

A FEASIBILITY STUDY OF A SOLAR DESICCANT AIR-CONDITIONING SYSTEM—PART I: PSYCHROMETRICS AND ANALYSIS OF THE CONDITIONED ZONE

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SUMMARY

A desiccant dehumidifier in conjunction with evaporative coolers can reduce air conditioning operating costs significantly since the energy required to power a desiccant cooling system is small and the source of this required energy (solar, waste heat, natural gas) can be diverse. Such a solid desiccant cooling system with a backup vapour compression system is simulated and the performance of the system is evaluated to study its feasibility in four cities in the United States. This paper describes the relevant psychrometric calculations and analyses of the conditioned zone required for simulating the transient performance of the system. Copyright © 1999 John Wiley & Sons, Ltd.

KEY WORDS: desiccants; air conditioning; psychrometrics; solar energy simulation

1. INTRODUCTION

1.1. Desiccants

Desiccants are chemicals with great affinity to moisture. They absorb (or release) moisture because of the difference in vapour pressure between the surface of the desiccant and the surrounding air. Dehumidification is said to occur when the vapour pressure of the surface of the desiccant is less than that of the surrounding air. Dehumidification continues until the desiccant material reaches equilibrium with the surrounding air. Regeneration of this desiccant is said to occur when the vapour pressure of the desiccant is larger than that of the surrounding air, which is usually achieved by heating the desiccant to its regeneration temperature and exposing it to an airstream.

Desiccants can be classified as either adsorbents, which absorb moisture without accompanying physical and chemical changes, or absorbents, which absorb moisture accompanied by physical or chemical changes. Desiccants can be solids or liquids and can hold moisture through adsorption or absorption as described earlier. Most absorbents are liquids and most adsorbents are solids. Several types of solid desiccants are widely used in desiccant cooling systems; silica gels, lithium chloride and molecular sieves. Silica gels are solid desiccants and adsorbents and contain numerous pores and capillaries in which water is condensed and contained. Silica gel has a high capacity to absorb moisture and then release it at a higher temperature. It is

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low in cost and available in sizes from 3/16 inch beads to powder-like grains. Lithium chloride (LiCl) is an absorbent and is found in dry form when each LiCl molecule holds two water molecules. If each LiCl molecule holds more than two water molecules, for former becomes liquid and continues to absorb moisture. Lithium chloride has a high capacity to absorb and hold moisture and is widely used in rotary wheel dehumidifiers. Molecular sieves are synthetic zeolites, a solid desiccant and an absorbent in the form of crystalline aluminosilicates produced by a thermal process. Molecular sieves show physical stability and high moisture-releasing capacity at high regenerating temperatures of 120 to 200°C and are recommended in direct gas-fired applications. Among the above, silica gels and lithium chloride are the most widely used desiccant materials in desiccant cooling systems.

1.2. Working principle of a solid desiccant cooling system

Solid desiccants are impregnated in a dehumidifier bed, usually a rotary disc which slowly rotates between the process and regeneration air streams. As the hot and humid process air passes through the desiccant wheel, the moisture is removed by the desiccant, and its temperature increases. The temperature of this process air, which is now hotter and drier, is reduced to the desired comfort conditions by means of sensible coolers (e.g. rotary heat exchangers, evaporative coolers, and cooling coils). The warm and humid return air from the conditioned space is further heated up to the required regeneration temperature of the desiccant and this regeneration stream of air is passed through the desiccant wheel to remove the moisture from the desiccant.

Advantages of using desiccant cooling systems include the following: (1) very small electrical energy is consumed and the sources for the regenerating thermal energy can be diverse (i.e. solar energy, waste heat, natural gas); (2) a desiccant system is likely to eliminate or reduce the use of ozone depleting CFCs (depending on whether desiccant cooling is used in conjunction with evaporative coolers or vapour compression systems, respectively); (3) control of humidity can be achieved better than those cases employing vapour compression systems since sensible and latent cooling occur separately; and, (4) improvement in indoor air quality is likely to occur because of the normally high ventilation and fresh air flow rates employed. Also, desiccant systems have the capability of removing airborne pollutants.

1.3. Munters' environmental cycle (MEC)

In 1955, Pennington saturated a rotary heat exchanger matrix with a desiccant solution making an adiabatic regenerative dehumidifier and coupled this dehumidifier with a heat source to a double regenerative evaporative cooler (Pesaran *et al.*, 1992). Munters environmental cycle is nothing but an improvement of the most famous Pennington (ventilation) cycle. In 1968, Munters improved Pennington's cycle by introducing parallel passages. The backup vapour compression system was provided to take care of cooling in case the desiccant cycle was unable to meet the cooling demand. Results clearly showed that there was no need for such a backup since the desiccant cooling system by itself was able to meet the cooling demand and during the simulation, the vapour compression system never came on.

Figure 1 shows a schematic of Munters' environmental desiccant cooling cycle along with a backup direct expansion (DX) cooling coil, whereas Figure 2 shows the corresponding psychrometric processes. Figure 3 gives a complete schematic of a solar-assisted desiccant air conditioning system which consists of two totally separated parallel passages, one serving the process air stream directed toward the conditioned space and the second serving the regeneration air stream directed towards the exhaust side.

The solar-assisted desiccant cooling cycle consists of the following major components in order to condition the process air to the desired comfort conditions: (1) a rotary wheel impregnated with a nominal silica gel matrix rotating continuously between the process and regeneration air streams; (2) a sensible heat wheel (rotary regenerator) also rotating between the process and regeneration air stream (the wheel transfers heat from the process side to the regeneration side); (3) process and regeneration side evaporative air coolers; (4) a solar collector storage subsystem for supplying the required thermal energy for regeneration; (5) a gas-fired

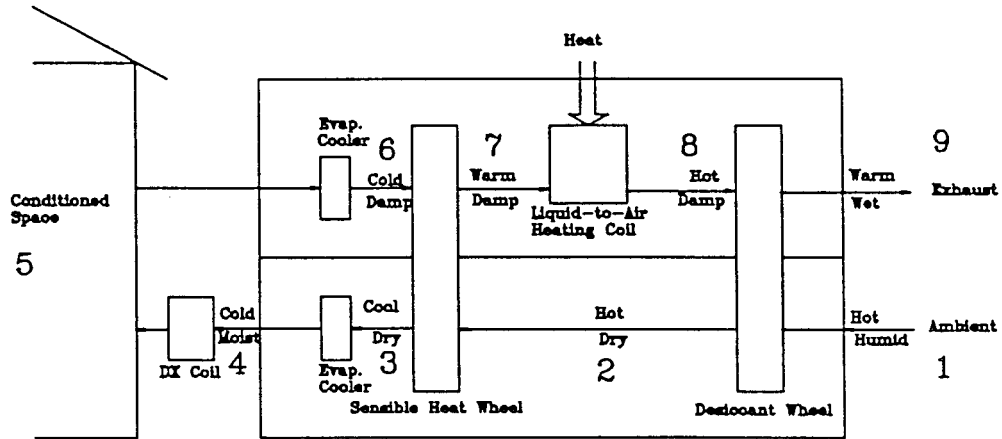


Figure 1. Schematic of the MEC cycle desiccant air conditioner with a cooling coil

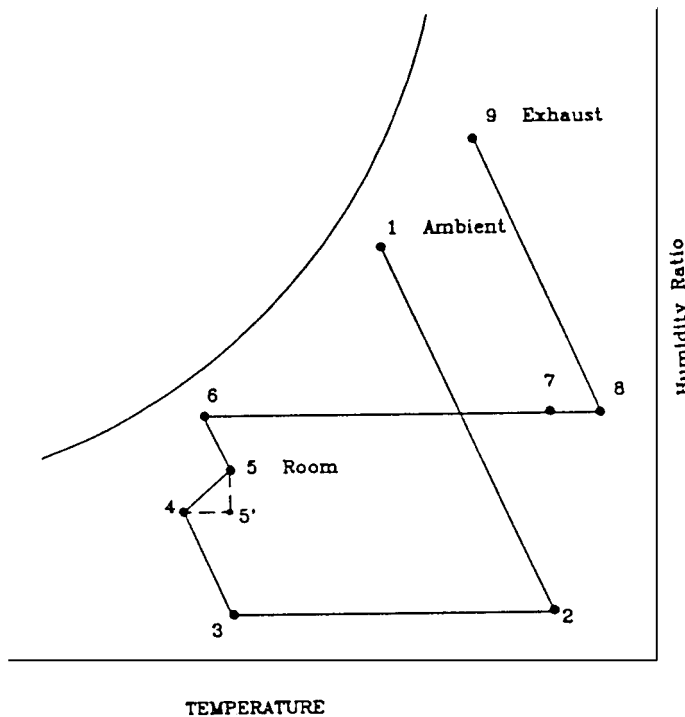


Figure 2. MEC cycle representation on a psychrometric chart

auxiliary heater as a backup for the solar subsystem; (6) a liquid-to-air heat exchanger coil; and, (7) two thermostats, one for activating the desiccant system and the other for activating the vapour compression system.

Referring to Figure 2, warm and wet ambient air is introduced in the process air at State 1. This process air is dried while it passes over the desiccant, resulting in hot, dry air as it exits the dehumidifier of State 2. This increase in temperature is due to the release of heat of condensation of the water vapour when moisture is removed by the desiccant material. This hot and dry air is then cooled sensibly as it passes through the rotary regenerator and is allowed to transfer much of the heat to the return air stream. The now cool and dry

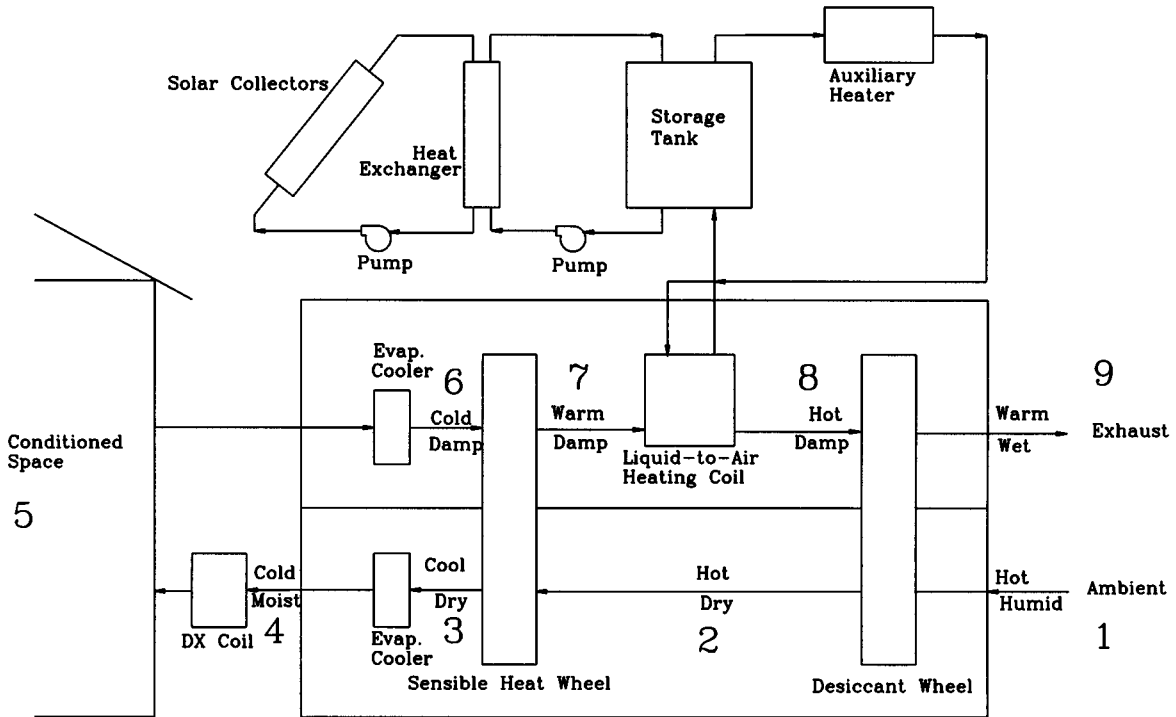


Figure 3. Schematic of a solar-assisted desiccant cooling system

process air at State 3 is further evaporatively cooled and humidified by the process side evaporative cooler to State 4 and is then introduced to the conditioned space through the supply duct. This cool air provides the comfort condition necessary to meet the building's cooling demands. After removing the building's sensible and latent cooling loads, air returns to the desiccant air conditioner through the return ducts and is again evaporatively cooled to State 6 from where it passes through the rotary regenerator and picks up the heat from the process air to attain the warm and damp State 7. The air then passes over the hot water coil, which circulates hot water obtained from the solar-auxiliary heater combination. Air is thus heated up to the required regeneration temperature of the desiccant material, and hence very hot and relatively humid air exits at State 8. This air then passes through the desiccant wheel in order to regenerate the desiccant. Even though this air is humid, it is sufficiently hot to regenerate the desiccant in the rotary dehumidifier matrix. Finally warm and very humid air exists at State 9 where it is exhausted to the surroundings.

The temperature of the return air at State 8 required to regenerate the desiccant is a function of the type of desiccant material used and the amount of dehumidification required in the process from 1 to 2. Smith and Hwang (1992) developed a dehumidifier model which relates the outlet and inlet states of the dehumidifier. The main energy input to such a system is primarily used for regenerating the desiccant. The necessary energy can be supplied by low-temperature heat sources for the regeneration, if the rotary dehumidifier matrix is properly constructed and appropriate selections are made for the type and amount of the desiccant. The thermal energy in this case, as described before, is provided by a combination of an array of solar flat plate collectors and natural gas. Other sources could be geothermal energy and waste heat recovery systems (such as heat pipes or condenser water heat). Only a small amount of electric energy is required for rotating the dehumidifier and the sensible heat wheel (rotary regenerator) along with the processes and evaporative side blowers and controls. Water consumption by the two evaporative coolers is small enough and hence the normal supply water pressure in a residential building is sufficient to keep them in continuous supply.

2. LITERATURE REVIEW

2.1. *Open-cycle solid desiccant systems*

The concept of desiccant cooling was first introduced by Hausen in 1935 (Pesaran *et al.*, 1992). Based on this, many inventors like Shipman (1936), Fleisher (1939), Larriva (1941) and Altenkirch (1941, 1944) tried to develop commercial desiccant cooling systems but were not successful in the subsequent years. However, Miller and Fonda (1933) invented the first rotary silica gel desiccant cooler, but because of lack of understanding, the potential of desiccant dehumidifiers was not realized.

Pennington patented the first ever desiccant cooling cycle which is commonly known as the ventilation cycle (Pesaran *et al.*, 1992). A rotary heat exchanger was saturated with a solid desiccant, converting the heat exchanger into an adiabatic regenerative dehumidifier, which takes in ambient air and adsorbs the moisture in it. This air is then sensibly and again evaporatively cooled before being introduced into the conditioned space. The return air is first evaporatively cooled and allowed to pass through a sensible heat exchanger to recover the heat of adsorption from the supply air. It is then heated with a low grade thermal energy source and used to regenerate the desiccant. Coefficient of performance (COP) values of about 0.8–1.0 are commonly predicted for this cycle (Kettleborough, 1980 and Lof *et al.*, 1988).

The Pennington cycle was improved by Munters in 1968 by introducing parallel passages (Pesaran *et al.*, 1992). Maclaine-Cross (1988) proposed a system called the Simplified Advanced Solid Desiccant cycle which gave COP values of above 2. Rush and Macriss (1969) and Macriss and Zawacki (1982) built a number of gas-fired and solar prototype systems. Solar energy prototypes were also built by private companies such as Ouseau, Airesearch, and Exxon in 1982 (Pesaran *et al.*, 1992). But this early attempt was discouraged by the lack of good analytical methods for the prediction of the performance of regenerative dehumidifiers. The methods available then were able to predict performance at rotational speeds so low that the whole matrix was in equilibrium with the process and regenerative air streams (Bullock and Threlkeld, 1966). Since then, a number of analytical methods have been developed in order to understand and analyze the performance of desiccant dehumidifiers and the desiccant cooling systems. Important among those were the 'Analogy Theory' by Banks (1972), the 'Finite Difference Method' by Maclaine-Cross (1988), the 'Pseudo-Steady State Model' by Barlow (1982), and Worek and Lavan's (1982) 'Finite Difference Method for Cross-Cooled Dehumidifiers.'

Based on the models of Maclaine-Cross (1988) and Barlow (1982), Collier developed a model in 1988, which is now widely used by other researchers (see Pesaran *et al.*, 1992). These models not only helped researchers to understand and analyse the factors and physical properties that influence the performance of desiccant cooling systems, but have also helped increase our understanding of new desiccant materials and cooling cycles.

2.2. *Hybrid desiccant cooling systems*

Moisture removal is achieved in a conventional vapour-compression system by condensation. In high-humidity regions like the Southeastern part of the United States, this method could be very inefficient since it usually involves reheating the air after dehumidification. Vapour compression systems are efficient in sensible cooling, whereas desiccant dehumidifiers are efficient in handling latent loads. Hybrid systems, which integrate desiccant dehumidifiers with conventional cooling systems are proven to provide substantial energy savings.

Maclaine-Cross (1988) found that energy costs were halved using his hybrid system. His system consisted of a regenerative dehumidifier, a heat exchanger, an evaporative cooler and heating coils and fans, to provide the latent and part of the sensible load, as well as a gas engine-driven chiller which is also used to take up the remaining sensible load. This also suggests that desiccant cooling systems may prove to be competitive with conventional systems when the desiccant units are commercially available.

Parsons *et al.* (1988) examined the merits of coupling a silica-gel desiccant dehumidification subsystem to a gas engine-driven vapour compression system. A desiccant enhanced air conditioner (DEAC) was patented

by Cromer (1988). This system is capable of improving the performance of a vapour compression air conditioner by adding a desiccant dehumidifier. In this air conditioner, a desiccant dehumidifier removes the moisture from the saturated cold air leaving the evaporator coil, thus lowering the humidity and increasing the temperature by adding the heat of sorption, thus eliminating the need for reheat.

A new idea of passive-solar cooling using a desiccant material was developed by Fairey *et al.* (1986) and Swami *et al.* (1990). This concept, called a Desiccant Enhanced Nocturnal Radiation (DESRAD), utilizes a desiccant bed integrated in the roof along with a conventional vapour compression system to achieve both latent and sensible cooling in hot and humid climates. In this system, sensible cooling is provided during the night and the desiccant is regenerated during the day by solar heat. This system, operates in two modes: the night adsorption mode and the daytime desorption mode. During the night, room air is circulated to the desiccant bed on the roof where the moisture is removed. The heat of sorption is transferred to the atmosphere. After the air passes through the desiccant bed, it goes through an evaporative cooler to increase the humidity level and to further cool the air. This air is then passed into the conditioned zone where it absorbs both the heat and moisture from the space. During the day, the desiccant system is disconnected from the house and the desiccant is regenerated by venting it to ambient air, which is heated by the sun-absorbing metal roof. The heat exchanged between the metal roof and the desiccant along with the hot air regenerates the bed, with a conventional air conditioning system being used as a back up. Swami *et al.* (1990) studied a building-integrated DESRAD system and found 79, 81, and 75% reductions in the total, sensible and latent loads, respectively.

2.3 Desiccant dehumidifier models

Much of the potential for further improvements in the performance of the desiccant cooling system lies in the dehumidifier. Detailed models of the heat and mass transfer processes that occur in a dehumidifier can be used to judge the potential benefits of various materials and matrix geometries. Optimization of performance in terms of flow-rate ratios, rotational speed, and the desiccant/matrix thermodynamic properties is also easily accomplished with these models.

MOSHMX, a computer program utilizing finite differences, developed by Maclaine-Cross (1988), was based on a detailed numerical analysis and extrapolation to a zero-grid size using four carefully chosen grid sizes. Jurinak *et al.* (1984) used MOSHMX to study the effect of isotherm shape, maximum water content, heat of adsorption, regenerative matrix thermal capacitance, matrix moisture diffusivity, and adsorption hysteresis on dehumidifier performance. MOSHMX has been used extensively to model dehumidifier operations like transient performance (Brandemuehl and Beckman, 1979), purging (Jurinak *et al.*, 1984), and desiccant property effects (Collier and Cohen, 1991 and Jurinak *et al.*, 1984). Finite difference techniques have been used by many researchers to obtain detailed models of dehumidifiers (Pla-Barby, 1978; Holmberg, 1979; Barker and Kettleborough, 1980; Mathiprakasam and Lavan, 1980; Pesaran and Mills, 1984). The program DESSIM was developed by Barlow (1982) where the matrix was discretized and each node was treated as a counterflow heat and mass exchanger in which both the mass and heat transfer were assumed to be uncoupled. Collier and Cohen (1991) developed a model based on the DESSIM program. This code, ET/DESSIM, incorporated several improvements over the DESSIM program. In another investigation, Collier added the approach of MOSHMX for solving the heat and mass transfer in a dehumidifier node to the ET/DESSIM program. This code, called DCSSMX, is more accurate than ET/DESSIM and is widely accepted and used for modeling solid rotary dehumidifiers. These codes (MOSHMX, DESSIM, AND DCSSMX) have been validated, with varying degrees of accuracy, by experimental data (Barlow, 1982; Schultz *et al.*, 1987). Recently, Kang and Maclaine-Cross (1989) modified MOSHMX to model the performance of a silica gel/zeolite, layered desiccant dehumidifier. A ventilation cycle using this dehumidifier performs slightly better than a cycle with a silica gel dehumidifier.

Van den Bulck *et al.* (1985a, b) solved the conservation equations for an equilibrium dehumidifier using wave analysis that includes the effects of 'shocks' (irreversible sharpening of the wave fronts). An analytical

expression for dehumidifier performance results in a case of either complete regeneration or complete saturation during adsorption. Intermediate cases were found by interpolation. The model employs property correlations for a nominal silica gel dehumidifier.

2.4. System models

GARD of Chamberlain National in conjunction with the Lawrence Berkeley Laboratory and under funding from GRI (Gas Research Institute), incorporated models of desiccant cooling systems in the computer code 'DOE 2.1' building load and HVAC simulation program. In a similar effort, S.E.I. Associates of Boudler, Colorado, incorporated several desiccant dehumidification systems in the Trane's TRACE (TRane Air Conditioning Economics) program.

Desiccant cooling system models can be developed by appropriately connecting the various component models together. Simulations can then be run by driving the system with time-dependent forcing functions such as weather data. The program TRNSYS (Klein *et al.*, 1991) provides a very flexible structure for doing that. TRNSYS contains models for solar collection/storage systems and building cooling loads along with routines for reading weather data and summarizing results. The modular nature of TRNSYS makes it convenient to consider a variety of component arrangements or replacement of one component model with another. The hour-by-hour nature of the simulation allows investigations of control strategies for responding to changing conditions and of the potential impact on the power utilities.

Jurinak *et al.* (1984) used TRNSYS to simulate the performance of a solar-assisted ventilation cycle for residential applications. Howe (1983) used TRNSYS as a simulation tool to model a desiccant/vapour compression hybrid system for a commercial building. He used the simplified analogy method to model the dehumidifier. Burns (1985) simulated the hybrid system for a Jewel supermarket in Chicago. Smith and Hwang (1992) simulated a desiccant cooling system as a proof-of-concept model to show that the dehumidifier wheel model they developed could be used in TRNSYS. Both Burns (1985) and Smith and Hwang (1992) developed the dehumidifier model using Van den Bulck's effectiveness correlations (Van den Bulck, *et al.*, 1985a, b) and Bank's simple analogy method (1972).

The present work uses the dehumidifier model and the proof-of-concept desiccant cooling system TRNSYS subroutine developed by Smith and Hwang (1992) to simulate and optimize a solar-assisted air conditioning system with a backup cooling coil and then conduct an economic analysis of the simulated system. The main thrust is to optimize the system based on thermal performance according to varying design parameters. Economic analysis (discussed in Part II of this work) should yield parameters such as life cycle savings, life cycle costs and pay back periods as functions of market discount rate, fuel cost, fuel inflation rate and the general inflation rate for a series of solar subsystem sizes.

3. SYSTEM DESIGN

3.1. Solar collector storage subsystem

This subsystem provides the energy required for the regeneration of the desiccant. Figure 3 shows a schematic of the system component configuration chosen. The system consists of an array of solar collectors with a 50% propylene glycol/water solution as the working fluid, which transfers energy to water via a heat exchanger. This hot water is stored in a storage tank. A natural gas-fired auxiliary heater is provided in series with the storage tank. The auxiliary heater aids in supplying the hot water at the required present regeneration temperature, before it flows to the air heating coil in the desiccant air conditioner.

3.1.1. Collector. The collector liquid (working fluid) is a 50% propylene glycol/water solution. Area of collector array is chosen to be 50 m². Recommended flow rates of the working fluid and water loops are 0.01–0.02 kg s⁻¹ m⁻² of collector area (Duffie and Beckman, 1991). Hence an average value of 0.015 kg s⁻¹ m⁻²

is chosen. This gives a flow rate of 0.75 kg s^{-1} or 2700 kg h^{-1} . Also, conventional design of a storage tank volume requires about $50\text{--}100 \text{ liters m}^{-2}$ of collector area (Duffie and Beckman, 1991). Hence a mean value of 75 liters ($0.075 \text{ m}^3 \text{ m}^{-2}$ of collector area) is chosen. This suggests a storage tank volume of 3.75 m^3 . Performance of any solar flat plate collector may be established by two quantities which can be obtained by tests conducted according to ASHRAE Standard 93-77.

Hottel–Whillier's equation gives a general expression for collector efficiency (Klein *et al.*, 1991)

$$\eta_c = \frac{Q_u}{A_c - I_T} = F_R (T\alpha) - F_R (T_i - T_o) \quad (1)$$

where, η_c is the collector efficiency, Q_u rate of useful energy gain by the collector, A_c collector surface area, I_T total incident energy per unit collector area F_R , overall collector heat removal efficiency factor, τ transmittance of collector covering, α collector absorptance, T_a ambient temperature, T_i inlet temperature of fluid of collector, and U_L overall loss coefficient of collector per unit aperture area.

Q_u can be expressed considering the collector energy balance

$$Q_u = F_R A_c [I_T (\tau\alpha)] - F_R U_L A_c (T_i - T_a) \quad (2)$$

The quantity $I_T (\tau\alpha)$ is the energy absorbed by the collector and the factor $(\tau\alpha)$ is often called the transmittance–absorbance product or factor. Equation (2) is the basis for collector tests prescribed in ASHRAE Standard 93-77 (Duffie and Beckman, 1991). Collector tests are then performed and presented as straight line plots of η versus $(T_i - T_a)/I_T$ with intercept of $F_R (\tau\alpha)$ and negative of slope $(F_R U_L)$ [Refer to equation (1)].

For the given collector, the best line through these plotted points indicates its performance under varying operating conditions. Hence two quantities are established and defined. The first is the intercept $F_R (\tau\alpha)$ and the second is the negative of the slope $(F_R U_L)$. For the collector used in this subsystem, $F_R (\tau\alpha) = 0.777$ and $(F_R U_L) = 15.8424 \text{ W m}^{-2}$.

3.1.2. Heat exchanger. The heat exchanger used is of the parallel flow mode constant effectiveness type. For this, the maximum possible heat transfer is calculated based on the minimum capacity rate fluid and the cold and hot side fluid inlet temperatures.

3.1.3. Storage tank. The storage tank used is of the fully mixed (no stratification) type. The load flow enters at the bottom of the tank and the hot source stream enters at the top of the tank. The tank is well insulated with a U_{TANK} value of $1.5 \text{ kW m}^{-2} \text{ K}^{-1}$ and a ratio of tank height to diameter of 1.5. The storage tank volume depends on the collector area (Duffie and Beckman, 1991).

3.1.4. Pumps and relief valve. Pumps were modelled to operate at a constant flow rate. The hot side pump operates only when commanded by the pump controller. The controller commands the pump to operate when the temperature of the fluid returning from the storage tank falls below the desired temperature. A relief valve is provided to dump energy from the fluid stream in the pipes whenever the fluid temperature exceeds 100°C . It was assumed that the mass loss is negligible when the valve is open and the outlet and inlet flow rates are equal.

3.2. Analysis of the conditioned zone

To evaluate the performance of this simulated cooling system, it is first necessary to simulate the thermal response information required for providing the loads. For this purpose a single-story, single family, detached house was chosen (ASHRAE, 1993). Figure 4 shows the plan of the selected zone. The following paragraphs explain the construction details for this residential zone. Four locations, Jacksonville (Florida), Albuquerque (New Mexico), New York City, and Houston (Texas), were chosen to be the representative locations for their respective regions since they have varying climates, thus demanding varying summer

cooling requirements. Since the same standard zone is used in all four locations, simulation results should give a comparative indication regarding the performance of the cooling system.

3.2.1. Construction details. Figure 4 shows the construction details along with the dimensions of the residential zone to be analyzed. As can be seen, the building length is 22.6 m, height is 2.45 m, and width is 11 m. The building orientation is such that the garage door is facing West. The building construction details are as follows:

3.2.1.1. Roof construction. Conventional roof–attic–ceiling combination, vented to remove moisture with 150 mm (6") of fibrous batt insulation and vapour retardant. The window has a 0.6 m (2 ft) overhang at the top.

3.2.1.2. Wall construction. Frame with 100 mm (4") face brick and 90 mm (3.5") of fibrous batt insulation. The interior walls consist of a 13 mm (0.5") thick dry wall.

3.2.1.3. Floor construction. Constructed using a 100 mm (4") thick slab on grade.

3.2.1.4. Windows. Table 1 below gives the dimensions and orientations of the various windows present throughout the house.

3.2.1.5. Fenestration. Clear double glass, 3 mm thick in and out, with closed, medium coloured venetian blinds.

3.2.2. Occupancy. There are four occupants that are assigned to the living room for calculation purposes based on two for the master bedroom and one each for the remaining bedrooms.

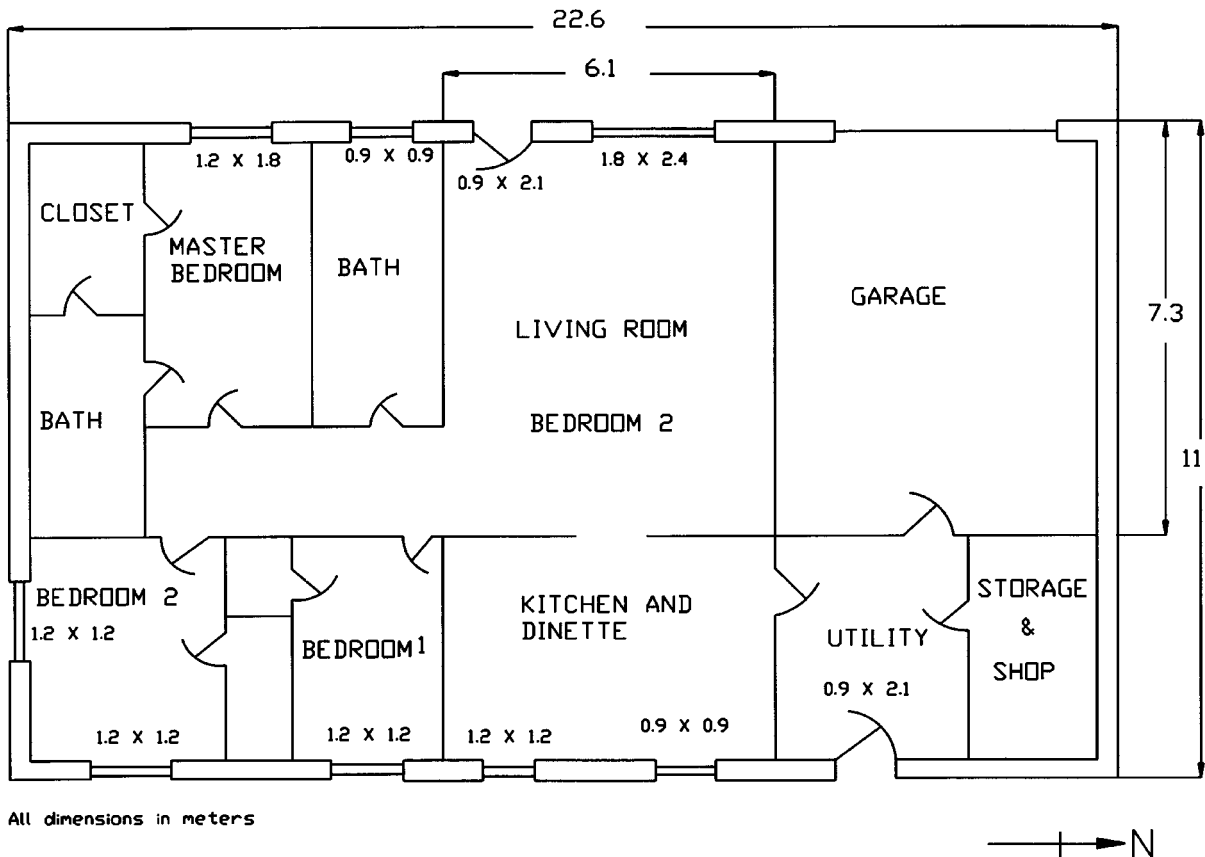


Figure 4. Plan of the detached residential building

3.2.3. *Appliances and lights.* Appliances make up 353 W for kitchen, 235 W for the living room and 353 W for the utility and storage rooms.

The cooling load temperature difference (CLTD) method used for performing the design residential cooling load calculations requires that the load for each room be calculated first and then added to obtain the total cooling load. Table 2 shows the ambient design conditions that were used for the design cooling load calculations.

3.3. Cooling load calculations

As mentioned above, the CLTD method has been used throughout this study to calculate the cooling load. Calculations were performed for all locations selected employing the construction details described earlier. All assumed data were based on information found in ASHRAE (1993). A cooling load summary for a detached house in Jacksonville is provided in Table 3. Additional details regarding the calculation procedures employed can be found in Davanagere (1994).

The total sensible cooling load is thus 5.56 kW. Assuming 10% of the load to account for duct losses and computing an outdoor infiltration load of 0.45 kW (Davanagere, 1994), a total load of 6.56 kW is obtained. A latent factor (LF) of 1.3 matches the performance for typical residential vapour compression cooling systems. A coil designed with a given latent capacity generally transfers more sensible heat in the absence of moisture in the air, but not in the same amount as the design latent heat transfer capacity (ASHRAE, 1993). Hence the total cooling load is 8.54 kW. To insure that the unit is large enough, a nominal unit with a size of 10 kW capacity is selected (the unit is oversized less than about 25%). The following table presents a summary of the cooling loads calculated for the four locations using the CLTD method.

It is now clearly evident why a 10 kW unit was selected in the design process. This unit is slightly oversized for the maximum load to be handled by the cooling system (Jacksonville). The same unit was considered in all four locations in order to make all the simulations perform consistently.

Table 1. Window dimensions and orientations

Room	No. of windows	Dimensions	Orientation
1. Living room	1	1.8 m × 2.4 m	West
2. Kitchen & dinette	2	1. 1.2 m × 1.2 m 2. 0.6 m × 0.9 m	East
3. Bedroom 1	1	1.2 m × 1.2 m	East
4. Bedroom 2	2	1. 1.2 m × 1.2 m 2. 1.2 m × 1.2 m	East
5. Master bedroom		1.2 m × 1.8 m	West
6. Bathroom	1	0.6 m × 0.9 m	West

Table 2. Outdoor climatic conditions for the four locations

	Jacksonville	Albuquerque	New York City	Houston
Latitude (°)	30.5	35.05	40.75	29.9
Longitude (°)	81.7	106.6	73.9	95.3
Outdoor design dry bulb temp. (°C)	36	36	33	34
Outdoor design wet bulb temp. (°C)	25	16	23	26
Mean daily range (°C)	11	15	9	12
Median of annual extremes (°C)	36.4–3.7	36.7–14.9	34.9–15.7	38.9–1.7

System simulations depend upon the hourly cooling loads which are in turn dependent on the wall, door, roof, window and floor constructions. Hence all construction details were carefully evaluated and the relative locations to the doors and windows were incorporated into the simulations, as well as window sizes, types, and orientations. Cooling loads due to air infiltration were calculated based on the recommended practices as described in ASHRAE (1993).

3.4. Desiccant cycle design

The design of the air conditioning system is controlled by many operating conditions. Referring to Figures 1 and 2, the following parameters may be used as a basis for designing the system: ambient conditions, inside (room) conditions, regeneration air temperature before the dehumidifier, supply and return air flow rates, and design sensible and latent cooling loads.

3.4.1. Ambient conditions. Ambient conditions are based on those of Jacksonville (highest values of dry and wet-bulb conditions during a year, among the four locations, are chosen to demand the best out of the air-conditioner). These values are: $T_1(\text{db}) = 36^\circ\text{C}$ and $T_1(\text{wb}) = 25^\circ\text{C}$ (ASHRAE, 1993).

3.4.2. Inside (room) conditions. Recommended standard design conditions for a residential air conditioner are $T_6(\text{db}) = 26.67^\circ\text{C}$ and $T_6(\text{wb}) = 19.94^\circ\text{C}$ (ASHRAE, 1993).

3.4.3. Regeneration air temperature. The air coming out of the conditioned space undergoes different processes of cooling and then heating depending upon the cooling and heating requirements of the process air. Hence, fixing the final state of air for the regeneration process would give a better picture of the influence of this constraint on the cycle. In this study, this was fixed at 80°C . This temperature is also proven acceptable

Table 3. Summary of cooling load calculations for a detached house in Jacksonville

Room	Walls, doors, roof (kW)	Glass (kW)	People (kW)	Appli- ances (kW)	Air Infl. (kW)	Total (kW)
Living room	0.4574	0.4806	0.268	0.235	0.2049	1.06461
Kitchen & dinette	0.2204	0.2181	—	0.353	0.1039	0.8954
Utility & storage	0.3865	—	—	0.353	0.1243	0.8639
Master bedroom & bath	0.4994	0.2173	—	—	0.2385	0.9552
Large bath and hallway	0.1593	0.073	—	—	0.0841	0.3165
Bedroom 1	0.1673	0.145	—	—	0.0785	0.3908
Bedroom 2	0.1906	0.2218	—	—	0.0785	0.4909
Total	2.081	1.3558	0.268	0.941	0.9127	5.5588

Table 4. Residential cooling loads for the four locations

	Jacksonville	Albuquerque	New York City	Houston
Total cooling load	8.53	6.62	6.5	7.5
Sensible load	6.56	6.62	5.90	5.77
Latent load	1.97	0.00	0.61	1.73

according to the studies performed on a rotary dehumidifier (Jurinak *et al.*, 1984; Nelson and Beckman, 1978).

3.4.4. Supply and return air flow rates. One of the assumptions made in the modelling of the dehumidifier is that regeneration and process air streams having same flow rates influence the designer to assume the same for the supply and return air flow rates. This assumption was observed to produce a satisfactory performance for the system. For this cycle, a conventional flow rate of about 450 cfm ton⁻¹ is assumed (ASHRAE, 1993). Hence supply and return volumetric flow rates are chosen at 1350 cfm.

3.4.5. Sensible and latent cooling loads. Based on the building cooling load calculations, a unit rated at 10 kW (2.84 tons) is considered. Normally 77% of the rated capacity is for sensible cooling while 23% is for latent cooling (ASHRAE, 1993). Based on this, the design sensible cooling capacity was selected as 7.7 kW, while that for the latent cooling capacity was selected as 2.3 kW.

4. PSYCHROMETRIC ANALYSIS

State Point 1 represents the ambient conditions for Jacksonville, Florida, which are, $T_1 = 36^\circ\text{C}$ (db) and 25°C (wb) with a corresponding humidity ratio of $W_1 = 0.01543 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$ (ASHRAE, 1993). State Point 5 represents the inside conditions for Jacksonville, Florida, which are, $T_5 = 26.27^\circ\text{C}$ (db) and 19.4°C (wb) with a corresponding humidity ratio of $W_5 = 0.01183 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$ (ASHRAE, 1993). Since the design loads are 10 kW (2.85 tons) for the total cooling load, 7.7 kW (2.19 tons) for the sensible load, and 2.3 kW (0.65 tons) for the latent load, and the recommended volumetric flow rate is $0.743 \text{ m}^3 \text{ s}^{-1} \text{ kW}^{-1}$ (450 cfm ton⁻¹) according to ASHRAE (1993), the supply and return volumetric flow rate was chosen at $7.43 \text{ m}^3 \text{ s}^{-1}$. Hence, the mass flow rate of air is $m_{\text{air}} = 0.715 \text{ kg s}^{-1}$ from continuity considerations.

In order to determine State Point 4, the following calculations are performed: $0.715 (57 - h_5) = 2.3$, hence $h_5 = 53.78 \text{ kJ kg}^{-1}$, and $0.715 (53.78 - h_4) = 7.7$, hence $h_4 = 43.01 \text{ kJ kg}^{-1}$. From the psychrometric chart, $T_4 = 17^\circ\text{C}$ and, $W_4 = 0.0105 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$. State Point 6 can be determined by using the effectiveness equation for evaporative coolers. A 95% evaporative effectiveness is assumed for the return side evaporative cooler; i.e. $[(T_5 - T_6)/(T_5 - T_5(\text{wb}))] = 0.95$, from which $T_6 = 20.28^\circ\text{C}$.

Process 5–6 is a constant enthalpy process. Hence, from the psychrometric chart, State Point 6 can be plotted along the constant enthalpy line of State Point 5. So, from the psychrometric chart, $W_6 = 0.0140 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$. State Point 8 is the state at which the hot regeneration air enters the desiccant wheel. The temperature at this point is fixed at 90°C (Smith and Hwang, 1992). Referring to the psychrometric chart, the corresponding humidity ratio, $W_8 = W_6 = W_7 = 0.014 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$. For the desiccant wheel, the following assumptions were made (Smith and Hwang, 1992): enthalpy effectiveness, $\Sigma_h = 100\%$, humidity effectiveness, $\Sigma_w = 100\%$, matrix specific heat, $C_m = 0.921 \text{ kJ kg}^{-1} \text{ }^\circ\text{C}$, mass of silica gel, $M_d = 3.2 \text{ kg}$.

The above values were used to obtain the supply side and regeneration side dehumidifier outlet states (States 2 and 9, respectively). State Point 2 (supply air outlet state, i.e. after desiccant wheel, process side) is determined as $T_2 = 62.2^\circ\text{C}$; $W_2 = 0.00655 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$. State Point 9 (return outlet state, i.e. after desiccant wheel, regeneration side) is also determined as $T_9 = 53.9^\circ\text{C}$; $W_9 = 0.02338 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$.

State Point 3 (after the regeneration wheel, process side) is determined assuming a regenerator effectiveness of 90%, $\Sigma_{\text{regen}} = 0.90 = (T_3 - T_2)/(T_2 - T_6)$, thus $T_3 = 24.47^\circ\text{C}$. Since $W_3 = W_2$, thus $W_3 = 0.00655 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$. The corresponding wet bulb temperature (from the psychrometric chart), $T_3(\text{wb}) = 14.8^\circ\text{C}$. Now that State Points 3 and 4 have been determined, the effectiveness of the process side evaporative cooler is determined and hence we can check how realistic the regenerator effectiveness used is. Thus, $\Sigma_{\text{pevap}} = [(T_3 - T_4)/(T_3 - T_3(\text{wb}))]$, which gives an effectiveness of $\Sigma_{\text{pevap}} = 86\%$, which is practically attainable. Therefore, the assumed regenerator effectiveness is a realistic value.

State Point 7 (after the regeneration wheel, regeneration side) is again determined using a 90% effectiveness for the regeneration wheel, $\Sigma_{\text{regen}} = (T_7 - T_6)/(T_2 - T_6)$ thus $T_7 = 58^\circ\text{C}$. Also, $W_7 = W_6 = 0.01462 \text{ kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$, and $h_7 = 96.14 \text{ kJ kg}^{-1}$. Now that all nine state points are known, the performance can be evaluated in terms of the required heat input, $Q_{\text{heat}} = m_{\text{air}} (h_8 - h_7) = 15 \text{ kW}$, the coefficient of performance, $\text{COP} = 0.66$, amount of water required by the process side evaporative cooler, $m_{\text{pw}} = m_{\text{air}} (W_4 - W_3) = 0.002924 \text{ kg s}^{-1}$ and amount of water required by the regeneration side evaporative cooler, $m_{\text{rw}} = m_{\text{air}} (W_6 - W_5) = 0.001994 \text{ kg s}^{-1}$.

5. CONCLUSIONS

A solid desiccant cooling system with a backup vapour compression system is analysed in four cities in the United States. This paper describes the relevant psychrometric calculations and analyses of the conditioned zone required for simulating the transient performance of the system. In the second half of this work, the thermal performance of the system is simulated and its economic feasibility is assessed.

NOMENCLATURE

Latin symbols

A_c	collector surface area, m^2
C_m	specific heat of desiccant matrix, $\text{kJ kg}^{-1} \text{ }^\circ\text{C}^{-1}$
C_p	specific heat of water, $\text{kJ kg}^{-1} \text{ }^\circ\text{C}^{-1}$
COP	coefficient of performance
F_R	overall collector heat removal efficiency factor, dimensionless
h	enthalpy, kJ kg^{-1}
I_T	total incident energy per unit collector area, kJ m^{-2}
M_d	mass of silica gel in the dehumidifier, kg
m	mass flow rate, kg h^{-1} or kg s^{-1}
Q_{ac}	total energy requirement for the desiccant air conditioner, kJ
Q_L	latent cooling load, kJ
Q_S	sensible cooling load, kJ
Q_{sol}	total energy supplied by the solar collectors, kJ
Q_T	total cooling load, kJ
Q_u	rate of useful energy gain by the collector, $\text{kJ (h m}^2\text{)}^{-1}$
SF	solar fraction, dimensionless
T	temperature, $^\circ\text{C}$
T_{db}	dry-bulb temperature, $^\circ\text{C}$
T_{wb}	wet-bulb temperature, $^\circ\text{C}$
U_L	overall loss coefficient of collector per unit aperture area
W	humidity ratio, $\text{kg}_{\text{water}} \text{ kg}_{\text{dry air}}^{-1}$

Greek symbols

α	collector absorptance
η_c	collector efficiency, %
τ	transmittance of collector covering

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REFERENCES

- Altenkirch, E. (1941). 'Separating and cooling apparatus', *US Patent 2,233,189*.
- Altenkirch, E. (1944). 'Air conditioning', *US Patent 2,344,384*.
- ASHRAE (1993). *ASHRAE Handbook-1993 Fundamentals*, The American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, Georgia
- Banks, P. J. (1972). 'Coupled equilibrium heat and single adsorbate transfer in fluid flow through porous media'—I, characteristic potentials and specific capacity ratios', *Chem. Engng. Sci.*, **27**, 1143–1155.
- Barker, J. M. and Kettleborough, C. F. (1980). 'The adiabatic adsorption-desorption characteristics of silica gel beds—Part I', *ASME Winter Annual meeting*, ASME Paper 80-WA/Sol-31, November.
- Barlow, R. (1982). Analysis of the adsorption process and of desiccant cooling systems: A pseudo-steady-state model for coupled heat and mass transfer', *Technical Report No. SERI/TR-631-1330*, Solar Energy Research Institute., Golden, CO.
- Brandemuehl, M. J. and Beckman, W. A. (1979). 'Economic evaluation and optimization of solar heating systems', *Solar Energy*, **23**(1), 1–10.
- Bullock, C. E. and Threlkeld, J. L. (1996). 'Dehumidification of moist air by adiabatic adsorption', *ASHRAE Trans.*, **72**, 301–313.
- Burns, P. R. (1985). 'An analysis of hybrid desiccant cooling systems in supermarket applications', M.S. Thesis, University of Wisconsin, Madison, WI.
- Collier, R. K. and Cohen, B. M. (1991). 'An analytical examination of methods for improving the performance of desiccant cooling systems', *J. Solar Energy Engng.* **113**, 157–163.
- Cromer, C. J. (1998). 'Cooling system,' *US Patent 34,719,761*.
- Davanagere, B. S. (1994). 'Transient simulation and economics of a solar assisted desiccant air conditioning system,' M.S. Thesis, University of Florida, Gainesville, Florida.
- Duffie, J. and Beckman, W. (1991). *Solar Engineering of Thermal Process*, Wiley, New York, 1980.
- Fairey, P., Kerestecioglu, A. and Viera, R. (1986). 'Analytical investigation of the desiccant enhanced nocturnal radiation cooling concepts', *Final Technical Report No. FSEC-CR-152-86*, Florida Solar Energy Center, Cape Canaveral, Florida.
- Fleisher, W. L. (1939). 'Adsorption system of air conditioning', *US Patent 2,147,248*.
- Holmberg, R. B. (1979). 'Combined heat and mass transfer in regenerators with hygroscopic materials', *J. Heat Transfer*, **101**, 205–210.
- Howe, R. R. (1983). 'Model and performance characteristics of a commercially-sized hybrid air conditioning system which utilizes a rotary desiccant dehumidifier', Ph.D. Dissertation, University of Wisconsin, Madison, WI.
- Jurinak, J. J., Mitchell, J. W. and Beckman, W. A. (1984). 'Open cycle desiccant air conditioning as an alternative to vapour compression cooling in residential applications', *J. Solar Energy Engng.* **106**(3), 252–260.
- Kang, T. S. and Maclaine-Cross, I. L. (1989). 'High performance solid desiccant open cooling cycles', *J. Solar Energy Engng.* **111**, 176–183.
- Kettleborough, C. F. (1980). 'Solar assisted desiccant dehumidification and cooling with an indirect evaporative cooler', *ASME Winter Annual Meeting*, ASME Paper 80 WA/Sol-29.
- Klein et al. (1991). 'A transient system simulation program, version 13.1', *Engineering Experiment Station Report*, University of Wisconsin-Madison, Solar Energy Laboratory, Madison, Wisconsin.
- Larriva, G. A. (1941). Air conditioning system and apparatus therefore, *US Patent 4,612,019*.
- Lof, G. O., Cler, G. and Brisbane, T. (1988). 'Performance of a solar desiccant cooling system', *J. Solar Energy Engng.* **111**, 165–171.
- Maclaine-Cross, I. L. (1988). 'Proposal for a desiccant air conditioning system,' *ASHRAE Trans.* **94**(2), 1997–2009.
- Macriss, R. A. and Zawacki, T. S. (1982). 'High COP rotating wheel solid desiccant system', *Proc. Ninth Energy Technology Conf.* Washington, DC, 16 Feb 1982, Institute of Gas Technology, Document Center, Chicago, Illinois.
- Mathiprakasham, B. and Lavan, Z. (1980). 'Performance predictions for adiabatic desiccant dehumidifiers using linear solutions', *J. Solar Energy Engng.* **102**(1), 73–79.
- Miller, E. B. and Fonda, B. P. (1993). 'Rotary adsorber unit', *US Patent 1,912,784*.
- Nelson, J. S. and Beckman, W. A. (1978). 'Simulations of the performance of open cycle desiccant systems using solar energy', *Solar Energy*, **21**, 273–278.
- Parsons, B. K., Pesaran, A., Bharatan, D. and Shelpuk, B. (1988). 'Improving gas-fired heat pump capacity and performance by adding a desiccant dehumidification subsystem', *ASHRAE Winter Meeting*, Chicago, IL, 28 Jan – 1 Feb 1989, pp. 25, SERI/TP-254-3402 CONF-9890105-2.
- Pesaran, A. A. and Mills, A. F. (1984). 'Modelling of solid-side mass transfer in desiccant particle beds', *Solar Engineering—1984*, ASME, New York, pp. 177–185.
- Pesaran, A. A., Penny, T. R. and Czanderna, A. W. (1992). 'Desiccant cooling: state-of-the-art assessment,' *Technical Report No. NREL/TP-254-4147*, National Renewable Energy Laboratory, Golden, CO.
- Pla-Barby, E. (1978). 'Solid desiccant air conditioning with silica gel using solar energy', Ph.D. Dissertation, University of Texas, Austin, Texas.
- Rush, W. F. and Macriss, R. A. (1969). 'Munters environmental control system', *Appliance Engineer*, March 23–28.
- Schultz, K., Barlow, R., Pesaran, A. A. and Kreith, F. (1987). 'An analysis of a direct radiation solar dehumidification system', *J. Solar Energy Engng.* **109**, 15–21.

- Shipman, B. C. (1936). 'Air cooling and conditioning apparatus and system', *US Patent 2,058,042*.
- Smith, R. R. and Hwang, C. (1992). 'Modelling of a solar-assisted desiccant air conditioner for a residential building', *Thermodynamics and the Design, Analysis, and Improvement of Energy Systems*, **266**, 409–418.
- Swami, M., Fairey, P. and Kerestecioglu, A. (1990). 'An analytical assessment of the desiccant enhanced radiative cooling concept', *Solar Engineering—1990*, ASME, New York, pp. 397–406.
- Van den Bulck, E., Mitchell, J. W. and Klein, S. A. (1985a). 'Design theory of rotary heat and mass exchangers—I: wave analysis of rotary heat and mass exchangers with infinite transfer coefficients', *Int. J. Heat and Mass Transfer*, **28** (8).
- Van den Bulck, E., Mitchell, J. W. and Klein, S. A. (1985b). 'Design theory of rotary heat and mass exchangers—II: effectiveness-number of transfer units method for rotary heat and mass exchangers', *Int. J. Heat and Mass Transfer*, **28** (8), 1587–1595.
- Worek, W. M. and Lavan, Z. (1982). 'Performance of a cross-cooled desiccant dehumidifier prototype', *J. Solar Energy Engineering*, **104** (3), 187–196.