      2061 FORMAT('T5','HOUR',T15,'COLLECTOR',T28,'COLLECTOR',T42,'HEAT GAIN',
195      T55,'EFFICIENCY',T71,'-UT',T87,'UL',T98,'FACTOR-FRUL',T113,'FACTOR
196      2-FRIA')
195      207 FORMAT(/T17,'(LEG C)',T30,'(DEG C)',T44,'(W)',T65,'(W/M**2-C)',T85
196      1,'(W/M**2-C)')
196      DO 250 J=1,NH
197      WRITE(6,208) IHR(I,J),TIC(I,J),TOC(I,J),QU(I,J),EFF(I,J),UT(I,J),U
198      IL(I,J),FP(I,J),FR(I,J)
198      208 FORMAT(/T5,I3,T16,F5.1,I30,F5.1,I43,F6.1,I57,F5.3,I70,F6.3,I85,F6.
199      13,I100,F5.3,I116,F5.3)
199      250 CONTINUE
200      251 CONTINUE
201      STOP
202      END

203      SUBROUTINE HUMIC (T1,T2,W1,TOUT,TEX,WEX,WOUT,W2,FLCR,XE)
204      COMMON CP,AC,LEN,TAU,ALPHA,NCGVS,EPP,EPG,D,WIDTH,AMU,AK,DEL,W,CB,PI

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1, ID, HFILMI, KFIN, UB, QDUCT, CTYPE, IFLAG, AMASS, WXE, PHI, ALCNG, E, SL, DTDC
2, DTCD, JC, AFLOW, SHR, SQU, SEFF, SQTL, SQAUX, QSURP, R, QAUX, QIL, QSURP, AL,
3FL, VOID, RPS, RHOF, U, DD, FRIC, ALFA, SIG, GAM, BLEAK, PLEAK, RLEAK, CIAU, CAL
4PA, AFF, ALF, GAMA, KCUNT, QDEHU, PERB, PERD, REFLEC, OMEGA, AF, AV, HMU, ENU,
5WMASS
205 DIMENSION SHR(1), SQU(1), SEFF(1), SQTL(1), SCAUX(1), QSURP(1), QU
1(1,24), QAUX(1,24), QSURP(1,24), QTL(1,24), ALFA(4,2), SIG(2,2), GAM(2,2
2), ALF(4,2), GAMA(4,2), QDEHU(1,24), QDUCT(1,24), R(1,24), CMEGA(1,24)
ITMUM=0
DO 198 I=1,2
DO 199 J=1,2
SIG(I,J)=0.0
199 CONTINUE
198 CONTINUE
TAU=1./(2.*RPS)
VEL=FLOR/(RHOF*AF*VOID)
STDF=.06
INC=20
NN=1
C CALCULATE PROPERTIES AT STATE 1, INLET CONDITIONS ON PROCESS SIDE.
N=1
DO 201 I=1,2
INCI=1
CALL ALFAV(T1,W1,I,N,INCI,ALFA(N,I))
201 CONTINUE
C DETERMINE PROPERTIES AT STATE 2, INLET CONDITIONS ON REGENERATION SIDE
N=2
DO 202 I=1,2
INCI=1
CALL ALFAV(T1,W1,I,N,INCI,ALFA(N,I))
202 CONTINUE
C ESTIMATE POINT 3, OUTLET CONDITIONS RECLNERATION SIDE
C F FUNCTION=T+ALPH*W -AT POINT3 F31=F32
W3=(ALFA(1,2)*W1-ALFA(2,1)*W2+T1-T2)/(ALFA(1,2)-ALFA(2,1))
T3=-ALFA(2,1)*(W3-W2)+T2
227
228

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C ESTIMATE PCINT 4, GUTLET PROCESS SIDE
C F FUNCTION=T+ALPH*W -AT POINT4 F41=F42
    W4=(ALFA(1,1)*W1-ALFA(2,2)*W2+T1-T2)/(ALFA(1,1)-ALFA(2,2))
    T4=-ALFA(2,2)*(W4-W2)+T2
C DETERMINE TRUE POINTS 3 AND4
C CALCULATE VALUE OF INCRIMENTAL STEPS ALCNG CONSTANT F LINE
    DELT23=(T3-T2)/FLCAT(INC)
    DELT13=(T3-T1)/FLCAT(INC)
    T31=T1
    T32=T2
    W31=W1
    W32=W2
    T51=T1
    W51=W1
    T52=T2
    W52=W2
    INC=3*INC
    NNN=3
    NN=2
    INDIC=0
    DO 203 LL=1,INC
    IF(INDIC.EQ.2) GO TO 210
    N=1
    I=2
    CALL EGV(T51,W51,I,LL,N,NNN,DELT13,NN,T31,W31)
210 N=2
    I=1
    IF(INDIC.EQ.1) GO TO 211
    CALL EGV(T52,W52,I,LL,N,NNN,DELT23,NN,T32,W32)
211 INDIC=1
C CHECK TO ASSURE EQUAL PROGRESSION FROM BOTH INITIAL POINTS TO POINT3
    IF(W31.GT.W32) INDIC=2
C IF T32.GT.T31 PCINT 3 HAS BEEN REACHED, DETERMINE W3,T3
    IF(T32.LT.T31) GO TO 205
203 CONTINUE

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258 C NOTE** POINT 3 VALUES DO NOT TAKE INTO CONSIDERATION ROTATING BED
259 205 W3=(ALFA(3,2)*W32-ALFA(3,1)*W31+T32-T31)/ALFA(3,2)-ALFA(3,1))
260 T3=-ALFA(3,1)*(W3-W32)+T32
261 GAM(2,1)=SIG(2,1)/ABS(W31-W1)
262 GAM(2,2)=SIG(2,2)/ABS(W32-W2)
263 C DETERMINE POINT 4 BY SIMILAR METHOD STARTING FROM POINTS 1 AND 2
264 DELT14=-DELT23
265 DELT24=-DELT13
266 I41=T1
267 I42=T2
268 W41=W1
269 W42=W2
270 TS1=T1
271 TS2=T2
272 WS1=W1
273 WS2=W2
274 ANN=4
275 NN=1
276 INDIC=0
277 DO 206 LL=1,INC
278 I=1
279 N=1
280 IF(INDIC.EQ.2) GO TO 212
281 CALL EGV(TS1,WS1,I,LL,NN,DELT14,NN,I41,W41)
282 212 I=2
283 N=2
284 IF(INDIC.EQ.1) GO TO 213
285 CALL EGV(TS1,WS1,I,LL,NN,DELT24,NN,I42,W42)
286 213 INDIC=1
287 C CHECK TO ASSURE EQUAL PROGRESSION FROM BOTH INITIAL POINTS TO POINT4
288 IF(W41.LT.W42) INDIC=2
289 C IF I42.GT.I41 POINT 4 HAS BEEN REACHED, DETERMINE W4,I4
290 IF(I42.LT.I41) GO TO 207
291 206 CONTINUE
292 C NOTE** POINT 4 VALUES DO NOT TAKE INTO CONSIDERATION ROTATING BED

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288 207 W4=(ALFA(4,2)*W42-ALFA(4,1)*W41+T42-T41)/(ALFA(4,2)-ALFA(4,1))
289 T4=-ALFA(4,2)*(W4-W42)+T42
290 GAM(1,1)=SIG(1,1)/ABS(W42-W2)
291 GAM(1,2)=SIG(1,2)/ABS(W41-W1)
C NOW DETERMINE TRUE STATES OF POINT 3 AND POINT 4, BY CONSIDERING THE ROTAT-
C TIONAL EFFECTS OF WHEEL USING ANALOGY CF ROTATING HEAT EXCHANGER OUTLINED BY
C BANKS AND CLOSE
FLE=37.244*FL
AU= 196.85*(FLOR/RHCF)/AF
DELPE=(.0053*AU+5.9E-05*AU**2.0)*FLE
DELP=DELPE*248.84
C DETERMINE EFFECTIVENESS OF ROTATING DESICCANT WHEEL
E11=((STDF*DELP*2.*(1.+HMU* GAM (1,1)))/(RHOF*VEL**2.*HMU*(GAM(1,1)
1+GAM(2,1)))
E12=(1.+HMU*GAM(2,1))/(1.+HMU*GAM(1,1))
E13=(1.+HMU*GAM(1,1))*FL/(TAU*VEL)
E21=((STDF*DELP*2.*(1.+HMU* GAM (1,2)))/(RHOF*VEL**2.*HMU*(GAM(1,2)
1+GAM(2,2)))
E22=(1.+HMU*GAM(2,2))/(1.+HMU*GAM(1,2))
E23=(1.+HMU*GAM(1,2))*FL/(TAU*VEL)
ETACF2=(1.-EXP(-E21*(1-E22)))/(1.-E22*EXP(-E21*(1.-E22)))
ETA11=E13
ETA12=E12*ETA11
ETA21=ETACF2*(1.-1./(9.*E23**1.93))
ETA22=ETA21*E22
C DETERMINE TRUE PROCESS OUTLET CCNDITICAS
TOUT=Y1+ETA11*(T2-T41)+ETA21*(T41-T1)
WOUT=W1+ETA11*(W2-W41)+ETA21*(W41-W1)
IF (WOUT.LE.0.) WOUT=0.
C DETERMINE TRUE OUTLET REGENERATION SIDE TEMPERATURE AND HUMIDITY
TEX=T2+ETA12*(T1-T31)+ETA22*(T31-T2)
WEX=W2+ETA12*(W1-W31)+ETA22*(W31-W2)
C CORRECT FOR LEAKAGE IN DEHUMIDIFIER
CALL LEAK (W1,W2,T1,T2,WOUT,WEX,TOUT,TEX,WOUT,WEX,TOUT,TEX,XE)
TIME=2.*TAU

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314      IF (KOUNT.NE.1) RETURN
315      C PRINT OUT DEHUMIDIFIER PARAMETERS
316      WRITE(6,299)
317      WRITE(6,300) AV
318      WRITE(6,301) FL
319      WRITE(6,302) AF
320      WRITE(6,303) VOID
321      WRITE(6,304) RPS
322      WRITE(6,305) RHOF
323      WRITE(6,307) DD
324      WRITE(6,308) FLOR
325      WRITE(6,309) FMU
326      WRITE(6,310)
327      WRITE(6,311) TIME
328      WRITE(6,312) VEL
329      WRITE(6,313) DCLP
330      299 FORMAT('1',I5,'***** DEHUMIDIFICATION PARAMETERS *****')
331      300 FORMAT(/I5,'SURFACE AREA OF PACKING PER UNIT VOL. ',F8.2,' M**2/M*
332      1*3')
333      301 FORMAT(/I5,'LENGTH IN FLOW DIRECTION ',F8.5,' M')
334      302 FORMAT(/I5,'TOTAL FRONTAL AREA EXPOSED TO FLOW ',F8.5,' M**2')
335      303 FORMAT(/I5,'VOID FRACTION ',F5.3)
336      304 FORMAT(/I5,'ROTATING SPEED OF MATRIX ',F8.6,' REV/S')
337      305 FORMAT(/I5,'DENSITY OF AIRSTREAM ',F6.3,' KG/M**3')
338      307 FORMAT(/I5,'DESICCANT DIMENSION ',F8.6,' M')
339      308 FORMAT(/I5,'AIR MASS FLOW RATE ',F8.5,' KG/S')
340      309 FORMAT(/I5,'MATRIX MASS/ AIR MASS IN MATRIX ',F6.1)
341      310 FORMAT(/I5,'***** CALCULATED DATA *****')
342      311 FORMAT(/I5,'TIME PER REVOLUTION ',F8.2,' S')
343      312 FORMAT(/I5,'AIR VELOCITY ',F6.3,' M/S')
344      313 FORMAT(/I5,'EST. PRESSURE DROP THRU MATRIX ',F6.2,' N/M**2 ')
345      RETURN
346      END
347
348      SUBROUTINE EGV(TC,WC,I,LL,N,NN,DELT,NN,ICN,WCN)

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346 C THIS SUBPROGRAM DETERMINES INCREMENTAL TEMPERATURE AND HUMIDITY RATIO VALUES
347 C FOR HUMID SUBPROGRAM AS IT STEPS ALONG PATHS OF CONSTANT F FROM INLET TO
348 C OUTLET CONDIDIGNS
      COMMON CP,AC,LEN,TAU,ALPA,NCOVS,EPP,EPG,D,WIDTH,AMU,AK,DEL,W,CB,PI
      1,ID,HFILMI,KFIN,UB,QDUCL,CTYPE,IFLAG,AMASS,WXE,PHI,ALCNG,E,SL,DIDC
      2,DICD,OD,AFLOW,SHR,SQU,SEFF,SQTL,SQAUX,SQSURP,R,CAUX,CIL,QSURP,AL,
      3FL,VGID,RPS,RHOF,U,DD,FRIC,ALFA,SIG,CAM,BLEAK,PLEAK,RLEAK,CTAU,CAL
      4PA,AFF,ALF,GAMA,KCUNT,QDEHU,PERB,PERC,REFLEC,CMEGA,AF,AV,HMU,ENU,
      5WMASS
347 DIMENSION SHR(1),SQU(1),SEFF(1),SQTL(1),SQAUX(1),SQSURP(1), QU
      1(1,24),QAUX(1,24),QSURP(1,24),QTL(1,24),ALFA(4,2),SIG(2,2),GAM(2,2
      2),ALF(4,2),GAMA(4,2),QDEHU(1,24),QDUCT(1,24),R(1,24),OMEGA(1,24)
      ALFC=ALFA(N,I)
      IF(11.EQ.1) GO TO 1005
      ALFC=ALFA(NNN,I)
      1005 TC=TCN
      WC=WCN
      TCN=TC+DELT
      DO 1002 MM=1,10
      IF(MM.EQ.1) ALFCB=ALFC
      WCN=WC-(TCN-TC)/ALFCB
      IF(MM.EQ.1) GO TO 1001
      DELTW=WCN-WC0
      IF(ABS(DELTW)/WCN.LE..050) GO TO 1004
      1001 INCI=1
      CALL ALFAV(TCN,WCN,I,NNN,INCI,ALFA(NNN,I))
      ALFCN=ALFA(NNN,I)
      ALFCB=(ALFCN+ALFC)/2.
      WCO=WCN
      1002 CONTINUE
      1004 ALFA(NNN,I)=ALFCB
      TMEAN=(TC+TCN)/2.
      WMEAN=(WC+WCN)/2.
      IF(1.EQ.1) I1=2
      IF(1.EQ.2) I1=1

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371      INCI=2
372      CALL ALFV(TMEAN,WMEAN,II,NN,INCI,GAM(NN,II))
373      SIG(NN,II)=GAM(NN,II)*ABS(WCN-WC)+SIG(NN,II)
374      RETURN
375      END

376      SUBROUTINE ALFV(T,W,I,NN,INDI,DUMMY)
377      C THIS PROGRAM SUPPLIES VALUES OF ALPHA FOR SUBPROGRAM EGV, USING BANKS-CLOSE
378      C ANALOGY FROM "CHEM. ENGG. SCI." VOL. 27,1972
379      DIMENSION ALF(4,2),GAMA(4,2)
380      IF=1.8*T+32.
381      HFG=1095.4-.5877*IF
382      TK=273.16+T
383      C DETERMINE AIR SATURATION PRESSURE USING 77 ASHRAE FUNDAMENTALS FORMULA
384      BETA=647.27-TK
385      A1=3.24378
386      A2=5.86826E-03
387      A3=1.170238E-08
388      A4=2.18785E-04
389      XPS=-(BETA/TK)*((A1+A2*BETA+A3*BETA**3)/(1.+A4*BETA))
390      PWS=218.167*10.0**((XPS))
391      TKN=TK+.01
392      BETN=BETA-.01
393      XPWSN=-(BETN/TKN)*((A1+A2*BETN+A3*BETN**3)/(1.+A4*BETN))
394      PWSN=218.167*10.0**((XPWSN))
395      DPWS=(PWSN-PWS)/(TKN-TK)*1.8)
396      ITNUM=1
397      IF (W.GT..0) GO TO 102
398      ALF (NN,1)=5.0E10
399      ALF (NN,2)=-1.0E10
400      GAMA(NN,1)=.5
401      GAMA(NN,2)=1.0
402      GO TO 103
403      115 KH=1.
404      PW=PWS

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402      W=.62198 * PWS/(1.-PWS)
403      GO TO 120
404
405      102 PW=W/(.622+W)
406      RH=PW/PWS
407      IF(RH.GE.1.) GO TO 115
408      120 X=PW/(2.009*PWS)
409      DO 105 K=1,20
410      C DETERMINE VALUE OF HSISOTHERMAL DIFFERENTIAL HEAT OF SORPTION)/HV(LATENT
411      C HEAT OF VAPORISATION OF WATER
412      IF(X.LE..01) RHF=22.79*X+1.0
413      IF(X.GT..01).AND.(X.LI..04)) RHH=8533.*X**3-542.*X**2+11.79*X+1.1
414      1557
415      IF(X.GT..04).AND.(X.LE..1)) RHH=45.4*X**2-8.97*X+1.5922
416      IF(X.GT..1).AND.(X.LE..35)) RHH=-9.1553*X**3+7.048*X**2-1.8966*X+
417      11.2789
418      IF(X.GT..35) RHH=-.58315*X+1.2894
419      PW=((29.91*2.009*X*PWS)**RHH)/29.91
420      WN=.622*PW/(1.-PW)
421      IF(ITNUM.EQ.1) GO TO 114
422      IF(ABS((WN-W)/W).LE..050)GO TO 107
423      C DRFH IS DERIVATIVE OF HS/HV WITH RESPECT TO SILICA GEL WATER CCNTENT
424      114 IF(X.LE..01) CRFH= 22.79
425      IF(X.GT..01).AND.(X.LE..04)) DRHH=25600.*X**2-1084.*X+11.79
426      IF(X.GT..04).AND.(X.LE..1)) DRHH=50.8*X-8.57
427      IF(X.GT..1).AND.(X.LE..35)) DRHH=-27.466*X**2+14.096*X-1.8966
428      IF(X.GT..35) DRHH=-.58135
429      ITNUM=ITNUM+1
430      ZETA=(ALOG(29.91*2.009*PWS*X)*DRHH+RHF/X)
431      ZETA=1./ZETA
432      DXCW=ZETA*(1.-PW)/WN
433      XSAVE=X/2.
434      C DETERMINE NEW VALUE CF SILICA GEL WATER CONTENT BY SIMPSONS RULE
435      X=DXDW*(W-WN)+X
436      IF(X.LI.0.0) X=XSAVE
437      105 CONTINUE

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431 WRITE(6,106) T,W,X
432 106 FORMAT(/13,'***ERROR*** NOT CONVERGING, T=',F5.1,'W=',F6.4,'X=',
      IF6.4)
C DETERMINE VALUE OF PARTIAL OF TEMP. WITH RESPECT TO HUMIDITY RATIO AT CONSTANT
C WATER CONTENT AND ENTHALPY
433 107 ALPH=(TF-32.+HFG)/(1.24+W*.4123)
434 ALPW=(PW-1.0)*PWS/(W*DPWS*RHH)
435 DHDX=(TF-32.0)+HFG*11.-RHH)
436 CHADW=TF-32.+HFG
437 ALAMCA=1.-DHDX/DHADW
C CALCULATE INTEGRAL OF (1-HS/HV) DX ;NOTE X= SILICA GEL WATER CONTENT
      IF(X.LE..01) SUMR=-11.395*X**2
438 IF(X.GT..01).AND.(X.LE..04)) SUMR=-2133.3*X**4+180.7*X**3-5.895*X
439 1**2-1.1557*X+8.476E-04
440 IF(X.GT..04).AND.(X.LE..1)) SUMR=-15.13*X**3+4.484*X**2-.5922*X+8.
      17731E-03
441 IF(X.GT..1).AND.(X.LE..35)) SUMR=2.2889*X**4-2.3493*X**3+.9483*X**
      12-.2789*X-2.0959E-04
442 IF(X.GT..35) SUMR= .29158*X**2-1.2894*X
443 SIGMA=(.22+X-.5877*SUMR)/(1.24+W *(1.-.5877))
444 ANU=PWS/(ZETA*RFF*DPWS)
445 TERM1=(ALPH*ALAMDA+SIGMA*ANU+ALPW)
446 TERM2=(TERM1)**2-4.*ALPW*ALAMDA*ALPH)**.5
C USING CALCULATED DATA DETERMINE ALF (-PARTIAL CF TEMP. WITH RESPECT TO HUMIDITY
C RATIO AT CONSTANT "F" FUNCTION)
447 ALF (NN,1)=TERM1+TERM2)/(2.*1.8)
448 ALF (NN,2)=(TERM1-TERM2)/(2.*1.8)
C CALCULATE CHARACTERISTIC SPECIFIC CAPACITY RATIOS GAMMA
449 GAMMA(NN,1) =(SIGMA*ALPW)/(ALPW-ALF(NN,1 )*.1.8)
450 GAMMA(NN,2) =(SIGMA*ALPW)/(ALPW-ALF(NN,2 )*.1.8)
451 103 IF(IND1.EQ.1) DUMMY=ALF(NN,1)
452 IF(IND1.EQ.2) DUMMY=GAMMA(NN,1)
453 RETURN
454 END
C LEAKAGE SUBPROGRAM

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455 SUBROUTINE LEAK (W11,W21,T11,T21,W1OP,W2OP,T1OP,T2OP,W1O,W2O,T1O,T
      12O,XEFF)
456 COMMON CP,AC,LEN,TAU,ALPA,NCQVS,EPP,EPG,D,WIDTH,AMU,AK,DEL,W,CB,PI
      1,IO,HFILMI,KFIN,UP,QDUCT,CTYPE,IFLAG,AMASS,WXE,PHI,ALONG,E,SL,DIDC
      2,DICD,OD,AFLOW,SHR,SQU,SEFF,SQTL,SQAU,SQSURP,R,GAUX,CIL,QSURP,AL,
      3FL,VOID,RPS,RHOF,U,DD,FRIC,ALFA,SIG,GAM,BLEAK,PLEAK,RLEAK,CTAU,CAL
      4PA,AFF,ALF,GAMA,KCUNT,QDEHU,PERB,PERC,REFLEC,CMEGA,AF,AV,HMU,ENU,
      5WMASS
457 DIMENSION SHR(1),SQU(1),SEFF(1),SQTL(1),SQAUX(1),SQSURP(1), QU
      1(1,24),QAUX(1,24),QSURP(1,24),QTL(1,24),ALFA(4,2),SIG(2,2),GAM(2,2
      2),ALF(4,2),GAMA(4,2),QDEHU(1,24),QDUCT(1,24),R(1,24),CMEGA(1,24)
      C CHANGE ALL TEMPERATURES TO DEGREES KELVIN.
      T11=T11+273.16
      T21=T21+273.16
      T1OP=T1OP+273.16
      T2OP=T2OP+273.16
458
459 C FIRST FOUR EQUATIONS ARE FOR DEHUMIDIFIER WITH NO HEAT EXCHANGER (IE. EFFECTI
      C VENESS EQUAL TO 0.0)
460
461 C T11,T21,W11,W21 ARE INLET CONDITIONS
      C T1OP,T2OP,W1OP,W2OP ARE OUTLET CONDITIONS WITH NO LEAKAGE
      C T1O,T2O,W1O,W2O ARE OUTLET CONDITIONS WITH LEAKAGE
462 T1O=(PLEAK*T11+RLEAK*T21+(1.0-BLEAK-PLEAK)*T1CP)/(1.0-BLEAK+RLEAK)
463 W1O=(PLEAK*W11+RLEAK*W21+(1.0-BLEAK-PLEAK)*W1OP)/(1.0-BLEAK+RLEAK)
464 IF(XEFF.NE.0.0) GO TO 10
      C THESE EQUATIONS USED IF NO HEAT EXCHANGER IN USE
465 T2O=(PLEAK*T21+BLEAK*T11+(1.0-PLEAK-RLEAK)*T2OP)/(1.0+BLEAK-RLEAK)
466 W2O=(PLEAK*W21+BLEAK*W11+(1.0-PLEAK-RLEAK)*W2OP)/(1.0+BLEAK-RLEAK)
467 T1C=T1O-273.16
468 T2O=T2O-273.16
469 T11=T11-273.16
470 T21=T21-273.16
471 T1OP=T1OP-273.16
472 T2OP=T2OP-273.16
473 RETURN
      C THESE EQUATIONS APPLY IF HEAT EXCHANGER HAS BEEN INSTALED

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474 10 T20=(1.0-BLEAK-PLEAK)*T2OP+PLEAK*T21+HLEAK*111
475 W20=(1.0-BLEAK-PLEAK)*W2OP+PLEAK*W21+BLEAK*W11
476 T10=T10-273.16
477 T20=T20-273.16
478 T11=T11-273.16
479 T21=T21-273.16
480 T1OP=T1OP-273.16
481 T2OP=T2OP-273.16
482 RETURN
483 END

484 SUBROUTINE COLPER (J,I,T,V,H,YFD,TRD,HR,EFF,QU,UL,UT,FP,FR,TIC,TOC
      1,IHR,IDATE,IMIN)
485 INTEGER*4 CTYPE
486 REAL*4 LEN,IC,NCOVS,KFIN,INFIST
487 COMMON CP,AC,LEN,TAU,ALPA,NCOVS,EPP,EPG,D,WIDTH,AMU,AK,DEL,W,CB,PI
      1,1D,HFILM1,KFIN,UB,QDUCT,CTYPE,IFLAG,AMASS,WXE,PHI,ALCNG,E,SL,DTDC
      2,DTCD,OD,AFLOW,SHR,SQU,SEFF,SQTL,SQAU,SQSURP,R,QAUX,CTL,QSURP,AL,
      3FL,VUID,RPS,RHOF,U,DD,FRIC,ALFA,SIG,GAM,BLEAK,PLEAK,RLEAK,CTAU,CAL
      4PA,AFF,ALF,GAMA,KCUNT,QDEHU,PERB,PERC,REFLEC,OMEGA,AF,AV,HMU,ENU,
      5WMASS
488 DIMENSION SHR(1),SQU(1),SEFF(1),SQTL(1),SQAUX(1),SQSURP(1), QU
      1(1,24),QAUX(1,24),QSURP(1,24),QIL(1,24),ALFA(4,2),SIG(2,2),GAM(2,2
      2),ALF(4,2),GAMA(4,2),QDEHU(1,24),QDUCT(1,24),R(1,24),CMEGA(1,24)
489 DIMENSION THETAT(1,24),THETAZ(1,24),RB(1,24),T(1,24),V(1,24),H(1,2
      24),EFF(1,24),TIC(1,24),TOC(1,24),TFD(1,24),TRD(1,24),HR(1,24),IDAT
      3E(1)
      K=I
      L=J
      SIGMA = 5.669E-C8
      IF (J.NE.1) GO TO 60
      C INITIALIZE DAILY ICTAL VALUES
      SHR(1)=0.0
      SQU(1)=0.0
      SEFF(1)=0.0

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497 SQT(L)=0.0
498 SQALX(I)=0.
499 QSURP(I)=0.
500 C CHANGE TMIN FROM DEG C TO DEG K
501 60 T(I,J)=T(I,J)+273.16
502 TMIN=TMIN+273.16
503 IFD(I,J)=TFD(I,J)+273.16
504 QAUX(I,J)=0.0
505 QSURP(I,J)=0.0
506 DATE=FLOAT(IDATE(K))
507 C DETERMINE AMOUNT OF SOLAR RADIATION INCIDENT ON TILTED COLLECTOR SURFACE
508 C USING TERMINOLOGY FROM BECKMAN AND DUFFIE "SCLAR ENERGY THERMAL PROCESSES"
509 C CHAP.#3
510 BB=360.*( DATE-81.)*PI/1364.*180.)
511 E=9.87*SIN(2.*BB)-7.53*COS(BB)-1.50*SIN(BB)
512 DEL=23.45*SIN(2.*PI*((1284.+DATE)/365.))
513 C CONVERT GEOMETRY CHARACTERISTICS FROM DEGREES TO RADIANS
514 RS=SL*PI/180.
515 RPHI=PHI*PI/180.
516 RDEL=DEL*PI/180.
517 ROMEGS=ARCCOS(-(TAN(RDEL)*TAN(RPHI)))
518 CMEGAS=ROMEGS*180.0/PI
519 TIME=FLOAT(IHR)
520 STIME=TIME-((-L-4.0*(90.0-ALCNG))/60.0)
521 OMEGA(K,L)=15.0*(12.0-STIME)
522 C CHECK SUNRISE AND SUNSET ANGLE
523 IF (ABS(OMEGA(K,L)).GE.ABS(OMEGAS)) GO TO 50
524 ROMEA=CMEGA(K,L)*PI/180.
525 C DETERMINE RB
526 THETAT(K,L)=180./PI*ARCOS((COS(RS)*COS(RDEL)*COS(ROMEGA))+SIN(RS)*S
527 IN(RDEL)))
528 THETAZ(K,L)=180./PI*ARCOS(SIN(RDEL)*SIN(RPHI)+CCS(RDEL)*COS(RPHI)*
529 COS(ROMEGA))
530 RB(K,L)=CCS(THETAT(K,L)*PI/180.)/COS(THETAZ(K,L)*PI/180.)
531 R(K,L)=PERB*RB(K,L)+PERD*(1.+COS(RS))/2.+(1.-COS(RS))*REFLEC/2.

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523   HR(K,L)=R(K,L)*H(K,L)
524   GO TO 40
525   50 OMEGA(K,L)=0.0
526   RB(K,L)=0.0
527   R(K,L)=0.0
528   HR(K,L)=0.0
529   THETAT(K,L)=0.0
530   THETAZ(K,L)=0.0
531   40 IF (KOUNT.NE.1) GO TO 360
532   IF (CTYPE.EQ.0) GO TO 360
533   AFLOW=D*WIDTH
534   IF (CTYPE.EQ.1) WRITE(6,201)
535   IF (CTYPE.EQ.2) WRITE(6,202)
536   IF (CTYPE.EQ.3) WRITE(6,203)
537   WRITE(6,300)
538   201 FORMAT(/15,'COLLECTOR TYPE #1 -WATER COOLED COLLECTOR')
539   202 FORMAT(/15,'COLLECTOR TYPE #2 -AIR COOLED COLLECTOR WITH GLASS/MET
540   IAL CHANNEL')
541   203 FORMAT(/15,'COLLECTOR TYPE #3 -AIR COOLED COLLECTOR WITH METAL/MET
542   IAL CHANNEL')
543   WRITE(6,301) CALPA
544   WRITE(6,302) CIAU
545   WRITE(6,303) EPP
546   WRITE(6,304) EPG
547   WRITE(6,305) NCGVS
548   WRITE(6,306) LEN
549   WRITE(6,307) WIDTH
550   IF (CTYPE.EQ.1) GO TO 350
551   WRITE(6,308) D
552   WRITE(6,309) AFLOW
553   WRITE(6,310) UB
554   WRITE(6,311) CP
555   WRITE(6,312) AMU
556   WRITE(6,313) AK
557   GO TO 360

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556 WRITE(6,314) CD
557 WRITE(6,315) ID
558 WRITE(6,316) h
559 WRITE(6,317) F-FILMI
560 WRITE(6,318) KFIN
561 WRITE(6,319) CB
562 WRITE(6,320) CP
563 FORMAT(1,1,1,15,1,***** COLLECTOR PARAMETERS *****)
564 FORMAT(15,1,ABSORPTANCE OF PLATE ',F5.3)
565 FORMAT(15,1,TRANSMITTANCE OF GLAZING ',F5.3)
566 FORMAT(15,1,EMITTANCE OF PLATE ',F5.3)
567 FORMAT(15,1,EMITTANCE OF GLAZING ',F5.3)
568 FORMAT(15,1,NUMBER OF COVERS ',F3.1)
569 FORMAT(15,1,LENGTH OF EACH COLLECTOR ',F5.2,1 M')
570 FORMAT(15,1,WIDTH OF EACH COLLECTOR ',F4.2,1 M')
571 FORMAT(15,1,FLOW CHANNEL DEPTH ',F5.3,1 M')
572 FORMAT(15,1,FLOW CHANNEL AREA ',F5.3,1 M**2)
573 FORMAT(15,1,BOTTOM LOSS COEFFICIENT ',F4.2,1 W/M**2-C')
574 FORMAT(15,1,SPECIFIC HEAT OF AIR ',F6.1,1 J/KG-C')
575 FORMAT(15,1,VISCOSITY OF AIR ',F9.7,1 KG/M-S')
576 FORMAT(15,1,THERMAL CONDUCTIVITY OF AIR ',F6.4,1 W/M-C')
577 FORMAT(15,1,OUTSIDE DIAMETER OF TUBE ',F6.4,1 M')
578 FORMAT(15,1,INSIDE DIAMETER OF TUBE ',F4.6,1 M')
579 FORMAT(15,1,DISTANCE BETWEEN TUBES ',F6.4,1 M')
580 FORMAT(15,1,FILM HEAT TRANSFER COEFFICIENT FOR INSIDE OF TUBE ',F5
      1.1,1 W/M**2-C')
581 FORMAT(15,1,THERMAL CONDUCTIVITY OF FIN ',F5.1,1 W/M-C')
582 FORMAT(15,1,BOND CONDUCTANCE ',F5.1,1 W/M-C')
583 FORMAT(15,1,SPECIFIC HEAT OF WATER ',F6.1,1 J/KG-C')
584 UT=0.
585 UL=0.
586 FP=0.
587 FR=0.

C DETERMINE COLLECTOR USEFUL HEAT GAIN-QU
C CHANGE HR UNITS FROM KJ/HR-M**2 TO W/M**2

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588 HR(I,J)=HR(I,J)/3.6
589 H(I,J)=H(I,J)/3.6
590 TIC(I,J)=IFD(I,J)-DTDC
591 IF (HR(I,J).LE.0.0) GO TO 42
592 CALL COLEFF(HR(I,J),T(I,J),V(I,J),EFF(I,J),QU(I,J),TIC(I,J),UT,UL,
      IFP,FR,IHEIAT(I,J))
593 GO TO 52
594 42 EFF(I,J)=0.0
595 QU(I,J)=0.
596 52 TOC(I,J)=(QU(I,J)/(AMASS*1005.0 )+TIC(I,J))
597 TRD(I,J)=TOC(I,J)-DTDC
      C DETERMINE HEAT LOSS IN DUCTWORK
598 QDUCT(I,J)=AMASS*CP*(IFD(I,J)-TIC(I,J))+TCC(I,J)-TRD(I,J))
      C DETERMINE AUXILIARY HEATING REQUIREMENTS AND TCTAL HEAT INPUT REQUIRED- QAUX,
      C QTL
599 IF (TRD(I,J).GE.TMIN) GO TO 80
600 QAUX(I,J)=AMASS*1005.0 *(TMIN-TRD(I,J))
601 QTL(I,J)=QAUX(I,J)+QU(I,J)
602 GO TO 90
      C DETERMINE HEAT AVAILABLE FOR STORAGE-QSURP
603 80 QSURP(I,J)=AMASS*1005.0*(TRD(I,J)-TMIN)
      C CHANGE TEMPERATURES FROM DEG K TO DEG C.
604 90 IFD(I,J)=IFD(I,J)-273.16
605 TRD(I,J)=TRD(I,J)-273.16
606 T(I,J)=T(I,J)-273.16
607 TIC(I,J)=TIC(I,J)-273.16
608 TOC(I,J)=TOC(I,J)-273.16
609 TMIN=TMIN-273.16
      C CALCULATE CAILY VALUES
610 SHR(I)=HR(I,J)+SHR(I)
611 SQU(I)= QU(I,J)+SQU(I)
612 SEFF(I)= EFF(I,J)+SEFF(I)
613 SQALX(I)=SQAUX(I)+QAUX(I,J)
614 QSURP(I)=QSURP(I)+QSURP(I,J)
615 RETURN

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616      END
617      SUBROUTINE COLEFF(HR,TA,VWIND,EFF,QU,YFIN,UT,UL,FP,FR,T+ETA)
C      THIS SUBPROGRAM THEORETICALLY DETERMINES THE SOLAR COLLECTOR PERFORMANCE
C      FOR COLLECTOR TYPES 1,2 AND 3 USING PROCEEDURES OUTLINED IN BECKMAN AND DUFF-
C      IE "SOLAR ENERGY THERMAL PROCESSES",CHAP.#7
C      COLLECTOR TYPES; #1 WATER COLLECTOR WITH BONDED TUBES (NOTE WATER/AIR
C      HEAT EXCHANGER WITH MUST BE SPECIFIED)
C      #2 AIR COLLECTOR WITH METAL/GLASS CHANNEL.
C      #3 AIR COLLECTOR WITH METAL/METAL CHANNEL
C
618      REAL*4 LEN,ID,NCOVS,KFIN,INFIST
619      INTEGER*4 CTYPE
620      COMMON CP,AC,LEN,TAU,ALPA,NCOVS,EPP,EPG,D,WIDTH,AMU,AK,DEL,W,CB,PI
1,ID,HFILMI,KFIN,UB,QDUCT,CTYPE,IFLAG,AMASS,WXE,PHI,ALONG,E,SL,DTDC
2,DTCD,UD,AFLOW,SHR,SQU,SEFF,SQTL,SQAUX,SQSURP,R,QAUX,CTL,QSURP,AL,
3FL,VOID,RPS,RHOF,U,DD,FRIC,ALFA,SIG,GAM,BLEAK,PLEAK,RLEAK,CTAU,CAL
4PA,AFF,ALF,GAMA,KCUNT,QDEHU,PERB,PERC,REFLEC,CMEGA,AF,AV,HMU,ENU,
5WMASS
621      DIMENSION SHR(1),SQU(1),SEFF(1),SQTL(1),SQAUX(1),SQSURP(1),
1      QAUX(1,24),QSURP(1,24),QTL(1,24),ALFA(4,2),SIG(2,2),GAM(2,2
2),ALF(4,2),GAMA(4,2),QDEHU(1,24),QDUCT(1,24),K(1,24),CMEGA(1,24)
SIGMA=5.669E-CE
UL=0.
QU=0.
I=0
G=AMASS/AC
IF(CTYPE.NE.0) GO TO 20
C      DETERMINE COLLECTOR EFFICIENCY IF PERFORMANCE CURVE IS GIVEN
C      DETERMINE VOLUME FLOW RATE TO COLLECTOR AREA RATIO
C      TREAT AIR AS IDEAL GAS
628      SVOL=(.2868*TFIN)/101.3
629      VDCI=AMASS*SVOL
C      PERFORMANCE EQUATIONS MUST BE PUT IN WITH UNITS OF VAC IN FT/MIN
630      VAC=(VDCI/AC)*(60./3048)
C      FOLLOWING EQUATIONS ARE FOR SOLARON SERIES 3000 COLLECTORS ONLY..

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631 FRIA=.1470+.2485*VAC-.035*VAC**2
632 FRUL=.098+.520*VAC-.073*VAC**2
633 COSIN=1./COS(THETA*PI/180.)
634 GM=COSIN-1.
635 AMOD=1.00636+.25793*GM-.38517*GM**2
C VALUE OF (TFIN-TA)/HR MUST BE IN HR*FI2*F/BTU
636 EFF=AMOD*(FRIA-FRUL*5.6785*(TFIN-TA)/HR)
637 QU=EFF*AC*HR
638 UT=0.0
639 UL=0.0
640 FP=FRUL
641 FR=FRIA
642 GO TO 120
643 20 TSAVE=TFIN
644 S= HR*CTAU*CALPA
645 TP =TA +25.0
646 IF (CTYPE.NE.1) GO TO 10
647 WCP=4200.
648 WG=WMASS/AC
649 CH=WMASS*4.2
650 CC=AMASS*1.0
651 ISP=1.0
652 THOUT=TFIN+(HR*AC/(WMASS*WCP))*5.5 +5.0
653 THIN=THOUT+.5*(HR*AC/(WMASS*WCP))
654 10 DO 100 IT=1,25
C PROCEED TO CALCULATE TOP LOSS COEFFICIENT
655 HWIND= 5.7+3.8*VWIND
656 FWIND= (1.0-0.04*HWIND+(5.00000E-04 )*HWIND**2.0)*(1.+0.058*NCUVS)
657 IEMPD= ABS(TP-TA)
658 FISTER = ((NCUVS/((344.0/TP)*((TEMPC)/(NCUVS+FWIND))**0.31)))+(1.
10/HWIND))
659 INFIST= 1.0/FISTER
660 RADTLR = SIGMA*(TP+TA)*(TP**2+TA**2)
661 RADID= (EPP+0.0425*NCUVS*(1.0-EPP))
662 ANRAD = 1.0/RADID

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663 RAD2D= ((2.0*NCOVS+FWIND-1.0)/EPG)-NCOVS
664 SECTER= RADIER/(ANRAD+RAD2D)
665 UT= INFISI+SECTER
C DETERMINE VALUE CF F PRIME
666 IF (CTYPE.NE.1) GO TO 40
667 UL =UT+UB
668 M2 = UL/(KFIN*DEL)
669 M=M2**0.5
670 SSS= (M*(W-OD))/2.0
671 F= (TANH(SSS))/SSS
672 FPN= 1.0/ UL
673 FPD=W*(1.0/(UL*(OD+(W-OD)*F)))+(1.0/CB)+(1.0/(PI*ID*HFILMI)))
674 FP= FPN/FPD
675 EXTER=EXP(-1.0*((LL*FP)/(WG*WCP)))
676 FR= (WG*WCP/UL)*(1.0-EXTER)
677 QU=AC*FR*(S-UL*(THOUT-TA))
678 EFF=QU/(AC*HR)
679 TFN=THOUT+QU/(2.*WMASS*WCP)
680 THOUS=THOUT
681 GO TO 80
682 EPS1=EPG
683 EPS2=EPP
684 IF (CTYPE.EQ.3) EPS1=EPP
685 HRAD=4.*SIGMA*TP**3/((1./EPS1)+(1./EPS2)-1.)
686 DH=D*2.
687 AFLCW=D*WIDTH
688 RE=AMASS*DH/(AFLOW*AMU)
689 IF (RE.GE.2000.) CNU=.0158*RE**.8
690 IF (RE.LT.2000.) CNU=5.2
691 H=CNU*AK/DH
692 IF (CTYPE.EQ.3) GO TO 50
C EQUATIONS FOR TYPE #2 COLLECTOR- METAL/GLASS CHANNEL
693 UL=(UT+UB)/(1.+(UT+UB)*F)/(H**2+2.*FRAD*H)
694 FP=1./((1.+HRAD*UT/(2.*HRAD*H+H*UT+H**2))
695 GO TO 60

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C EQUATIONS FOR TYPE #3 COLLECTOR- METAL/METAL CHANNEL
50 UL=UB+UT
696 FP=1./((1.+(UL/(H+1.)/(1./H+1./HRAD))))
697
698 60 EXTER=EXP(-1.0*(UL*FP)/(G*CP)))
699 FR= (G*CP/UL)*(1.C-EXTER)
700 QU= AC*FR*(S-UL*(TFIN-TA))
701 EFF=QU/(AC*HR)
702 TFM=TFIN+QU/(2.*AMASS*CP)
703 IF (CTYPE.NE.1) GO TO 70
704 RPF= 1.0 / (HFILMI*PI*ID*NGVS*LEN)
705 TPM=(QU*RPF)+TFM
706 GO TO 80
707 70 TC=((HRAD*S-(H+HRAD)*UT*(TFM-TA))/(2.*HRAD*H+H*UT+H**2+UT*HRAD))+T
    IFM
708 TPM= (QU/AC-H*(TC-TFM))/H+TFM
C CHECK T PLATE TO THAT ESTIMATED
709 80 DELTP = ABS(TPM-TP)
710 IF (DELTP.LE.0.5) GO TO 120
711 TP = TPM
712 I=I+1
713 IF (I.GE.25) GO TO 130
714 100 CONTINUE
715 130 WRITE(6,160)
716 160 FORMAT(//'***** NUMRR OF ITERATIONS EXCEEDED IN COLEFF SUBPROGRAM
    1 *****')
717 RETURN
C IF COLLECTOR HAS LOST HEAT COLLECTOR IS TURNED OFF
718 120 IF (CTYPE.EQ.1) GO TO 150
719 170 IF (QU.GE.0.0) RETURN
720 QU=0.0
721 EFF=0.0
722 RETURN
723 150 THIN=THOUT+(QU*AC*EFF)/(WCP*WMASS)
C DETERMINE HEAT EXCHANGER EFFECTIVENESS BASED UPON CALCULATED TEMPERATURES.
724 WXS= (CH*(THIN-THOUT))/(CC*(THIN-TFIN))

```

```

725 DXE=ABS(WXES-WXE)
726 C CHECK COMPUTED HEAT EXCHANGER EFFECTIVENESS AGAINST GIVEN VALUE.
727 IF (DXE.LE..01) GC TO 170
728 C IF DIFFERENCE GREATER THAN 1.0% DETERMINE NEW IHCUT
729 THOUT=THIN-CC*WXE*(THIN-TFIN)/CH
730 THOUT=THOUT+(THOUS-THOUT)/2.0
731 ISP=ISP+1
732 IF (ISP.GE.30) GO TO 130
733 GO TO 10
734 END
735 C SUBPROGRAM HEATX
736
737 SUBROUTINE HEATX (THIN,TCIN,THOUT,ICOUT,FLOR,QDOT,WHIN,WCIN,WHOUT,
738 IHCOUT,XE)
739 COMMON CP,AC,LEN,IAU,ALPA,NCOVS,EPP,EPG,D,WIDTH,AMU,AK,DEL,W,CB,PI
740 I,IO,HFILMI,KFIN,UB,QDUCT,CTYPE,IFLAG,AMASS,WXE,PHI,ALONG,E,SL,DTDC
741 2,DTCD,OC,AFLOW,SHR,SQU,SEFF,SQTL,SQAU,SCSURP,R,CAUX,CTL,QSURP,AL,
742 3FL,VOID,RPS,RFOF,U,DO,FRIC,ALFA,SIG,GAM,BLEAK,PLEAK,RLEAK,CTAU,CAL
743 4PA,AFF,ALF,GAMA,KCUNT,QDEHU,PERB,PERL,REFLEC,CMEGA,AF,AV,HMU,ENU,
744 5WMASS
745 DIMENSION SHR(1),SQU(1),SEFF(1),SQTL(1),SQAU(1),SCSURP(1), QU
746 I(1,24),QAUX(1,24),QSURP(1,24),QTL(1,24),ALFA(4,2),SIG(2,2),GAM(2,2
747 2),ALF(4,2),GAMA(4,2),QDEHU(1,24),QDUCT(1,24),R(1,24),OMEGA(1,24)
748 THOUT=-XE*(THIN-TCIN)+THIN
749 ICOUT=XE*(THIN-TCIN)+TCIN
750 WCOU=WCIN
751 WHOUT=WHIN
752 C DETERMINE HEATX LEAKAGE
753 IF (XE.EQ.0.0) GC TO 20
754 CALL LEAK(WCIN,WHIN,TCIN,THIN,WCIN,WHIN,ICOUT,THOUT,WCOU,WHOUT,IC
755 OUT,THOUT,XE)
756 20 QDOT=AMASS*(CP*(TFIN-THOUT)+1061.*1055.7*2.205*(WHIN-WHOUT))
757 QDOT=ABS(QDOT)
758 RETURN
759 END

```

RJS,P=50,TIME=(,25),NOLIST

\$JOB

\$ENTRY

[illegible]

***** DEHUMIDIFICATION PARAMETERS *****

SURFACE AREA OF PACKING PER UNIT VOL. 1962.00 M**2/M**3

LENGTH IN FLOW DIRECTION 0.01700 M

TOTAL FRONTAL AREA EXPOSED TO FLOW 0.03640 M**2

VOID FRACTION 0.750

ROTATING SPEED OF MATRIX 0.001500 REV/S

DENSITY OF AIRSTREAM 1.110 KG/M**3

DESICCANT DIMENSION 0.003100 M

AIR MASS FLOW RATE 0.02835 KG/S

MATRIX MASS/ AIR MASS IN MATRIX 592.0

***** CALCULATED DATA *****

TIME PER REVOLUTION 666.67 S

AIR VELOCITY 0.936 M/S

EST. PRESSURE DROP THRU MATRIX 292.68 N/M**2

**** HOURLY TEMPERATURE AND HUMIDITY MAP ****

THURSDAY MAY 1, 1981
HEAT EXCHANGER EFFECTIVENESS 0.000

BYPASS LEAKAGE RATE= 0.400
PERIPHERY LEAKAGE RATE= 0.200
RECIRCULATION LEAKAGE RATE= 0.400

***** VARIABLE REGENERATION TEMPERATURES *****

TEMPERATURE IN DEG C

HUMIDITY RATIO IN KGW/KGA

**** PROCESS SIDE ****

HOUR	AMBIENT			DEHUMIDIFIER			SENSIBLE HEAT EXCHANGER							
	TEMP	HUMIDITY	TEMP	HUMIDITY	TEMP	HUMIDITY	INLET		OUTLET		INLET		OUTLET	
13	21.0	0.0031	24.0	0.0031	45.8	0.0021	45.8	0.0021	45.8	0.0021	45.8	0.0021	45.8	0.0021
14	21.5	0.0030	24.5	0.0030	46.4	0.0020	46.4	0.0020	46.4	0.0020	46.4	0.0020	46.4	0.0020
15	22.0	0.0029	25.0	0.0029	45.5	0.0020	45.5	0.0020	45.5	0.0020	45.5	0.0020	45.5	0.0020
16	22.2	0.0027	25.2	0.0027	42.5	0.0019	42.5	0.0019	42.5	0.0019	42.5	0.0019	42.5	0.0019
17	21.3	0.0027	24.3	0.0027	38.2	0.0019	38.2	0.0019	38.2	0.0019	38.2	0.0019	38.2	0.0019
18	21.1	0.0024	24.1	0.0024	32.0	0.0019	32.0	0.0019	32.0	0.0019	32.0	0.0019	32.0	0.0019

**** HOURLY TEMPERATURE AND HUMIDITY MAP ****

THURSDAY MAY 1, 1981

HEAT EXCHANGER EFFECTIVENESS 0.000

BYPASS LEAKAGE RATE= 0.400

PERIPHERY LEAKAGE RATE= 0.200

RECIRCULATION LEAKAGE RATE= 0.400

**** VARIABLE REGENERATION TEMPERATURES ****

TEMPERATURE IN DEG C

HUMIDITY RATIO IN KGW/KGA

**** REGENERATION SIDE ****

HOUR	SENSIBLE HEAT EXCHANGER				SOLAR COLLECTOR				AUX. HEATER		DEHUMIDIFIER			
	INLET		OUTLET		INLET		OUTLET		INLET		INLET		OUTLET	
	TEMP	HUMIDITY	TEMP	HUMIDITY	TEMP	HUMIDITY	TEMP	HUMIDITY	TEMP	HUMIDITY	TEMP	HUMIDITY	TEMP	HUMIDITY
13	24.0	0.0031	24.0	0.0031	24.0	0.0031	71.3	0.0031	70.2	0.0031	65.6	0.0031	48.1	0.0041
14	24.5	0.0030	24.5	0.0030	24.5	0.0030	69.1	0.0030	68.1	0.0030	70.5	0.0030	49.0	0.0040
15	25.0	0.0029	25.0	0.0029	25.0	0.0029	64.8	0.0029	63.0	0.0029	68.3	0.0029	48.0	0.0038
16	25.2	0.0027	25.2	0.0027	25.2	0.0027	53.7	0.0027	52.7	0.0027	61.4	0.0027	44.3	0.0034
17	24.3	0.0027	24.3	0.0027	24.3	0.0027	26.1	0.0027	25.1	0.0027	52.3	0.0027	38.9	0.0033
18	24.1	0.0024	24.1	0.0024	24.1	0.0024	24.1	0.0024	24.1	0.0024	35.6	0.0024	32.0	0.0028

*** TOTAL HOURLY HEAT BALANCE MAP ***

THURSDAY MAY 1, 1981

RATES IN WATTS
TOTALS IN KJ

HEAT EXCHANGER EFFECTIVENESS 0.000

BYPASS LEAKAGE RATE= 0.400

PERIPHERY LEAKAGE RATE= 0.200

RECIRCULATION LEAKAGE RATE= 0.400

HOURLY	HEAT TRANSFERRED IN DEHUMIDIFIER RATE TOTAL	HEAT TRANSFERRED IN HEAT EXCHANGER RATE TOTAL	HEAT LOST IN DUCTING RATE TOTAL	SOLAR HEAT GAIN IN COLLECTOR RATE TOTAL	AUXILIARY HEAT SUPPLIED RATE TOTAL	SURPLUS SOLAR HEAT AVAILABLE FOR STORAGE RATE TOTAL
13	634.8	2285.1	0.0	0.0	0.0	0.0
14	639.3	2301.5	0.0	0.0	0.0	0.0
15	606.7	2184.1	0.0	0.0	0.0	0.0
16	523.0	1882.7	0.0	0.0	0.0	0.0
17	426.8	1536.4	0.0	0.0	0.0	0.0
18	275.6	992.2	0.0	0.0	0.0	0.0

HOURLY SUMMARY OF AMBIENT CONDITIONS FOR THURSDAY MAY 1, 1981

HOUR	AMBIENT TEMP. (DEG C)	WIND SPEED (M/S)	RADIATION (W/M**2)	OMEGA (DEG)	HR (W/M**2)	R
13	21.00	3.50	934.7	-9.1	979.6	1.048
14	21.50	3.10	880.6	-24.1	911.9	1.036
15	22.00	2.70	797.2	-39.1	803.7	1.008
16	22.20	2.20	652.8	-54.1	623.1	0.554
17	21.30	1.80	465.3	-65.1	391.9	0.842
18	21.10	1.80	222.2	-84.1	118.0	0.531

HOURLY PERFORMANCE OF COLLECTOR

COLLECTOR TYPE #0- MANUFACTURERS COLLECTOR PERFORMANCE CURVES INPUT

HOURLY	INLET TEMP. TO COLLECTOR (DEG C)	OUTLET TEMP. FROM COLLECTOR (DEG C)	USEFUL RATE OF HEAT GAIN (kW)	EFFICIENCY	TOP LOSS COEFFICIENT -UI (W/M**2-C)	TOTAL LOSS COEFFICIENT UL (W/M**2-C)	COLLECTOR EFFICIENCY FACTOR-FRUL	COLLECTOR EFFICIENCY FACTOR-FRFA
13	24.0	71.3	1346.8	0.433	0.000	0.000	0.708	0.438
14	24.5	69.1	1271.9	0.440	0.000	0.000	0.709	0.439
15	25.0	64.8	1134.0	0.445	0.000	0.000	0.709	0.439
16	25.2	53.7	811.1	0.410	0.000	0.000	0.710	0.439
17	24.3	26.1	51.9	0.042	0.000	0.000	0.708	0.438
18	24.1	24.1	0.0	0.000	0.000	0.000	0.708	0.438

CORE USAGE OBJECT CODE= 52032 BYTES,ARRAY AREA= 4856 BYTES,TOTAL AREA AVAILABLE= 159840 BYTES

DIAGNOSTICS NUMBER OF ERRORS= 0, NUMBER OF WARNINGS= 0, NUMBER OF EXTENSIONS= 1

CUMPLE TIME= 2.41 SEC,EXECUTION TIME= 5.57 SEC, 22.00.58 MONDAY 13 JUL 81 WAIFIV - JAN 1976 V115

APPENDIX C

BED LEAKAGE ANALYSIS

Three avenues of leakage around the desiccant and sensible heat exchanger wheels are available for the airstreams. These leakages are termed bypass, recirculation and periphery. The equations developed to simulate these leakages and their effects are influenced by the existence of a sensible heat exchanger in the dehumidification system. No Sensible Heat Exchanger Utilized. The three leakage flow paths for a single bed dehumidification system are shown on Figure C.1 (a). Side 1 is the process flow stream and side 2 is the regeneration flow stream. Each side is assumed to have a normalized flow rate of unity.

An energy balance on side 1 of the desiccant wheel results in the following equations.

$$\begin{aligned} \text{Heat}_{\text{out}} &= \text{Heat}_{\text{in}} \\ (1.0 - \text{BLEAK} + \text{RLEAK}) \cdot T_{10} \cdot C_p &= \\ &\text{PLEAK} \cdot C_p \cdot T_{1I} + (1.0 - \text{BLEAK} - \text{PLEAK}) \cdot C_p \cdot T_{10}' \\ &+ \text{RLEAK} \cdot C_p \cdot T_{2I} \end{aligned}$$

Solving for the outlet temperature of side 1, T_{10} , results in the following equation.

$$T_{10} = \frac{\text{PLEAK} \cdot T_{1I} + \text{RLEAK} \cdot T_{2I} + (1.0 - \text{BLEAK} - \text{PLEAK}) \cdot T_{10}'}{1.0 - \text{BLEAK} + \text{RLEAK}}$$

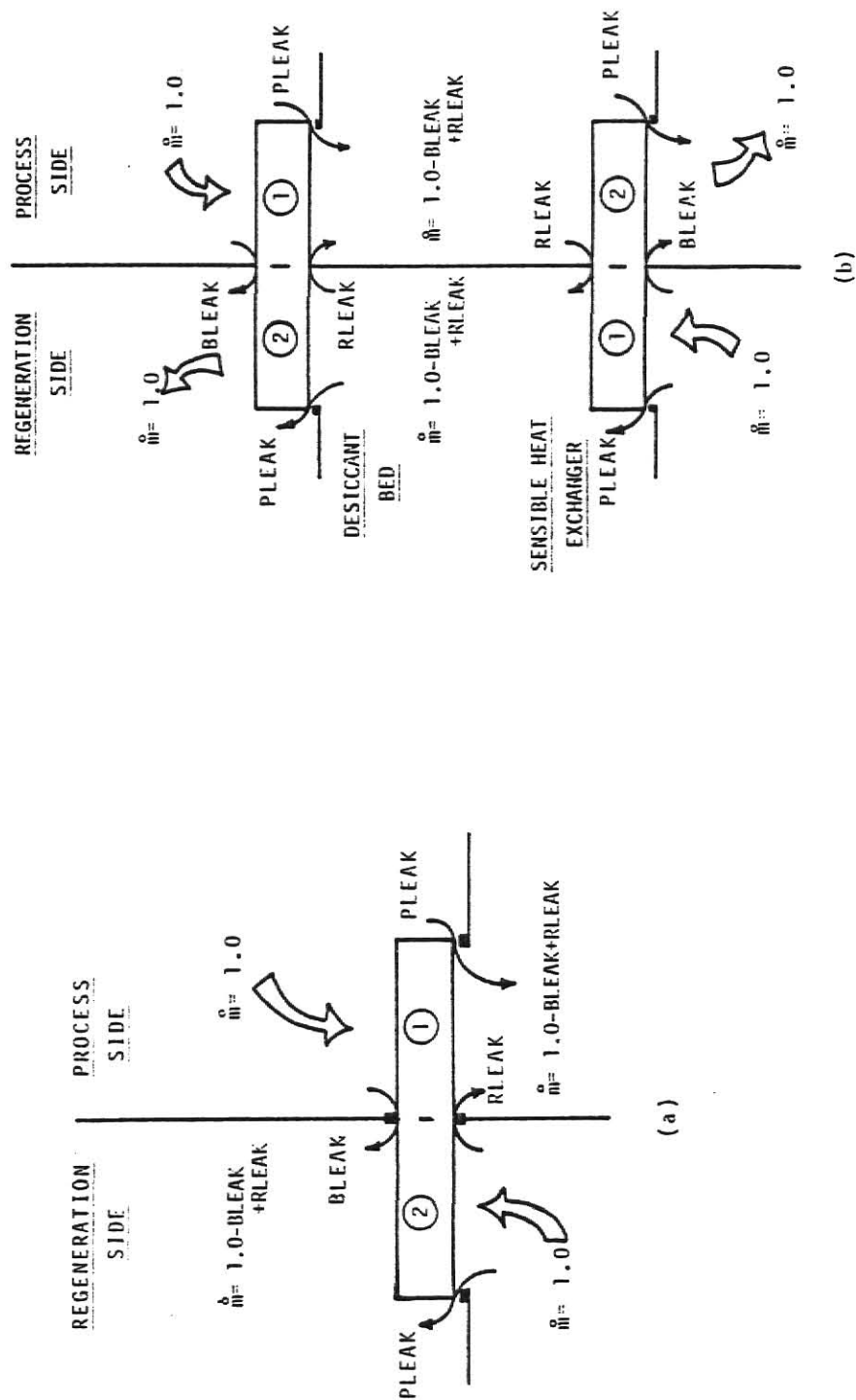


FIGURE C.1 SCHEMATIC OF LEAKAGE FLOW PATHS FOR DEHUMIDIFICATION SYSTEMS WITHOUT (a) AND WITH (b) SENSIBLE HEAT EXCHANGERS.

From a similar analysis of side 2 the following equation for outlet temperature results.

$$T_{20} = \frac{\text{PLEAK} \cdot T_{2I} + \text{BLEAK} \cdot T_{1I} + (1.0 - \text{PLEAK} - \text{RLEAK}) \cdot T_{20}}{1.0 + \text{BLEAK} - \text{RLEAK}}$$

A water mass balance on side 1 is used to determine the outlet humidity ratio including the effects of leakage.

$$\text{Mass Water}_{\text{out}} = \text{Mass Water}_{\text{in}}$$

$$W_{10} (1.0 - \text{BLEAK} + \text{RLEAK}) =$$

$$\text{PLEAK} \cdot W_{1I} + \text{RLEAK} \cdot W_{2I} + (1.0 - \text{BLEAK} - \text{PLEAK}) \cdot W_{10}$$

Therefore,

$$W_{10} = \frac{\text{PLEAK} \cdot W_{1I} + \text{RLEAK} \cdot W_{2I} + (1.0 - \text{BLEAK} - \text{PLEAK}) \cdot W_{10}}{1.0 - \text{BLEAK} + \text{RLEAK}}$$

A similar analysis of side 2 results in the following equations for the outlet humidity ratio of side 2 including leakage effects.

$$W_{20} \cdot (1.0 + \text{BLEAK} - \text{RLEAK}) =$$

$$\text{PLEAK} \cdot W_{2I} + \text{BLEAK} \cdot W_{1I} + (1.0 - \text{PLEAK} - \text{BLEAK}) \cdot W_{20}$$

$$W_{20} = \frac{\text{PLEAK} \cdot W_{2I} + \text{BLEAK} \cdot W_{1I} + (1.0 - \text{PLEAK} - \text{RLEAK}) \cdot W_{20}}{1.0 + \text{BLEAK} - \text{RLEAK}}$$

Sensible Heat Exchanger Utilized. When a sensible heat exchanger is utilized in the dehumidifier system the same three leakage paths may exist. However, they may have different numerical values from those of the no heat exchanger design. To assure the values of RLEAK, BLEAK, and PLEAK are constant for both the desiccant and heat exchanger wheel the pressure drops through each wheel are assumed to be approximately the same. It should be noted, that from the leakage standpoint, the heat exchanger wheel is identical to an inverted desiccant wheel, thus permitting the

analysis of only one wheel to be applied to both cases. In each case, the normalized flow into side 1 of the wheel is unity and the flow into side 2 is $1.0 - \text{BLEAK} + \text{RLEAK}$.

An energy balance of side 1 of either wheel results in the following equations.

$$\begin{aligned} \text{Heat}_{\text{out}} &= \text{Heat}_{\text{in}} \\ (1.0 - \text{BLEAK} - \text{RLEAK}) \cdot C_p \cdot T_{10} &= \\ (1.0 - \text{BLEAK} - \text{PLEAK}) \cdot C_p \cdot T_{1p} &+ \text{PLEAK} \cdot T_{1I} \cdot C_p \\ &+ \text{RLEAK} \cdot C_p \cdot T_{2I} \end{aligned}$$

Solving this equation for the outlet temperature yields the following:

$$T_{10} = \frac{(1.0 - \text{BLEAK} - \text{PLEAK}) T_{10} + \text{PLEAK} \cdot T_{1I} + \text{RLEAK} \cdot T_{2I}}{(1.0 - \text{BLEAK} + \text{RLEAK})}$$

Performing a similar analysis of side 2 gives the following results,

$$\begin{aligned} ((1.0 - \text{BLEAK} + \text{RLEAK}) + \text{BLEAK} - \text{RLEAK}) \cdot C_p \cdot T_{20} \\ = ((1.0 - \text{BLEAK} + \text{RLEAK}) - \text{RLEAK} - \text{PLEAK}) \cdot C_p \cdot T_{20} \\ + \text{PLEAK} \cdot C_p \cdot T_{2I} + \text{BLEAK} \cdot C_p \cdot T_{1I} \end{aligned}$$

Therefore,

$$T_{20} = (1.0 - \text{BLEAK} - \text{PLEAK}) \cdot T_{20} + \text{PLEAK} \cdot T_{2I} + \text{BLEAK} \cdot T_{1I}$$

Considering a mass balance on side 1 of the wheel results in the following derivation for the outlet humidity ratio.

$$\begin{aligned} \text{Mass Water}_{\text{out}} &= \text{Mass Water}_{\text{out}} \\ (1.0 - \text{BLEAK} + \text{RLEAK}) \cdot W_{10} &= \end{aligned}$$

$$W_{10} = \frac{\text{PLEAK} \cdot W_{1I} + \text{RLEAK} \cdot W_{2I} + (1.0 - \text{BLEAK} - \text{PLEAK}) \cdot W_{1p}}{(1.0 - \text{BLEAK} + \text{RLEAK})}$$

A similar mass balance of side 2 yields the following equation for outlet humidity ratio.

$$W_{20} = (1.0 - \text{BLEAK} - \text{PLEAK}) \cdot W_{20} + \text{PLEAK} \cdot W_{2I} + \text{BLEAK} \cdot W_{1I}$$

APPENDIX D

UNCERTAINTY ANALYSIS

The uncertainties in the experimental dehumidification parameters; temperature, dew point, air flow rate, and solar radiation, were determined based upon the following relationship:

$$\lambda_p = \sqrt{\sum_{N=1}^N S_N^2 \lambda_N^2} \quad (D.1)$$

This equations states that the uncertainty, λ_p , in a measured parameter, is equal to the square root of the sum of the squares of the λ_N and S_N product. The value S_N is the sensitivity factor for the λ_N contributing uncertainties. This relationship allows the determination of the combined effects of several sources of uncertainty encountered while one measurement is taken.

For this analysis, S_N was given the value of unity, meaning each source of uncertainty was assumed to have equal significance. The values of λ_p are presented as an uncertainty range in the units of the parameter measured.

Temperature. The uncertainty in temperature is a result of three major sources of error: the uncertainty resulting in the nonhomogeneity of the thermocouples, thermocouple nonlinearity, and data logger resolution and compensation. Based upon manufacturers information, test results, or educated approximation these uncertainties were given the following values.

$$\lambda_{\text{thermocouples}} = 0.1^{\circ}\text{C}$$

$$\lambda_{\text{linearity}} = 0.1^{\circ}\text{C}$$

$$\lambda_{\text{resolution}} = 0.5^{\circ}\text{C}$$

Therefore, the overall temperature uncertainty is 0.5°C , from equation D.1.

Dew Point. The uncertainty in dew point temperature could be attributed to three uncertainty sources also; the uncertainty in linearity, dew point meter accuracy, and data logger resolution. These values were given the following values based upon manufacturers data and experimental approximation.

$$\lambda_{\text{linearity}} = 0.5^{\circ}\text{C}$$

$$\lambda_{\text{dew point meter}} = 0.3^{\circ}\text{C}$$

$$\lambda_{\text{data logger}} = 0.1^{\circ}\text{C}$$

Using equation D.1 results in an overall dew point uncertainty of 0.6°C .

Air Flow Rate. Because the regeneration and process flow streams were measured by two separate methods the uncertainty of each method was determined.

For measurement of the process flow stream, an electronic flowmeter was utilized. The individual uncertainties are a result of the following causes: flow meter and transducer accuracy, flow rate linearity, and data logger resolution. These uncertainties have the following estimated or specified values.

$$\lambda_{\text{flow meter}} = 1 \text{ cfm}$$

$$\lambda_{\text{transducer}} = 1 \text{ cfm}$$

$$\lambda_{\text{linearity}} = .5 \text{ cfm}$$

The total process flow rate uncertainty is therefore 1.5 cfm (0.7 l/s).

No manufacturers' data was available for the oil manometers used to measure regeneration flow rate. Therefore the linearity and flow meter uncertainty were assumed to be one half the smallest division of the manometer or .05 in Wg. This reading corresponds to approximately 4.0 cfm at a total flow of 50 cfm. The total regeneration flow rate uncertainty is therefore 5.6 cfm (2.5 l/s) by using equation D.1.

Solar Radiation. The uncertainty in the measurement of solar radiation results from the pyranometer accuracy and linearity and the data logger resolution. These values based upon information from the manufacturers are as follows.

$$\lambda_{\text{pyranometer}} = 4.5 \text{ BTU/hr ft}^2$$

$$\lambda_{\text{data logger}} = 0.2 \text{ BTU/hr ft}^2$$

$$\lambda_{\text{linearity}} = 2.0 \text{ BTU/hr ft}^2$$

The overall solar radiation uncertainty is therefore 4.9 BTU/hr ft² (15.5 W/m²).

ACKNOWLEDGEMENTS

I would like to express my sincere appreciation to my major advisor, Dr. Herbert Ball for his patience, encouragement, comments and sense of humor, and, especially for physical support during the construction of the apparatus.

I would also like to thank Prof. Robert Crank and Dr. Do Sup Chung for the time they spent serving on my committee. Also a thanks to the Mechanical Engineering Department and Dr. Paul Miller for financial support during my graduate program.

Finally a special thanks to Pat for being my favorite friend and constant source of support and understanding.

VITA

Richard J. Singer

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THE DEHUMIDIFICATION OF AIR USING
SOLAR REGENERATED ROTATING
BEDS OF SILICA GEL

by

RICHARD J. SINGER

B.S., University of Missouri-Rolla, 1975

ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the
requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY

Manhattan, Kansas

1981

ABSTRACT

The dehumidification of air utilizing rotating beds of silica gel regenerated by solar energy was computer modeled and experimentally investigated. The use of rotating beds of silica gel was found to be a workable method for air dehumidification. The silica gel bed was continuously regenerated using high temperature air heated by solar collectors. When the solar collectors provided insufficient energy an auxiliary electric heater was energized. The computer model was developed to predict the performance of the solar dehumidifier, and included numerous subroutines modeling the desiccant bed, heat exchanger, solar collectors, and flow stream leakage. This computer program was found acceptable to predict the performance of the experimental solar dehumidifier.