1 of 1

Contract Report

Open Cycle Liquid Desiccant Dehumidifier and Hybrid Solar/Electric Absorption Refrigeration System

ANNUAL REPORT

FSEC-CR-713-94

Calendar Year 1993

Prepared for:
U.S. Department of Energy
Golden Field Office
1617 Cole Boulevard
Golden, Colorado 80401

Prepared by:
Bruce G. Nimmo, Ph.D.
Mark D. Thornbloom

Florida Solar Energy Center
300 State Road 401
Cape Canaveral, FL 32920

Open Cycle Liquid Desiccant Dehumidifier and Hybrid Solar/Electric Absorption Refrigeration System

**ANNUAL REPORT
for CY 1993**

By

**Bruce G. Nimmo
Mark D. Thornbloom
Florida Solar Energy Center
300 State Road 401
Cape Canaveral, Florida 32920**

USDOE Contract No. DE-FC02-86SF16305, Sub. B

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

**Prepared for
U.S. Department of Energy
Golden Field Office
1617 Cole Boulevard
Golden, Colorado 80401**

MASTER

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

TABLE OF CONTENTS

EXECUTIVE SUMMARY	ii
TASK 1: Open Cycle Liquid Desiccant Dehumidifier	
1.1. Introduction	1
1.1.1. The Need for Dehumidification Technology	1
1.1.2. Dehumidification Using Liquid Desiccants	1
1.1.2.1. The Absorption Dehumidifier	1
1.1.2.2. The Collector/ Regenerator	3
1.1.3. The FSEC Open Cycle Liquid Desiccant Dehumidifier Project	3
1.1.4. Task Objectives	3
1.2. Literature Review	4
1.2.1. Introduction	4
1.2.2. Unglazed Collector/Regenerators	5
1.2.2.1. Analytical Efforts	5
1.2.2.2. Experimental Efforts	5
1.2.3. Glazed Collector/Regenerators	5
1.2.3.1. Analytical Efforts	5
1.2.3.2. Experimental Efforts	6
1.2.4. Comparison of Glazed and Unglazed Collector/Regenerators	7
1.2.4.1. Analytical Efforts	7
1.2.4.2. Experimental Efforts	8
1.3. Collector/Regenerator Test Bed	9
1.3.1. Collector/Regenerator Research Needs	9
1.3.2. Parameters To Be Tested	10
1.3.2.1. Ambient Conditions	10
1.3.2.2. Collector/Regenerator Length	11
1.3.2.3. Collector/Regenerator Gap Height	11
1.3.2.4. Solution Flow Rate	12
1.3.2.5. Air Flow Rate	12
1.4. Performance Analysis	12
1.5. Bibliography	13
TASK 2: Hybrid Solar/Electric Absorption Refrigeration System	
2.1. Introduction	16
2.1.1. The Need for Low Temperature Absorption Refrigeration	16
2.1.2. The FSEC Hybrid Solar/Electric Absorption System	17
2.1.3. Task Objectives	17
2.2. Literature Review	19
2.3. Performance Analysis	21
2.3.1. Analysis Methodology	21
2.3.2. Results of Performance Analysis	26
2.4. Compressor Design	27
2.5. Bibliography	31
Appendix: Performance Analysis of Open Cycle Liquid Desiccant Dehumidifier	34

EXECUTIVE SUMMARY

This annual report presents work performed during calendar year 1993 by the Florida Solar Energy Center under contract to the U.S. Department of Energy. Two distinctively different solar powered indoor climate control systems were analyzed: the open cycle liquid desiccant dehumidifier, and an improved efficiency absorption system which may be fired by flat plate solar collectors. Both tasks represent new directions relative to prior FSEC research in Solar Cooling and Dehumidification.

Task 1: Open Cycle Liquid Desiccant Dehumidifier

Dehumidification is a significant part of the air conditioning load in humid climates such as in Florida. The Florida Solar Energy Center plans to develop and field test a fully operational desiccant dehumidifier system, using solar energy to regenerate the liquid desiccant. Task 1 is part of that effort. All materials required to construct the system are commercially available. Most auxiliary components, such as the absorption dehumidifier, pumps, and tanks, are also commercially available. The collector/ regenerator is built incorporating conventional roofing construction techniques. The collector/ regenerator regenerates the desiccant solution by using solar energy to heat the solution. Although this type of system has had intensive study, additional work is required to optimize performance.

An extensive literature search of open cycle collector/ regenerator systems was conducted. Based on this search, several parameters were identified which require further study for optimization. These parameters include collector/ regenerator length, glazing gap height, solution flow rate, and air flow rate. Ranges of testing for each parameter were identified. Calcium chloride was selected as the preferred desiccant, primarily due to its low cost. Design of a collector/ regenerator test bed has been started to allow testing of these parameters. Design and construction are in progress. The test bed will have three identical collector/ regenerator units to allow simultaneous testing of three different parameter levels. This will significantly reduce the impact on test results of uncontrollable variables such as ambient conditions. In an effort to determine the lowest possible solution flow rate while maintaining full wetting, the collector/ regenerator tilt will be varied in the initial test sequence.

The results of the test bed research will be used to develop a lab scale Open Cycle Liquid Desiccant Dehumidification System. The lab scale system will provide the operating experience necessary to design, build, and operate a commercial scale system.

Task 2: Hybrid Solar/Electric Absorption Refrigeration System

Flat plate solar space heating systems often are idle during much of the summer. During this period, conventional electric vapor compression systems must be used to meet the summer cooling load. Using solar heat to fire an absorption refrigeration system would increase the annual availability of the solar heating system. However, flat plate collectors produce heat at low temperature relative to the conventional firing temperatures for absorption systems. Cooling

capacity of absorption systems drops off significantly with reduction of the generator firing temperature. This task describes and analyzes a design modification of the absorption refrigeration system which adds a refrigerant vapor compressor between the evaporator and the absorber. With the addition of a relatively small amount of electricity, the hybrid system will allow absorption refrigeration systems to operate at full capacity and coefficient of performance, in spite of low generator temperatures. As a result, this will allow the use of standard flat plate collectors for firing the absorption system during the cooling season.

An extensive literature search has been completed. A simple performance analysis of a lithium bromide/water absorption system with and without the compressor installed has been conducted. The analysis showed that, with the addition of only about 200 watts to the compressor, the increase in refrigeration is from 0.81 kW (0.23 ton) to 3.52 kW (1 ton), in spite of a lowered generator temperature. These results indicate that the compressor addition has significant potential for improving solar-fired absorption performance. A technical paper describing this concept was presented at the ASME International Solar Conference in San Francisco, California, on March 27-30, 1994.

Task 1: Open Cycle Liquid Desiccant Dehumidifier

1.1. Introduction

1.1.1. The Need for Dehumidification Technology

Substantial portions of the United States have hot and humid climates. In commercial practice, vapor compression systems are typically used to meet both the sensible and the latent fraction of the air conditioning load. To remove humidity from the air stream, a vapor compression system sensibly cools the air past the dew point to condense the moisture out, then reheats the cool saturated air stream up to the delivery temperature and relative humidity for delivery to the conditioned space. In arid climates where latent loads are low, this dehumidification method is acceptable. However, it is inefficient in humid climates.

One means of improving the system efficiency is to remove the moisture from ventilation and space return air before the air stream is cooled to the delivery temperature. This drying of the air may be accomplished with desiccant materials. Solar-regenerated desiccant dehumidification has significant advantages over conventional vapor compression dehumidification. Since the moisture is removed from the air stream prior to cooling, reheat is unnecessary and the total load on the air conditioner can be reduced. In addition, the vapor compression system may be designed to be more efficient if it does not need to cool to the low temperatures required for dehumidification. Benefits of liquid desiccants over solid desiccants include isothermal dehumidification (improving efficiency), and regeneration at a location remote from the dehumidification location (allowing design flexibility). If solar energy is used to regenerate the desiccant, the energy cost of conditioning the air stream is reduced further. Availability of solar energy also closely follows the air conditioning demand curve. Desiccant systems also allow separate control of latent and sensible loads, which can be useful in industrial and commercial applications.

1.1.2. Dehumidification Using Liquid Desiccants

1.1.2.1. The Absorption Dehumidifier

Open cycle desiccant dehumidification may be accomplished as shown in Figure 1.1. Outdoor ventilation air for a building is blown through the absorption dehumidifier. Strong desiccant solution is pumped from the storage tank and sprayed into the air stream, absorbing humidity from the air. Heats of condensation and absorption are carried away by a conventional cooling tower. The dehumidified ventilation air then passes through a demister to remove any entrained desiccant, and is cooled by a conventional or high sensible heat ratio/ high energy efficiency ratio vapor compression system for delivery to the building. The weak desiccant solution is collected at the bottom of the dehumidifier and drained to the weak solution storage tank.

Figure 1.1 Schematic of Open Cycle Liquid Desiccant Dehumidification System.

1.1.2.2. The Collector/ Regenerator

When solar energy is available, the weak desiccant solution is pumped to the Collector/ Regenerator (C/R). As the weak solution flows down the C/R surface, it is heated by solar energy. This raises the vapor pressure of the water in solution. Air is passed over the solution, either by forced convection or by natural convection. As the desiccant solution is heated, the vapor pressure of the water exceeds the partial pressure of the water in the air, and water mass is transferred from solution to the air. Thus, the desiccant solution is reconcentrated, or "regenerated". The regenerated desiccant solution is drained to the strong solution storage tank until needed for dehumidification.

1.1.3. The FSEC Open Cycle Liquid Desiccant Dehumidifier Project

The Open Cycle Liquid Desiccant Dehumidifier System (OCLDDS) is proposed as a ventilation air dehumidification treatment alternative to conventional vapor compression dehumidification for commercial applications such as supermarkets. The OCLDDS could be designed to meet all or part of the latent portion of the ventilation air conditioning load to a building. Thus, the conventional vapor compression air conditioning unit need meet only the sensible portion of the load. This allows a smaller vapor compression unit to be specified and also allows the vapor compression unit to operate more efficiently. Solar energy is used to regenerate the liquid desiccant. Solar-powered liquid desiccant regeneration is described in detail later in the report.

The Florida Solar Energy Center (FSEC) has a multi-year goal to design, build, and field test a solar thermally powered OCLDDS. This is to be accomplished in three parts: (1) construction and operation of an experimental C/R test bed, (2) construction and operation of a fully-operational small scale system, (3) construction and operation of a field system for demonstration in a monitored commercial building.

All components of the OCLDDS are commercially available, with the exception of the C/R. The C/R is constructed from commercially available materials using conventional roofing construction techniques. Some experimental research is required to determine the optimum C/R design. Data from the test bed will be used to design and then construct a small scale operation open cycle dehumidifier system at FSEC. Subsequently, a field demonstration site will be sought. The small scale system will provide operating experience prior to installation of the commercial-sized demonstration field system. Operation of the field system will be used to demonstrate the viability of the OCLDDS.

1.1.4. Task Objectives

Two major milestones relating to Task 1 were identified in the Federal Assistance Management Summary Reports submitted during CY 1993. To accomplish the first milestone, "Analysis of Open-Cycle System Collector/Regenerator", a detailed search of the literature was conducted to determine the state of the art in open cycle regeneration, noting software performance model predictions and experimental results. Based on the literature search, several significant

parameters were identified which require additional study in order to determine optimum performance of the C/R component. This work is discussed later in the report.

To accomplish the second milestone, "Design/Fabricate Test Facility for Collector/ Regenerator Tests", testing ranges were identified based on the literature study. Experimental data gained by testing in these ranges will help determine optimum C/R designs. Calcium chloride was selected as the preferred desiccant, primarily due to its low cost. Design and fabrication of the C/R test bed has begun.

1.2. Literature Review

1.2.1. Introduction

This literature review is limited to glazed and unglazed collector/regenerator (C/R) systems. Many different designs which use solar energy to regenerate liquid desiccants have been studied. For example, solar stills have been proposed to recover water from brines (e.g. Hollands, 1963; Kakabaev and Golaev, 1971). The use of solar heated air to indirectly regenerate desiccants in packed towers has also been discussed (e.g. Lenz and Potnis, 1991; Merrifield and Fletcher, 1977, Robison *et al.*, 1984). This present literature review covers open cycle designs, such as those done by Lenz and Potnis or Robison *et al.*, only. Open cycle systems allow the desiccant to be in direct contact with ambient air during regeneration. The FSEC research focusses only on direct regeneration of liquid desiccants. Direct regeneration means the desiccant is in direct contact with the solar-heated surface within the collector itself. Indirect regeneration, on the other hand, means the desiccant is regenerated in a packed tower by air that is heated elsewhere.

Open cycle solar regeneration of liquid desiccants is accomplished by heating a solution until the water vapor pressure exceeds the partial vapor pressure of the water in the ambient air. It is evident that heating of the solution is directly related to the ambient air temperature and to how much insolation is transferred to the solution, and inversely related to the amount of humidity in the air. Thus, water evaporation rates will be higher on warm, sunny days than on cool, cloudy days, and higher in dry climates than in humid climates. In addition, other parameters, such as C/R geometry and solution inlet conditions, also affect evaporation rates. The impacts of these parameters on performance have been studied in the literature, but the results have not been consistent. In some cases, tests were conducted under widely varying operating conditions, making comparison of results difficult. In an effort to summarize the results obtained in the literature, the literature review presented here is in two sections. Section 1.2 presents a general summary of the literature, grouping research efforts according to the type of C/R tested and type of research (*i.e.*, analytical or experimental). Section 1.3 examines specific research efforts as they pertain to the FSEC OCLDDS project, and uses them to help guide the research efforts planned for the OCLDDS test bed.

Regeneration performance is measured by different means throughout the literature. Some researchers use evaporation rate, (also called water loss rate), because the amount of evaporated water directly impacts how much cooling or dehumidification can be achieved. Others measure

performance in terms of efficiency (commonly defined as the ratio of energy used to evaporate water divided by available radiation), solution outlet temperature, or outlet concentrations. This report uses evaporation rate to measure performance, unless otherwise indicated.

1.2.2. Unglazed Collector/Regenerators

1.2.2.1. Analytical Efforts

Kakabaev and Khandurdyev (1969) described an open cycle lithium bromide absorption system for the dry climate (partial pressures of 5 - 10 mm Hg at 35°C) of Ashkhabad, Turkmenistan. An energy balance showed that performance was dependent on the ratio of inlet solution flow rate over length, solution inlet conditions, and ambient conditions (insolation, humidity, wind speed). Collier (1979) expanded the analysis and included lithium chloride as a desiccant material option. He showed that performance increases to an optimum as inlet solution flow rate (per unit area) decreases. That is, lower flow rates and/or longer C/R lengths lead to higher evaporation rates. Gandhidasan (1983) studied a small system using calcium chloride. A simple mass balance showed performance increasing with solution inlet temperature, since the solution vapor pressure is higher. Likewise, performance decreases with increasing concentration.

1.2.2.2. Experimental Efforts

A 58 kW (16 ton) unglazed open cycle absorption cooling system was built and operated near Ashkhabad, Turkmenistan (Kakabaev, *et al.*, 1977). Operating experience was described. For example, using a tilt of 5 degrees and a rubberoid C/R absorber surface, a minimum of 80 l/h per C/R width was required to maintain full wetting. The desiccant solution was a mixture of calcium chloride and lithium chloride in water. Grodzka and Rico (1982) experimentally tested Collier's model and their own finite difference model. Both models showed good agreement, but each was slightly higher in predicting the experimental outlet solution temperature on days with low wind. Maximum evaporation rates were around 0.40 kg/m² hr. Gandhidasan (1983) operated a 1 m² regenerator with calcium chloride, and reported peak evaporation rates of 0.45 kg/m² hr. Over the past decade, Arizona State University has done extensive work with open cycle systems, both unglazed and glazed. For example, Hawlader *et al.* (1993) conducted experiments on an 11 m x 11 m collector, measuring solution flow rate, concentration and temperature, under various ambient and flow conditions.

1.2.3. Glazed Collector/Regenerators

1.2.3.1. Analytical Efforts

Kakabaev *et al.* (1972) simulated an open cycle system with a gap height of 5 cm, length of 5 m, and inlet solution flow rate of 8 kg/m² hr. For humid conditions (ambient partial pressure of 18 mm Hg), evaporation was a weak function of the ratio of inlet solution flow rate to inlet air flow rate. Optimum values of this ratio varied from 0.1 to 0.25 (with corresponding peak evaporation rates of 0.5 - 0.3 kg/m² hr). In each case, increasing inlet air flow rate (per unit

area) above the optimum resulted in a significant performance decrease. Gandhidasan (1982) provided a correlation for the evaporation rate as a function of the ratio of inlet solution flow rate to inlet air flow rate in a glazed C/R. He considered a glazed C/R in a humid climate (partial pressures up to 21 mm Hg), with a 45% calcium chloride solution flow rate held at $5 \text{ kg/m}^2 \text{ hr}$. He confirmed Kakabaev's glazed peak evaporation rates, but at inlet flow rate ratios around 0.01. Again, increasing inlet air flow rate (per unit area) above the optimum resulted in a significant performance decrease. He also pointed out that preheating either the solution or the air gives better performance. Gandhidasan (1983) also considered a C/R with air and solution in parallel flow (same direction), rather than counterflow (opposite direction), as is commonly done. This analysis showed peak evaporation rates slightly higher than the counter flow mode. Optimum values of the inlet flow rate ratio were also slightly higher. Haim *et al.* (1992) performed detailed simulations of two open cycle absorption systems: one with direct regeneration (using a C/R), one with indirect regeneration (using solar-heated air in a packed tower). The direct system was considered for dry ambient conditions (temperature of 29.4°C , humidity ratio of 0.010), using a C/R with a length of 2 m and a gap height of 5 cm. For both the air flow rate and the lithium chloride solution flow rate, base case values were set for flow rate, inlet temperature and concentration. A "figure of merit" (reflecting the ability of the C/R to evaporate water, for a given insolation) described performance as a function of individual parameters. Performance dropped for inlet solution flow rate under $1.5 \text{ kg/m}^2 \text{ hr}$ for the range of inlet solution temperatures. Performance also dropped for inlet air flow rate under about $6 \text{ kg/m}^2 \text{ hr}$. Optimum figures of merit were evident as desorption conditions became difficult, (*i.e.*, inlet solution and air temperatures decreased, or inlet solution concentration increased). Solution flow rates around $3 \text{ kg/m}^2 \text{ hr}$ showed an optimum figure of merit around 0.4 for low inlet solution temperature.

1.2.3.2. Experimental Efforts

Mullick and Gupta (1974) built and tested a small (length of 0.8 m) glazed C/R in the hot, humid climate of Madras, India. The air flowed by natural convection, and the calcium chloride solution concentration varied from 45 to 50%. The C/R efficiency (energy for evaporation divided by incident solar energy) dropped off as insolation decreased. A maximum efficiency of 40% was obtained under bright skies. Kakabaev *et al.* (1978) tested a natural convection glazed C/R using lithium chloride, varying inlet solution concentration and flow rate, and gap height. Using the data, they developed empirical correlations for evaporation rate and outlet solution temperature, basing the correlations on the theory that natural convective air flow is dependent on an algebraic term that includes the gap height, length, and C/R tilt angle. They claimed that performance is a weak function of inlet solution concentration, but a strong function of the aspect ratio and the algebraic term above. For a length of 1.5 m and tilt of 30 degrees, maximum evaporation was achieved with a gap height of 3 cm.

Robison built and operated a fully-functional glazed open cycle desiccant cooling system (see *e.g.* Robison, 1981; Robison and Harris, 1982; Robison, 1983a; Robison, 1983b; Robison, 1984). The system supplied about 12.3 kW (3.5 ton) of cooling as well as winter heating for a house on the South Carolina coast, using a collector with total regenerating area of 56 m^2 (length of 4.6 m, gap height of 5 cm), and a mixture of lithium chloride and calcium chloride. All of the weak

solution (inlet solution concentration of 40%) and about half of the air flow was preheated using a 24 m² preheater collector. During typical operation, ambient air (ambient temperature of 28°C, humidity ratio of 0.01) was preheated to 52°C, and the solution was preheated to 62°C. In the preheated section, natural air flow rates were estimated at 7.5 kg/m² hr, and 0.42 kg/m² hr of water was removed. Natural flow rates of the non-preheated air were estimated at 4.7 to 7.2 kg/m² hr, and 0.29 to 0.31 kg/m² hr of water was removed. This demonstrated the performance benefit of preheating. The Robison system also demonstrated the technical feasibility of open cycle liquid desiccant cooling systems.

At Arizona State University, Buck (1992) approximated a glazed C/R using a vertical channel in natural convection. He solved the governing equations for the channel and tested his solution experimentally. He looked at local film temperatures, fluid velocities, and concentration gradients across the solution film and the air stream. Experimental data showed lower film temperatures and higher air flow rates than predicted by the analytical solution, possibly due to flow conditions more turbulent than expected. Ji and Wood (1993a) experimentally tested a glazed C/R (1.06 m wide x 10.52 m long) using lithium chloride solution and forced air flow. Although the system was tested in a dry climate, a humidifier was added to simulate humid ambient conditions. They used system efficiency (ratio of energy to evaporate water over insolation minus parasitic energy) to measure performance as a function of four controlled variables: inlet solution flow rate, inlet air flow rate, humidity ratio, and gap height. Ambient conditions, inlet solution temperature, and inlet solution concentration also changed during the tests. An analysis of variance of the four controlled variables showed that, in all cases, performance was a strong inverse function of inlet solution flow rate. Ji and Wood (1993a) recommended reducing inlet solution flow rate to the lowest value possible that would still ensure complete wetting. The remaining three variables showed approximately similar influence on performance. Plots of the data showed that gap heights under 25.4 cm influenced performance very little, especially as air flow rate decreased. The highest efficiencies under high humid and medium humid conditions were between 46 to 48%. These corresponded to an inlet solution flow rate of 5 kg/m² hr, an inlet air flow rate of 34 to 35 kg/m² hr, and a gap height of 17.8 cm.

1.2.4. Comparison of Glazed and Unglazed Collector/Regenerators

1.2.4.1. Analytical Efforts

Peng (1980) developed a numerical model for liquid desiccant absorption systems using triethylene glycol as the desiccant. He modeled both glazed and unglazed regenerators under fixed humid ambient conditions (humidity ratio of 0.018, ambient temperature of 32.2°C) and the same solution flow rate as Collier (1979). His unglazed model compared well with Collier's results. Peng's model predicted solution temperatures that were lower and evaporation rates that were slightly higher than Collier's results. Peng's results showed the glazed regenerator far outperformed the unglazed regenerator: optimum glazed evaporation rates were 2-3 kg/m² hr, versus unglazed evaporation rates under 0.75 kg/m² hr. The high evaporation rates for glazed C/R are not supported elsewhere in the literature. Peng's model indicated that optimum collector

lengths were around 3-4 m, with substantial performance drops for lengths under 1 meter. Solution flow rate was fixed at 112 kg/hr per unit width, or 28 kg/m² hr for a 4 m long collector. Solution flow rates below 28 kg/m² hr (*i.e.*, longer collector lengths) showed no impact on performance. Performance also showed little correlation to the ratio of inlet flow rates. Gandhidasan (1984) compared glazed and unglazed C/R for humid (ambient temperature of 40°C, partial pressure of 22 mm Hg) and dry climates, using models developed earlier (Gandhidasan, 1982, 1983). Evaporation rate in the glazed C/R outperformed the unglazed C/R for both climates. Gandhidasan attributed this to the fact that optimum air flow rate can be controlled in a glazed collector. Glazed evaporation rates reached 0.4 kg/m² hr under high insolation (950 W/m²). Nelson and Wood (1990) found that glazed C/Rs had significant performance improvement over unglazed C/Rs in windy, humid climates. This was attributed to the fact that, in a glazed C/R, the reduced mass transfer potential is offset by the reduced heat loss potential. Glazed C/Rs were also less sensitive to changing ambient conditions. They analyzed a glazed, natural convection C/R using a numerical model that calculated local heat and mass transfer rates. They found that evaporation was mildly sensitive to inlet solution temperature, and almost unaffected by inlet solution flow rate and concentration over the values studied. They suggest the ratio of gap height raised to the fourth power divided by length as one of the characteristic parameters of C/Rs.

1.2.4.2. Experimental Efforts

McCormick *et al.* (1983) placed clear fiberglass corrugated roofing over the Lockheed test bed described by Grodzka and Rico (1982). On average, glazed performance was comparable to unglazed performance. Kaudinya and Kaushik (1986) experimentally compared glazed C/R performance to unglazed performance using a 1.3 m² collector (with length of 1.9 m) in a relatively dry climate having a relative humidity of 35-65%. A fan forced the air through the glazing gap. Air flow rate and gap height were not given, although the gap could be estimated at 5 - 10 cm from a photo. Comparison of glazed and unglazed was difficult due to changing ambient conditions; however, glazed tended to outperform unglazed. Maximum evaporation rates were between 0.8 and 1.2 kg/m² hr. Operation of the air stream in parallel flow with the solution stream did not produce any consistent advantage over counter flow. Folkman *et al.* (1989) compared natural convection glazed and unglazed C/R performance using the Arizona State University test bed. In the dry Arizona climate, the unglazed C/R consistently outperformed the glazed C/R. However, at high inlet concentrations and low inlet solution flow rate, glazed performance approached unglazed performance. They suggest that a glazed C/R may be better than an unglazed C/R in a humid climate due to higher average solution temperatures. Their performance model showed evaporation rate was inversely proportional to gap height, with an optimum gap height below 7 cm. Hawlader *et al.* (1992) also compared a natural convection glazed C/R to an unglazed C/R, finding that performance in the glazed C/R was about 8% lower than performance in the unglazed C/R. However, the glazed C/R was less sensitive to ambient changes. Correlations for Nusselt number and Sherwood number as a function of Lewis number, Grashoff number, and Rayleigh number were developed from data collected for various flow rates, gap heights, concentrations, and ambient conditions. The influence of these parameters on performance was not expressly given. However, simulations showed that evaporation rates

increased slowly with decreasing inlet solution flow rate, and increasing inlet solution temperature. Evaporation was relatively unaffected by ambient temperature below 35°C, and by humidity ratio below 0.017.

It is evident that researchers have reached different conclusions as to which C/R performs better: glazed or unglazed. Simulations by Peng, Gandhidasan, Nelson and Wood indicate that glazed C/Rs outperform unglazed C/Rs. Experimental work by McCormick *et al.*, Kaudinya and Kaushick also showed comparable performance or better with glazed C/Rs. Experimental work by Folkman *et al.* and Hawlader *et al.* showed unglazed C/R performance to be slightly better than glazed performance in dry climates. Benefits of glazed C/Rs include solution contamination protection from dust and debris, and less sensitivity to fluctuating environmental conditions (Nelson and Wood, 1990). Based on the foregoing information, the FSEC Open Cycle C/R will be glazed.

1.3. Collector/Regenerator Test Bed

1.3.1. Collector/Regenerator Research Needs

It is evident from the above literature review that dehumidification using liquid desiccants and desiccant regeneration using solar energy has been studied extensively. Liquid desiccant dehumidification technology could be close to commercial application. Technical feasibility has been demonstrated by the operation of complete open cycle systems (Robison, 1983a; Kakabaev *et al.*, 1977). Performance models have been proposed (with varying degrees of accuracy and detail) for unglazed C/R performance (*e.g.* Collier, 1979; Peng, 1980; Gandhidasan, 1983; Folkman *et al.*, 1989) and for glazed C/R performance (*e.g.* Kakabaev *et al.*, 1972; Gandhidasan, 1982; Hawlader *et al.*, 1992; Haim *et al.*, 1992). Analytical and experimental research has considered optima for some operating parameters under various conditions (*e.g.* Folkman *et al.*, 1989; Nelson and Wood, 1990; Haim *et al.*, 1992; Ji and Wood, 1993a).

Parameter operating levels specific to actual system operation now need to be established. Several detailed performance models show good agreement with experimental data, but fall short of recommending optimum levels for actual operating conditions. Elsewhere, optima have been recommended for some parameters, but not for others. The FSEC C/R test bed will be used to collect experimental data to determine optimum operating conditions for a C/R in the sunny, humid Florida climate. The data collected can be used to design an actual OCLDDS for commercial installation. Specific parameters to be studied are discussed below.

In an open cycle C/R, the objective is to use solar energy to maximize mass transfer of water from solution to ambient air. As discussed above, regeneration of a liquid desiccant solution occurs when the vapor pressure of the water in solution exceeds the partial pressure of the water in the air blowing over the solution. The desiccant solution becomes concentrated (regenerated) as water evaporates from the solution into the air. The vapor pressure of the water in solution increases with solution temperature and decreases with desiccant concentration. The partial

pressure of water in air is a function of ambient temperature and the amount of moisture in the air (measured by either the humidity ratio or the relative humidity). If moisture is added (*i.e.*, the humidity ratio increases) to air at constant temperature, the partial pressure will increase. Adding moisture to the air decreases the tendency of air to absorb moisture from solution. If no moisture is added to the air but the temperature is increased, the partial pressure will decrease. Increasing air temperature (at constant humidity ratio) increases the tendency of air to absorb moisture from solution.

In addition to the above influences on performance, there is a complex interaction between the heat transfer mechanisms and the mass transfer mechanisms in an open cycle C/R, making C/R performance difficult to predict. As the solution stream flows down the C/R surface, it is heated by the absorbed solar energy, increasing the solution water vapor pressure and encouraging mass transfer. As the air stream flows up the C/R, it absorbs moisture from the solution, increasing the partial pressure of water vapor in the air and discouraging mass transfer. If the air stream is cooler than the solution stream, it will tend to cool the solution off. Also, the heats of evaporation and condensation are carried away with the evaporated moisture, cooling the solution stream further. As mentioned before, cooling the solution stream discourages mass transfer.

Inlet conditions of the air stream and solution stream affect C/R performance. Some conditions, such as flow rates and temperatures, can be controlled. Other conditions, such as inlet humidity ratio and solution concentration are set by ambient conditions or design requirements in the absorption dehumidifier, and can not be controlled in the C/R. In addition, the benefit of preheating and controlling inlet temperatures of the air stream or solution stream must also be balanced against the capital cost of preheater equipment. Geometry of the C/R also plays an important part in performance, especially for C/R designs in which the air stream is heavily influenced by natural convection.

The three C/R units of the FSEC Open Cycle test bed will collect data to help determine optimum dimensions for an operational C/R. Some of the parameters to be studied by the FSEC Open Cycle C/R test bed are discussed below. Open cycle C/Rs can be operated in counter flow (solution stream flowing down, air stream flowing up) or parallel flow (both solution stream and air stream flowing down). Only counter flow C/Rs are to be considered initially.

1.3.2. Parameters To Be Tested

1.3.2.1. Ambient Conditions

Virtually every researcher noted in the literature has dutifully reported that regeneration increased with increasing insolation and decreased with increasing ambient humidity. However, few have attempted to circumvent the uncontrollable ambient conditions while studying other important parameters. One exception is Hawlader *et al.* (1992) who conducted side-by-side tests of a glazed and an unglazed C/R. The FSEC C/R test bed has three side-by-side units to allow testing of the parameters discussed above under identical ambient conditions.

1.3.2.2. Collector/Regenerator Length

Analytical work which examines the impact of C/R length on performance indicates that there is a minimum length below which evaporation rate drops off significantly (e.g. Collier, 1979 for unglazed C/R; Peng, 1980 for glazed C/R). Under some conditions, evaporation rate drops slightly as length increases beyond an optimum. The optimum length depends on ambient and other C/R operating conditions. For an unglazed C/R in hot, humid conditions, Collier (1979) predicted optimum values of inlet solution flow rate between 2 and 6 kg/m² hr. Assuming a solution flow rate per unit width of collector of 112 kg/m hr from Kakabaev's work, Collier noted that required collector lengths could be as long as 56 m for unglazed C/Rs. Analysis of glazed C/Rs predicted optimum lengths of only 3 to 4 meters with significant performance drop off at lengths below one meter (Peng, 1980). However, these optima were based on a relatively high solution flow rate, and predicted unusually high evaporation rates. Representative experimental glazed C/R lengths range from 0.8 m (Mullick and Gupta, 1974) to 10.52 m (Ji and Wood, 1993a). Robison (1983a) obtained good results with a C/R length of 4.6 m.

The impact of C/R length on regeneration performance will be studied. Although many analytical results in the literature have indicated a minimum desired C/R length and some have indicated an optimum length, experimental research has held C/R length constant while studying other parameters. Each unit of the FSEC Open Cycle test bed can test C/R lengths up to 7 m long, allowing examination of optimum C/R length. The seven meter maximum length exceeds most lengths tested in the literature, and allows construction using standard 4 x 8 ft plywood sheeting. Furthermore, the three units can be linked in series with additional ducting to approximate C/R performance at lengths up to 21 m.

1.3.2.3. Collector/Regenerator Gap Height

Gap height influences natural convection and the development of the air flow in the C/R. Kakabaev *et al.* (1972) and Robison (1983a) fixed gap height at 5 cm and 3.8 cm, respectively. Folkman *et al.* (1989) recommended gap heights of 2 to 3 cm, based on a performance model run at gap heights from 15 to 7 cm. Simulations by Hawlader *et al.* showed almost negligible change over a range of gap height from 7.4 to 26.3 cm (for C/R length of 10.5 m). Ji and Wood (1993a) tested gap heights from 1.3 cm to 25.4 cm. Optimum gap heights for high humidity conditions were approximately 7.6 to 17.8 cm. However, there was little variation in performance down to 1.3 cm. Only data for a 25.4 cm gap height (the largest gap tested) showed significant performance decrease.

Based on the above discussion, it is anticipated that the optimum gap height and our intended test range will be within 1.5 to 10 cm. Gap heights outside this range will be studied as warranted. Each of the three units in the FSEC Open Cycle test bed can be set at a different gap height, allowing for testing under identical ambient conditions.

1.3.2.4. Solution Flow Rate

In addition to C/R length, solution flow rate helps determine the time that the solution is in contact with the air stream. As the solution contact time with the air is increased, equilibrium between the solution and the air is approached. This would indicate an optimum flow rate of zero, which is illogical if continuous regeneration is desired. Through the use of simulations, optima of approximately 3 kg/m² hr have been predicted (Haim, 1992). For proper operation, a minimum flow rate is necessary to assure complete wetting of the C/R surface. Hawlader *et al.* (1992) tested the impact of solution flow rates down to 6.4 kg/m² hr, and recommended flow rates as low as possible, providing the C/R surface is completely wetted. Ji and Wood (1993a) echoed Hawlader's recommendation based on data for flow rates as low as 5 kg/m² hr. Both were unable to test lower flow rates to insufficient wetting issues.

An experimental approach to addressing the wetting issue is to lower the tilt angle to reduce the influence of gravity on the solution flow rate. A disadvantage is that the low angle will decrease incident insolation. To address this point, the tilt angle of each unit of the FSEC Open Cycle test bed will be varied during initial testing.

1.3.2.5. Air Flow Rate

As the air stream flows up the C/R, changing the flow rate has two counteracting effects on performance. At a relatively high air flow rate the air has a lower average humidity which encourages mass transfer. At the same time, a high air flow rate tends to cool the solution which discourages mass transfer. In fact, the simulations of Haim (1992) show no substantial performance reduction for air flow rates down to 4.5 kg/m² hr. A similar conclusion can be drawn from Kakabaev *et al.* (1972). Note that Haim's simulations showed an increase in this "cut-off" air flow rate as air temperature increased. Ji and Wood (1993a) published experimental data for air flow rates from 18 to 54 kg/m² hr. Performance data for humid climate and their optimum gap height (3.8 cm) showed that air flow rate influenced performance somewhat, with a flow rate of 54 kg/m² hr approaching an optimum efficiency.

The FSEC Open Cycle test bed will allow study of a range of air flow rates from 4.5 kg/m² hr to around 60 kg/m² hr in each unit.

1.4. Performance Analysis

As part of an exercise reviewing the DOE Solar Cooling program, we were asked to perform an analysis of the OCLDDS to get a comparison of the benefits of an Open Cycle Liquid Desiccant Dehumidification System as compared to a conventional Vapor Compression System for treating ventilation air. A load profile for a 2787 m² (30,000 ft²) supermarket in Miami was developed in order to provide a rough estimate of the collector and storage sizes. A copy of our submission is included as an appendix at the end of this annual report.

1.5. Bibliography

- Buck, G.A., and Wood, B.D., 1992. Final Report: Open-Cycle Absorption Solar Cooling: Natural Convection Heat and Mass Transfer from Falling Films in Vertical Channels, Report # DOE/SF/16345--5, Solar Energy Research Laboratory, Arizona State University, Tempe AZ.
- Collier, R.K., 1979. "The Analysis and Simulation of an Open Cycle Absorption Refrigeration System", Solar Energy, v.23, pp.357-366.
- Ertas, A., Anderson, E.E., and Kiris, I., 1992. "Properties of a New Liquid Desiccant Solution-Lithium Chloride and Calcium Chloride Mixture", Solar Energy, v.49, #3, pp.205-212. FSEC RF92-547.
- Folkman, C.C., Stack, A.P., Wood B.D., 1989. Heat and Mass Transfer from Glazed and Unglazed Open Flow Liquid Absorbent Solar Collector/Regenerators, Report #DOE/SF.16345--4, Final Report for USDOE Contract #DE-FG03-86SF16345, Solar Energy Research Laboratory, Arizona State University, Tempe, AZ.
- Gandhidasan, P., 1982. "Simple Analysis of a Forced Flow Solar Regeneration System", Journal of Energy, v.6, no. 6, (#AIAA 82-4281), pp. 436-437.
- Gandhidasan, P., 1983a. "Testing of an Open Solar Regenerator", Proceedings of the 18th IECEC Meeting, v. 4, p.1946-1951, Aug, 21-26, 1983, Orlando, FL.
- Gandhidasan, P., 1983b. "Thermal Performance Predictions and Sensitivity Analysis for a Parallel Flow Solar Regenerator", Journal of Solar Energy Engineering, v.105, pp. 224-228.
- Gandhidasan, P., 1984. "Comparative Study of Two Types of Solar Regenerators for Liquid Absorption Dehumidification System", ASHRAE Transactions, v.90, Pt. 2.
- Haim, I., Grossman, G., Shavit, A., 1992. "Simulation and Analysis of Open Cycle Absorption Systems for Solar Cooling", Solar Energy, v. 49, no. 6, pp. 515-534.
- Hawladar, M.N.A., Stack, A.P., Wood, B.D., 1992. "Performance Evaluation of Glazed and Unglazed Collectors/ Regenerators in a Liquid Absorbent Open-cycle Absorption Cooling System", International Journal of Solar Energy, v.11, pp.135-164.
- Hollands, K.G.T., 1963. "The Regeneration of Lithium Chloride Brine in a Solar Still", Solar Energy, v.7, no. 2, pp. 39-43.
- Ji, L.-J. and Wood, B.D., 1993a. "Performance Enhancement Study of Solar Collector/ Regenerator for Open-Cycle Liquid Desiccant Regeneration". Solar 93, ASES Annual Conference Proceedings, S. Burley and M. Arden, eds. Washington, D.C., April 25, 1993.

- Ji, L.-J. and Wood, B.D., 1993b. Experimental Study of the Heat and Mass Transfer in Solar Collector/ Regenerator for Open-Cycle Liquid Desiccant Regeneration. Draft Final Report for DOE Contract # DE-AC0283CH10093, Subcontract #A0-2-12024-1, Solar Energy Research Laboratory, Arizona State University, Tempe, AZ.
- Kakabaev, A., and Golaev, M., 1971. "Glazed Flat Surface as a Solution Regenerator for Use in an Absorption Solar Cooling System", Geliotekhnika, v.7, no. 4, pp.44-49.
- Kakabaev, A., and A. Khandurdyev, 1969. "Absorption Solar Refrigeration Unit with Open Regeneration of Solution", Geliotekhnika, v.5, no. 4, pp.28-32.
- Kakabaev, A., Klyshchaeva, O., Khandurdiev, A., 1972. "Refrigeration Capacity of an Absorption Solar Refrigeration Plant with Flat Glazed Solution Regenerator", Geliotekhnika, v.8, no. 2, pp.60-67.
- Kakabaev, A., Klyshchaeva, O., Khandurdyev, A., Kurbanov, N., 1977. "Experience in Operating a Solar Absorption Cooling Plant with Open Solution Regenerator", Geliotekhnika, v.13, no.4, pp. 73-76.
- Kakabaev, A., Klyshchaeva, O., Tuiliev, S., Khandurdyev, A., 1978. "Experimental Study of Thermotechnical Characteristics of Glazed Solution Regenerator", Geliotekhnika, v.14, no.4, pp. 42-45.
- Kaudinya, J.V. and Kaushik, S.C., 1986. "Experimental Validation of Theoretical Studies on Open and Forced Flow Solar Regenerator", International Journal of Solar Energy, v.4, pp. 13-23.
- Lenz, T.G., and Potnis, S.V., 1991. "Energy Balance and Mass Transfer Studies for a Liquid Desiccant Based Solar Cooling System", Solar World Congress, Proceedings of ISES, M. Arden, S. Burley, M. Coleman, eds., Pergamon Press, Denver, CO, Aug 19-23, 1991.
- McCormick, P.O., Brown, S.R., Tucker, S.P., 1983. Performance of a Glazed Open Flow Liquid Desiccant Solar Collector for Both Summer Cooling and Winter Heating, Report # LMSC-HREC TR D86/7353, Final Report for USDOE Contract #DE-AC03-82SF11658, Lockheed Missiles and Space Co, Inc, 4800 Bradford Drive, Huntsville AL, 35807.
- Merrifield, D.V., and Fletcher, J.W., 1977. Analysis and Development of Regenerated Desiccant Systems for Industrial and Agricultural Drying, (Final Report), Report #ORNL/Sub-7296/1, Lockheed Missiles and Space Company, Inc., Huntsville AL.
- Mullick, S.C., and Gupta, M.C., 1974. "Solar Desorption of Absorbent Solutions", Solar Energy, V. 16, pp.19-24.

- Nelson, D.L., and Wood, B.D., 1990. "Evaporation Rate Model for a Natural Convection Glazed Collector/ Regenerator", Journal of Solar Energy Engineering, v.112, pp.51-57.
- Peng, C.S.P., 1980. The Analysis and Design of Liquid Absorbent/Desiccant Cooling/ Dehumidification Systems for Low Grade Thermal Energy Applications, PhD Thesis, University of Texas, Austin, TX.
- Peng, C.S.P., and Howell, J.R., 1984. "The Performance of Various Types of Regenerators for Liquid Desiccants", Journal of Solar Energy Engineering, v. 106, pp. 133-141.
- Robison, H.I., 1981. Chemical Heat Pump and Method U.S. Patent # 4,287,721 (Sept. 8, 1981).
- Robison, H.I., 1983a. "Operational Experience with a Liquid Desiccant Heating and Cooling System", Proceedings of the 18th IECEC Meeting, v. 4, p.1952-57, Aug, 21-26, 1983, Orlando, FL.
- Robison, H.I., 1983b. Open-Cycle Chemical Heat Pump and Energy Storage System, Final Report of Research Program for Period June 1982 - September 1983, USDOE Contract #DE-AC03-82SF11657, Coastal Carolina College / University of South Carolina, Conway, SC.
- Robison, H.I., 1984. "Passive and Low-Energy Research and Practices- Dehumidification", Energy and Ecology in Houses - PLEA 1984, A. Bowen, ed., Pergamon Press, Ltd, Oxford, England.
- Robison, H.I., Griffiths, W., Arnas, O., 1984. "Liquid Desiccant/Vapor-Compressor Hybrid Air Conditioner and Energy Storage System", "Solar in the Southeast", Conference Proceedings, April 27-29, 1984, North Carolina Solar Energy Association, Wrightsville Beach, NC.
- Robison H.I., and Harris, H., 1982. "Year-Round Operational Test Results of a Chemical Heat Pump Powered by Waste Heat, Off-Peak Electricity, or Solar Energy", Energex '82, Conference Proceedings, Vol.I/II, Aug. 23-29, 1982, Regina, Sask., Canada.
- Robison, H.I., and Houston, S.H., 1979. "Thermo-Chemical Energy Storage for Heating and Cooling", Solar Energy Storage Options, Vol. II, CONF-790328-P2, A.I. Michaels, Argonne National Laboratory, chair, Proceedings of USDOE Workshop, San Antonio, TX, March 19-20, 1979.

Task 2: Hybrid Solar/Electric Absorption Refrigeration System

2.1. Introduction

2.1.1. The Need for Low Temperature Absorption Refrigeration

Absorption refrigeration technology has been commercially proven, and is most commonly found in commercial and industrial applications using waste process heat. Manufacturers such as The Trane Company and York International Corporation offer commercial units of 350 kW (100 tons) and up. These units vary according to number of stages, firing method, and refrigerant/absorbent pair. Of the many refrigerant/absorbent pairs tested, the most commonly used are water/lithium bromide and ammonia/water. Residential units, while not as common, are also available. Servel, a familiar name in the solar absorption field from the 1970's and early 1980's, is now marketed by Robur USA. The ammonia/water systems are fired from natural gas or propane, are air cooled, and are marketed in 10.6, 14, and 17.6 kW units. American Yazaki offers single effect water/lithium bromide units in 17.6, 26.4, and 35.2 kW sizes. These units may be fired with hot water as low as 75°C (167°F), but with significantly reduced cooling capacity.

Absorption systems operate at two pressures and two concentrations. The high pressure is set by the condenser temperature, and is the saturation pressure of the refrigerant. The generator pressure operates in equilibrium with the condenser pressure. For a given pressure and generator temperature, the absorbent solution will tend to boil off refrigerant to reach a saturation concentration that is in quasi-equilibrium with that pressure and temperature. The low pressure is set by the evaporator temperature. The absorber operates in equilibrium with the evaporator pressure. For a given pressure and absorber temperature, the absorbent solution will tend to absorb refrigerant to reach a saturation concentration that is in quasi-equilibrium with that pressure and temperature.

A mass balance on either the absorber or generator shows that refrigeration capacity is a function of the difference in concentrations. That is, a smaller difference in concentrations means a reduction in refrigeration capacity. As the generator temperature decreases, the generator equilibrium concentration decreases. This results in a smaller difference between the absorbent strong and weak concentrations (with a corresponding reduction in cooling capacity), and/or an increase in evaporator temperature to compensate for the decreased generator output.

Few standard solar flat plate collectors attain outlet temperatures in excess of 80°C (176°F), except perhaps at high noon. Highly efficient or concentrating collectors are necessary to consistently achieve design firing temperatures for absorption systems. Their high cost (relative to standard flat plate collectors) generally precludes the use of solar energy in solar residential absorption cooling. Thus, residences with existing solar space heating systems have been forced to use electric vapor compression or gas-fired absorption systems during the cooling season, while their solar system sits idle. If the solar panels can be used to meet the cooling load as well

as the heating load, the system annual availability (number of hours operation per year) will increase considerably. Utilities will also experience lower summer peaking loads.

2.1.2. The FSEC Hybrid Solar/Electric Absorption System

The Florida Solar Energy Center is studying a design modification to an absorption system which will allow the system to maintain refrigeration capacity in spite of lower generator temperatures. The hybrid solar/electric project modifies the absorption system by the addition of a refrigerant compressor between the evaporator and the absorber, as shown in Figure 2.1. With the addition of a relatively small amount of electric energy, the hybrid system will allow absorption systems to operate at full capacity and COP in spite of low generator temperatures. This should allow solar-fired absorption using standard flat plate collectors. A logical application of this concept is on residences with both heating and cooling loads annually, and existing solar heating systems to satisfy the heating load.

2.1.3. Task Objectives

Task 2 outlines work to be done analytically evaluating and designing an advanced solar desiccant cooling cycle. The Hybrid Solar/Electric Absorption system is evaluated. A performance analysis has shown that thermal output improvement will offset increased electrical cost for compression. This task was performed concurrently with Task 1.

A work plan was developed as indicated in the November, 1992 Technical Progress Report, and was defined by three subtasks as follows:

1. Define, jointly with University of Wisconsin staff, a TRNSYS modeling effort which will characterize the potential benefits of the hybrid system. University of Wisconsin is to develop software as required for this task.
2. Based on literature study and analysis, define preferred compressor characteristics.
3. Fabricate flexible test bed for performance tests of candidate compressors and initiate test program.

The University of Wisconsin has indicated that they will be unable to conduct TRNSYS modeling due to restricted funding. Evaluation of system parameters has been conducted by FSEC using spreadsheets.

A review of the literature related to low temperature absorption refrigeration has been conducted. This has been useful in identifying typical operating conditions for absorption systems. Analysis of the lithium bromide/water absorption refrigeration cycle has been made and the impact of a compressor on system performance has been studied. This has lead to basic operating specifications for, and design characteristics of, the compressor. A paper summarizing the results of the work on the Hybrid Solar/Electric Absorption system was prepared and presented at the 1994 ASME/JSME/JSES International Solar Energy Conference in San Francisco, California.

Figure 2.1 Schematic of Hybrid Solar/Electric Absorption System.

Fabrication of a compressor test bed has not been completed due to program delays. These delays have been due, in part, to time and money spent responding to a DOE request for information on the solar cooling program.

2.2. Literature Review

Extensive research has been conducted in absorption refrigeration, including solar-fired refrigeration. Various applications of the technology have been proven, and absorption units using high generator temperatures (relative to typical flat plate collectors) are commercially available in a wide range of sizes. A bibliography at the end of this report is a partial listing of documents in the solar absorption refrigeration field. References in this report may be found in the bibliography.

The basic absorption cycle has been well documented, and discussions of varying complexity may be found in the textbooks and handbooks listed (*e.g.* Stoecker and Jones, 1982; Threlkeld, 1970; Perry and Green, 1984), as well as in conference and journal papers.

Chinnappa (1992) presented a good cost comparison of a solar absorption system with a "conventional" piston-type vapor compression system. He examined a solar heated absorption system with auxiliary backup to determine what solar fraction would be needed to match the benefit of a conventional vapor compression system. He compared the non-solar energy cost to operate an absorption system with backup with the non-solar energy cost to operate a vapor compression system. Non-solar energy costs included parasitic cost plus either auxiliary heat cost (in the case of absorption), or electric cost (in the case of vapor compression). He determined that a fairly high solar fraction is necessary, depending on the relative unit cost of heat versus electricity. For example, using his values for a representative system, if electric unit costs were twice those of thermal energy, the critical solar fraction for the absorption system was 0.67. If the unit costs were the same, the critical solar fraction was 0.84. High solar fractions are more easily obtained using concentrating or evacuated tube collectors, rather than standard flat plate collectors.

Firing absorption systems using high temperature collectors is technically feasible (*e.g.* Ward, 1979). However, concentrating collectors require direct radiation, and high temperature collectors, including high efficiency flat plate collectors, are more expensive and less common than "conventional" flat plate collectors. Consequently, research has also been conducted using low temperature heat from flat plate collectors to fire absorption systems (*e.g.* Anderson *et al.* 1976; Bierman, 1979; Chinnappa, 1962; Kochhar and Satcunanathan, 1981; Porter, 1976; Ward *et al.*, 1978; Ward, 1979; Whitlow, 1976). Anderson (1974, 1976) reported on work done by Arkla. A gas fired system modified to use solar heat required 99°C (210°F) temperatures supplied to the generator, but could operate at lower temperatures with reduced capacity. Anderson also commented that testing was underway on another unit to use 90.5°C (195°F) water, with 29.4°C (85°F) cooling tower water, but he had no results at the time of his presentation. Krusi, *et al.* (1981) operated a 7kW Yazaki chiller with flat plate collectors. They demonstrated the technical feasibility, but noted reduced capacity, as is generally predicted (*e.g.*

Anderson *et al.*, 1976; Miller, 1976; Porter, 1976). In addition, the Krusi system achieved a maximum COP of only 0.6. The Krusi system collectors supplied 76.7°C (170°F) heat. Bergquam (1993) also successfully operated absorption systems with flat plate collectors. One 35.2 kW (10 ton) system operated at a firing temperature of 87.8°C (190°F). On the hottest days, the solar storage supplied 96.1°C (205°F) heat, increasing capacity to 49.3 kW (14 tons).

These experiences have shown the technical feasibility of firing absorption systems with relatively low heat, although the minimum generator temperature requirements referenced are still generally higher than average operating temperatures of flat plate collectors. Such high temperature requirements would lead to reduced solar fractions calling into question the economic feasibility, as demonstrated by Chinnappa (1992).

Several design modifications to the absorption cycle have been suggested to counteract reduced capacity due to low generator temperatures. Peng and Howell (1981) suggested two designs to direct some of the strong solution exiting the generator through an open absorber to dehumidify an air stream. The COP increased slightly, and the lower limit for the generator temperature decreased considerably. Grossman, *et al.* (1981) suggested adding a solution preheater, using waste solar heat, to reduce generator size and cost. The preheater would heat the weak solution downstream of the standard heat exchanger. They also studied a design using an auxiliary generator to help maximize available solar radiation.

Clark (1979, 1981) filed two patents on the concept of a hybrid absorption/vapor compression refrigeration system, basing the concept on a similar patent by Costello (1977). Chinnappa and Wijesundera (1992) have analyzed the concept as a compressor assisted absorption system with vapor compression backup. Their ammonia/water system is modeled with hot water storage and with refrigerant storage. A similar analysis was given in Chinnappa (1992), as mentioned above. System performance was simulated for a year to show that increasing generator temperatures with solar energy reduces the compressor power. The minimum generation temperature was given as 62.8°C (145°F). However, the direct effect of compressor power on minimum generator temperature was not explicitly shown. The vapor compression backup mode also required that the compressor be sized for relatively large compression ratios. This may require multi-staged compression, especially in a water/lithium bromide unit, which leads to questions of practicality and economic feasibility (Van Orshoven, 1991). Aso *et al.* (1981) developed a solar absorption refrigerator with compressor, using R-22/dimethylformamide as the refrigerant/absorbent pair. They operated in three modes: absorption, hybrid, and vapor compression, depending on the insolation level. They reported thermal COPs of 0.62 to 0.76 in the hybrid mode, slightly better than their COP of 0.62 in absorption mode. System COPs (which include collector efficiency) were less.

Our Task 2, which builds on the work described above for water/lithium bromide absorption refrigeration, is based on a modification of Clark's concept. The current concept differs on three basic points. First, the compressor is not a piston design, as recommended by Clark. Our preliminary design employs a non-positive displacement fan-type compressor driven by a magnetic coupling, thereby eliminating shaft seals. Noncorrosive impeller materials are available

from vendors. Second, water/lithium bromide is a better choice than R-22/dimethylformamide primarily due to environmental considerations. Water/lithium bromide is considered over ammonia/water primarily because ammonia systems generally are more complex (due to rectification needs), operate at lower COPs, and are considered more toxic (Threlkeld, 1970; Ward, 1979; Porter, 1976). The third difference is how the absorption system is operated during periods of little or no sun. If the hybrid system is operated as a solar absorption system with fossil backup, the system will not operate in vapor compression mode, as Clark and Chinnappa suggest. Thus, the compressor need not be sized to provide the high compression ratios seen in the vapor compression cycle. This will reduce the compressor capital cost and improve feasibility.

2.3. Performance Analysis

2.3.1. Analysis Methodology

Our analysis of the water/lithium bromide absorption refrigeration cycle is made using a methodology based on Threlkeld (1970), Stoecker and Jones (1982), and ASHRAE (1985). Given input data, calculation of state points (temperature, pressure, enthalpy, and concentration), mass flow rates, and power is determined at various points around the system in a sequential order. No line losses or irreversibilities are considered, and quasi-steady state operation is assumed. Component temperatures are assumed to be average temperatures within the component, not the temperature of the supply heat or heat rejection streams. Saturated conditions are assumed for the states exiting the evaporator, absorber, generator, and condenser.

Three representative cases are analyzed to establish preliminary feasibility of the concept. The base case is a "typical" absorption system supplying a 3.52 kW (1 ton) load at a temperature of 93.3°C (200°F) in the generator and 4.4°C (40°F) in the evaporator. Case Two assumes the same system and temperatures, but with the generator temperature reduced to 79.4°C (175°F), which results in a reduced capacity. Case Three maintains the low generator temperature, but adds a refrigerant compressor between the evaporator and absorber. The amount of compressor work needed to regain "design" capacity of 3.52 kW is then calculated.

In the Base Case, temperatures in the condenser and absorber are held at 37.8°C (100°F), and an approach temperature of 11.1°C (20°F) at the cold side of the heat exchanger is assumed. Given this information, pressures in the generator/condenser and the evaporator/absorber may be set by the saturation temperatures in the condenser and evaporator, respectively.

Given these pressures and temperatures in the generator and absorber, concentrations are set at 64.6% (strong) and 56.8% (weak), respectively. Refrigerant flow rate may be found from the specified load and the enthalpy difference across the evaporator. Given the pressure difference across the pump and an estimate for specific volume using Perry and Green (1984), pump work is shown to be minimal, and the enthalpies and temperatures across the pump are approximately the same. A total mass balance and a lithium bromide mass balance on the absorber yields the strong solution flow rate. The heat lost by the strong stream across the heat exchanger may now

be found. This leads to the temperature and enthalpy of the weak stream entering the generator, and the overall heat transfer area product of the heat exchanger, $(UA)_{hx}$. The power for each component may now be determined, and the COP calculated. The Base Case cycle is plotted on a lithium bromide/water equilibrium diagram in Figure 2.2. While not all the points shown are truly in equilibrium, the diagram serves as a qualitative representation of the cycle.

Case Two assumes a generator temperature of 79.4°C (175°F) and an unknown load, with the same temperatures as the Base Case in the evaporator, condenser, and absorber. This generator temperature is selected because it is very close to the minimum temperature possible, given other system temperatures. To reduce the generator temperature much lower, it would be necessary to have lower condenser and absorber temperatures. The overall heat transfer area product, rather than the heat exchanger approach temperature, is held fixed to allow some increase in flexibility when determining heat exchanger temperatures.

Given the above information, it is seen that refrigerant saturation pressures and absorber (weak) concentration are the same as in the Base Case, but generator concentration is reduced to 58.8%. Since the pressure change is the same, the weak solution flow rate through the pump is the same as well. Having the concentrations and exiting flow rate, mass balances on the absorber yield the refrigerant flow rate. The heat exchanger NTU and effectiveness may also be calculated to give the power transferred by the heat exchanger. The rest of the state points, powers, and the COP may now be found. The cycle for Case Two is shown in Figure 2.3. It is evident that, although the weak concentration remains unchanged, the strong concentration is significantly reduced. This leads to a concentration difference much smaller than that of the Base Case. In other words, refrigeration capacity is a function of the concentration difference between the strong and weak lithium bromide/water streams.

Case Three considers an absorption system with a refrigerant compressor located between the evaporator and the absorber. Case Three maintains the same input temperatures as Case Two, but a capacity of 3.52 kW is specified. As in Case Two, $(UA)_{hx}$ is held constant. Also, in order to determine new concentrations, it is necessary to specify a solution flow rate. This is because the weak solution concentration, which was fixed in previous cases by the evaporator pressure, can not be determined until the absorber pressure is known. Thus, writing mass balance equations on the absorber leads to two equations and three unknowns. The strong solution flow rate is specified to be the same as in the Base Case.

Given this information, state points, power and COP may then be calculated in a manner similar to the previous calculations. The state of the refrigerant entering the compressor is fixed by the evaporator pressure and temperature. The compressor is assumed isentropic, fixing the entropy of the exiting refrigerant. Given the entropy at the exit and taking the compressor discharge pressure to be the same as that of the absorber, the compressor discharge state is fixed, and isentropic compressor work may be calculated. The cycle for Case Three is shown in Figure 2.4. It is evident that the concentration difference has been regained by shifting the weak solution concentration to the left, *i.e.* in the weak direction. The influence of the compressor on system pressures is also evident.

Figure 2.2 Lithium Bromide/Water Equilibrium Diagram, Base Case.

Figure 2.3 Lithium Bromide/Water Equilibrium Diagram, Case Two.

Figure 2.4 Lithium Bromide/Water Equilibrium Diagram, Case Three.

2.3.2. Results of Performance Analysis

Key parameters for the three cases discussed above are shown in Table 1. These calculations have been independently verified using a computer code developed by Oak Ridge National Laboratories (Zaltash and Ally, 1991). As described above, the Base Case represents somewhat

Table 1: Parameters in Hybrid System for three typical cases

	Base Case	Case 2	Case 3
T generator, [°C]	93.3	79.4	79.4
T evaporator, [°C]	4.4	4.4	4.4
T condenser, [°C]	37.8	37.8	37.8
T absorber, [°C]	37.8	37.8	37.8
Load, [kW]	3.52	0.81	3.52
Strong conc., [%]	64.6	58.8	58.8
Weak conc., [%]	56.8	56.8	51.7
Press. across compressor, [kPa]	0	0	0.686
Isentropic compressor work, [W]	0	0	117
COP	0.77	0.63	0.79

typical operating conditions. In Case Two, the generator temperature is reduced to about 79.4°C (175°F). This reduces the system capacity to one quarter that of the Base Case, or 0.81 kW. Case Three shows that compressing the refrigerant an additional 0.686 kPa changes the weak concentration sufficiently to provide the same load as in the Base Case. This corresponds to a 51% increase in refrigerant density and a ratio of compressor outlet to inlet pressure of 1.82. To provide 3.52 kW of refrigeration at 4.4°C (40°F), approximately 229 l/s of water vapor are needed. Because of water vapor's high specific volume, this corresponds to 1.23 l/s (2.6 SCFM) of air at standard conditions. Given the flow rate and the enthalpies at compressor inlet and outlet, the isentropic work imparted by the compressor on the fluid is 117 W.

If one compares the increase in refrigeration (0.81 kW to 3.52 kW) to the electricity supplied (234 W, assuming 50% compressor efficiency), one would obtain an incremental EER of

$$\text{EER}_{\text{inc.}} = (12,000 - 2,760 \text{ Btu/hr})/234 \text{ W} = 39$$

Note that this is an incremental EER and does not include the thermal energy supplied by the collectors. Thus, it can not be compared to EER's obtained by vapor compression systems. However, it demonstrates that very little electrical energy is needed to regain refrigeration capacity at low generator temperatures.

One interesting question would be how much could be gained by continuing to increase the compressor power, so that the refrigerant capacity is increased beyond that of the Base Case. The Performance Analysis was put in spreadsheet form, and plots made for a range of refrigeration loads. Figure 2.5 shows these results. While the solution leaving the absorber becomes more dilute, reflecting an increasing absorber pressure, the COP approaches 0.81 for loads of 8.8 kW and above. In fact, the COP is already 98% of the maximum for the "design" load of 3.52 kW. The existence of a maximum COP is due to non-linear increases in compressor power as shown in Figure 2.6. There is also an increase in the generator energy. These plots demonstrate that it is not beneficial to increase the compressor power much beyond that needed to regain "design" capacity.

2.4. Compressor Design

Two basic compressor designs are currently being considered to supply the pressure increases and flow rates specified above. Since insolation and refrigeration loads are variable, the compressor should not demonstrate stall at any flow rates between zero and the maximum. A narrow backward curved centrifugal impeller with a diameter under 18 cm has a specific speed that allows rotations under 5000 rpm. However, this design often demonstrates stall at low flow rates. An axial design would avoid stall problems, but has a specific speed requiring rotations on the order of 10,000 rpm. Continental Fan Manufacturing Inc. is an example of one manufacturer who supplies both impeller designs made from glass reinforced polyamide, which has excellent resistance to a wide range of chemicals, including water and ammonia. These impellers also boast a 50% weight reduction from steel impellers (requiring less torque and bearing load).

Magnetic drives currently exist which can supply 7500 rpm and torques typical of liquid pumping. For example, Tuthill Pump Company offers a range of such magnet driven pumps. These drives are used in hazardous and corrosive applications where leakage can not be tolerated. A motor spins a toroidal driver magnet around a corrosion resistant cup made of stainless steel or Hastelloy C which spins a driven magnet inside the cup. The cup is attached to the refrigerant conduit wall with static seals similar to those used elsewhere in the system. Thus, leaky shaft seals are unnecessary. Dry bearings also exist to support the impeller shaft in the presence of water vapor. For example, American Industrial Plastics offers "self-lubricating" (dry) bearings made of Teflon Delrin, which will allow speeds of 6000 rpm and above, depending on the shaft size. This material may also be polished, tapped, and machined to suit specific needs. Specific manufacturing firms are mentioned here to demonstrate that the components needed for the Hybrid Solar/Electric Absorption system are commercially available. The firms listed are not necessarily recommended, but are simply examples of the many manufacturing firms that can competitively offer the needed components. In addition, the seal-less pump industry and the oil-free compressor industry may provide solutions to the water vapor compression sealing problem.

Concern regarding the feasibility of compressing water vapor has been considered. The use of water as a refrigerant in the vapor compression cycle has been studied extensively by Van Orshoven (1991). As is shown in his thesis, water has an extremely high specific volume compared to common refrigerants used in the vapor compression cycle, (900 times that of R-22),

Figure 2.5 Weak Concentration and COP vs. Refrigeration Load.

Figure 2.6 Compressor Power vs. Refrigeration Load.

leading to relatively large volumetric flow rates. The large compression ratios required under typical water vapor compression operating conditions dictated multi-staged compressors, which were found not to be economically viable. However, typical operating conditions of the hybrid system require compression ratios less than even those required of compressors found in conventional vapor compression cycles. Because of reduced frictional losses, Van Orshoven also supports the use of "dynamic" designs such as centrifugal or radial, rather than positive displacement designs for the compressor impeller.

The performance analysis described above indicates that the Hybrid Solar/Electric Absorption concept shows great potential benefit. The commercial availability of the needed components increases the buildability of the refrigerant compressor, and increases the technical feasibility of the Hybrid Solar/Electric Absorption system. A laboratory test bed would allow experimental testing of the compressor designs for addition to an actual absorption system.

2.5. Bibliography

- Allen, R., 1974. "Optimization of Solar Powered Absorption Systems", Solar Cooling for Buildings, Workshop Proceedings, NSF-RA-N-74-063, Feb. 6-8, 1974, Los Angeles, California.
- American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1985. ASHRAE Handbook, 1985 Fundamentals, Atlanta, Georgia.
- Anderson, L.B., Conti, R.J., Whitney, A.K., 1976. "Collector and Storage Considerations for Solar Driven Cooling Systems", Use of Solar Energy for the Cooling of Buildings, 2nd Workshop Proceedings, deWinter and deWinter, eds, NTIS# SAN/1122-76/2, Aug. 4-6, 1975, Los Angeles, California.
- Anderson, P., 1974. "Current Lithium Bromide Hardware as used in Solar Applications", Solar Cooling for Buildings, Workshop Proceedings, NSF-RA-N-74-063, Feb. 6-8, 1974, Los Angeles, California.
- Anderson, P.P., 1976. "Progress Report on Solar Cooling Work at ARKLA", Use of Solar Energy for the Cooling of Buildings, 2nd Workshop Proceedings, deWinter and deWinter, eds, NTIS# SAN/1122-76/2, Aug. 4-6, 1975, Los Angeles, California.
- ASHRAE, see American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.
- Aso, S., Harada T., Tagashira, M., Ebisu, K., Wakamatsu, N., Takeshita, I., Hozumi, S., 1981. "Hybrid Solar Cooling and Heating System", National Technical Report, (Vol) 27, No. 2, pp. 50-60, Apr. 1981, Matsushita Electric Industrial Company, (in Japanese).
- Beckman, W., 1974. "The Lithium Bromide Systems used in Solar Applications", Solar Cooling for Buildings, Workshop Proceedings, NSF-RA-N-74-063, Feb. 6-8, 1974, Los Angeles, California.
- Bergquam, J., 1993. "A Hybrid Solar Absorption Air Conditioning System", Solar Today, July/August, pp.23 - 25.
- Biermann, W.J., 1979. "An Absorption Machine for Solar Cooling", ASHRAE Transactions, v. 85, part 1, 1979 Semiannual meeting, Philadelphia, Pennsylvania.
- Chinnappa, J.C.V., 1962. "Performance of an Intermittent Refrigerator Operated by a Flat-Plate Collector", Solar Energy, v.6, No. 4., pp.143 - 150.
- Chinnappa, J.C.V., 1992. "Principles of Absorption System Machines", Solar Air Conditioning and Refrigeration, A.A.M. Sayigh and J.C. McVeigh, eds. Pergamon Press, New York, New York, pp. 13 - 65.

- Chinnappa, J.C.V., and Wijesundera, N.E., 1992. "Simulation of Solar-Powered Ammonia-Water Integrated hybrid Cooling System", Journal of Solar Energy Engineering, v.114, pp. 125-127.
- Clark, S.W., 1979. "Compressor Assisted Absorption Refrigeration System", U.S. Patent #4,171,619, Oct. 23, 1979.
- Clark, S.W., 1981. "Compressor-Assisted Absorption Refrigeration System", U.S. Patent #4,285,211, Aug. 25, 1981.
- Costello, F.A., 1977. "Combined Absorption and Vapor-Compression Refrigeration System", U.S. Patent #4,031,712, June 28, 1977.
- Curran, H.M., 1982. "Hybrid Refrigeration/Sorption Solar Cooling Systems", Solar Engineering - 1982, Proceedings of the ASME Solar Energy Div. 4th Annual Conference, Albuquerque, NM, 4/26-29/1982, American Society of Mechanical Engineers, 345 E. 47th St, New York, NY, 10017, pp. 538 - 544.
- Farber, E., 1974. "Ammonia and other Absorption Systems Used in Solar Applications", Solar Cooling for Buildings, Workshop Proceedings, NSF-RA-N-74-063, Feb. 6-8, 1974, Los Angeles, California.
- Grossman, G., Bourne, J.R., Ben-Dror, J. Kimchi, Y., Vardi, I., 1981. "Design Improvements in LiBr Absorption Chillers for Solar Applications", Journal of Solar Energy Engineering, v.103, p. 56-61.
- Kochhar, G.S., and Satcunanathan, S., 1981. "Optimum Operating Conditions of Absorption Refrigeration Systems for Flat Plate Collector Temperatures", Solar Engineering - 1981, Proceedings, ASME, Solar Energy Division, Reno, Nevada, pp. 260 - 267.
- Krusi, P., and Schmid, R., 1981. "A critical review of solar absorption air conditioning", Australian Refrigeration, Air Conditioning and Heating, v. 22, (March), Yaffa Pub. Group, Sydney, Australia, pp. 12-19.
- Miller, D., 1976. "Performance of Water Cooled Lithium Bromide Absorption Units for Solar Energy Applications", Use of Solar Energy for the Cooling of Buildings, 2nd Workshop Proceedings, deWinter and deWinter, eds, NTIS# SAN/1122-76/2, Aug. 4-6, 1975, Los Angeles, California.
- Peng, C.S.P., and Howell, J.R., 1981. "Analysis and Design of Hybrid Double-Absorption Cooling Systems for Low Grade Thermal Energy Applications", Journal of Solar Energy Engineering, v.103, pp. 331-338.

- Perry, R.H., and Green, D.W., 1984. Perry's Chemical Engineer's Handbook, 6th ed., McGraw-Hill Book Co. New York, New York.
- Porter, J.M., 1976. "The Use of Commercially Available Absorption Units on Solar-Powered Cooling Systems", ASHRAE Transactions, vol. 82, part 1, (Proceedings, Dallas, Texas) pp. 943-949.
- Sayigh, A.A.M., and McVeigh, J.C., 1992. Solar Air Conditioning and Refrigeration, Pergamon Press, New York, New York.
- Simko, F.A. Jr., 1993. "High Performance Flat-plate Solar Collectors for Industrial Applications", Solar Engineering 1993, Proceedings of ASME International Solar Energy Conference, Washington, D.C., A. Kirkpatrick and W. Worek, eds., American Society of Mechanical Engineers, United Engineering Center, 345 E. 47th St, New York, NY 10017.
- Stoecker, W.F., and Jones, J.W., 1982. Refrigeration and Air Conditioning, 2nd ed., McGraw-Hill Book Co., New York, NY.
- Threlkeld, J.L., 1970. Thermal Environmental Engineering, 2nd ed., Prentice-Hall, Inc., Englewood Cliffs, New Jersey.
- Van Orshoven, D., 1991. The Use of Water as a Refrigerant - an Exploratory Investigation, Master's Thesis, University of Wisconsin, Madison, Wisconsin.
- Ward, D.S., Löf, G.O.G., Uesaki, T., 1978. "Cooling Subsystem Design in CSU Solar House III", Solar Energy, v. 20, pp. 119-126.
- Ward, D.S., 1979. "Solar Absorption Cooling Feasibility", Solar Energy, v. 22, pp.259-268.
- Whitlow, E.P., 1976. "Relationship between Heat Source Temperature, Heat Sink Temperature and Coefficient of Performance for Solar-Powered Absorption Air Conditioners", ASHRAE Transactions, v. 82, part 1, pp. 950-958.
- Zaltash, A. and Ally, M.R., 1991. Validation of a PC Based Program for Single Stage Absorption Heat Pump, Final Report, #ORNL/TM-11923, Oak Ridge National Laboratory, Oak Ridge, Tennessee.

Appendix: Performance Analysis of Open Cycle Liquid Desiccant Dehumidifier

NOTE: This material is taken directly from an informational report submitted in September of 1993 to Science Applications International Corporation, a U.S.D.O.E. contractor assigned the task of reviewing the D.O.E. solar cooling program. The September submission was a revision of an earlier submission. The schematic referred to in the text below is identical to Figure 1.1 in the body of the Annual Report.

Open Cycle Liquid Desiccant Dehumidification System

The Open Cycle System is proposed as a ventilation air treatment alternative to conventional vapor compression air conditioning for commercial applications. The Open Cycle System is a solar-regenerated liquid desiccant system that is proposed to meet all or part of the latent portion of the ventilation air conditioning load. Thus, the total air conditioning load is reduced to only the sensible portion, which allows a smaller vapor compression unit to be specified. This desiccant system would be best used in climates with relatively high latent loads. To this end, a load profile for a 30,000 SF supermarket in Miami is developed in order to provide a rough estimate of the collector size and storage size.

Discussion of Load Calculation Procedure

For supermarkets, ASHRAE Standard 62-89 specifies minimum outside air requirements of 8 l/s per person, and estimates an average of 8 people per 100 m². Thus, an air flow rate of 1,784 l/s outside air is specified for a 2787 m² (30,000 SF) supermarket. The indoor conditions are set at 23.3C (74F) and 50% relative humidity. This gives an indoor humidity ratio of 0.009 kg water per kg dry air. It is assumed that these conditions are maintained 24 hours per day and, as noted above, that the Open Cycle System conditions only the outside air flow. Recirculation air flow is conditioned in a separate path. Thus, indoor loads such as cases are not considered. In this simulation, the Open Cycle System only dehumidifies the air. Therefore, a conventional vapor compression system is located downstream to meet the sensible cooling load.

Outdoor air conditions are based on monthly averages of TMY data for each hour. Data for solar radiation, atmospheric pressure, dry bulb temperature, and dew point temperature are tabulated for each hour of the day. Thus, for each month of the year, an average daily profile is given in hourly increments. Correlations from the ASHRAE 1985 Fundamentals Handbook are used to derive all other air and water conditions such as humidity ratio, specific volume, vapor pressure, and enthalpy. Tables and graphs published by Allied Chemical provide correlations for calcium chloride solutions.

A schematic of the Open Cycle System is given in the original report. Outdoor air is supplied to the absorption dehumidifier tower. In this tower, strong calcium chloride solution is sprayed

over a packing material in a counterflow manner to the air stream. The difference in water vapor pressure allows the desiccant solution to remove water vapor from the air. A similar system operated by Robison *et al.* demonstrated 80% "effectiveness" in removing humidity from the air. The entering solution concentration must be strong enough to assure that the air stream is at the target humidity ratio of 0.009 as it exits the dehumidifier tower. Solution flow rates are based on those selected by Robison for operation of a similar system. Cooling water from a cooling tower or ground water is circulated through a heat exchanger to remove any heat of absorption, maintaining the air stream at constant dry bulb temperature through the dehumidifier tower. The air stream leaving the dehumidifier tower is then cooled to the target indoor dry bulb temperature by a vapor compression unit (labeled "Air Cooling System" in schematic). Thus, the absorption dehumidifier removes the latent heat and the vapor compression unit removes the sensible heat.

The weak solution exiting the dehumidifier tower is stored in a tank until solar radiation is available to regenerate it. The solution is regenerated as described in the original report. The regenerator/collector is located on the roof of a commercial building. As solution flows down the roofing material of the collector, outside air is blown over it in a counterflow manner as described by Robison *et al.*, Gandhidasan, and Collier. The solution is heated and water is driven off and absorbed into the air. TMY radiation data are used and a 47% collector efficiency is assumed. Glazing protects the solution from contaminants such as dust, rain, and leaves, and provides a channel through which the air passes. The regenerated solution returns to a storage tank until used in the dehumidifier tower.

Load Calculation Results

Monthly average daily sums are shown in the Summary Table for each month. These daily sums are the summation of monthly-averaged hourly TMY data. The latent load is given in the first column. This is the load that is met by the absorption dehumidifier tower. The useful solar energy is also given. This allows a collector area and solution storage size to be estimated for each month. The values shown represent the collector area and solution storage needed to supply the daily load. Since the daily load varies from month to month, these also vary. Latent load, insolation, and collector area are plotted together to show trends relative to each other. It is evident that, although insolation stays somewhat constant from March to August, latent load increases to a peak in September. Thus, collector area also peaks in September. In other words, a collector sized to meet the average September daily latent load would be oversized in all other months. Note that sensible cooling and heating is also necessary. Sensible cooling is met by a vapor compression unit as described above. Sensible heating could be met by the desiccant dehumidifier tower with proper control of the cooling tower water. This potential added benefit of the open cycle system is not considered in this simulation.

Based on the load profile just described, the preliminary estimate of required collector area is 428 m² for this supermarket application. A collector area of this size will allow sufficient solar energy to be collected to meet the average daily latent load for all months except July, August, and September. A preliminary estimate of storage size of 5.28 m³ is sufficient to meet the average daily load for all months of the year, for each month's optimum collector area. Note

Summary Table for 30,000 SF Supermarket, Miami; ASHRAE Outside Air Requirements:
(Daily Loads for Conventional and Open Cycle; Open Cycle Collector Size, and Storage Size Estimates)
*based on daily sum of monthly-average hourly TMY data

Open Cycle										Conventional Vapor Compression			
Mo.	Latent [MJ/day]	Insolation [MJ/m ² d]	Qu [MJ/m ² d]	Ac [m ²]	Storage [m ³]	Sensible Cooling [MJ/day]	Sensible Heat [MJ/day]	Cooling [MJ/day]	Reheat [MJ/day]	Heat [MJ/day]			
J	831	11.86	5.57	149	4.79	3	620	2,916	2,703	0			
F	202	15.22	7.16	28	4.35	27	523	2,403	2,698	0			
M	1,403	18.08	8.50	165	4.64	104	252	3,957	2,701	0			
A	1,525	21.59	10.15	150	4.67	217	146	4,290	2,694	0			
M	2,629	20.37	9.57	275	4.59	393	14	5,699	2,692	0			
J	3,701	19.00	8.93	414	4.81	655	0	7,044	2,689	0			
J	4,105	19.85	9.33	440	4.89	710	0	7,507	2,692	0			
A	4,250	19.21	9.03	471	4.92	849	0	7,790	2,690	0			
S	4,387	16.98	7.98	550	4.93	756	0	7,831	2,689	0			
O	2,891	14.59	6.86	421	5.28	335	18	5,895	2,688	0			
N	1,569	13.44	6.32	248	4.97	139	220	4,180	2,693	0			
D	103	11.70	5.50	19	4.32	2	730	2,022	2,701	59			
ANN	843,652					127,985	76,267	1,877,017	983,317	1,820			
	[MJ/yr]					[MJ/yr]	[MJ/yr]	[MJ/yr]	[MJ/yr]	[MJ/yr]

that, during the months of July, August, and September, the average daily latent load will not be met, in spite of sufficient storage size. This is because a collector area of 428 m² will not collect sufficient energy on the average day.

It is important to point out that, due to the nature of the TMY data used, the sizing estimates for collector area and storage size are only preliminary. Final system sizing should be based on meeting some percentage of the maximum anticipated load. For example, ASHRAE publishes 97.5% design dry-bulb temperatures, so that equipment may be sized to meet the load 97.5% of the time. The TMY data presented here are monthly-averaged hourly data which are useful in determining an annual load. However, they are not optimum in determining equipment size. A more accurate sizing estimate for collector area and storage size would be based on maximum loads calculated for each hour of the year. However, the estimates given provide a good preliminary indication of the approximate size of the system.

Parasitic energy costs associated with the Open Cycle System include: 1)pumping of strong solution from storage to the absorption dehumidification tower, 2)pumping of weak solution from storage to collector on roof, 3)pumping of cooling tower water, and 4)power to blow air through the dehumidification tower. Both solution pumping costs (1 and 2) are considered negligible. Pumping solution to the roof requires more energy, due to the longer distance and height difference. A collector flow rate of 112 kg/m/hr is taken from Baum as quoted in Collier, giving 1093 kg/hr for the 9.7 m collector. Assuming a 60% pump efficiency and 75% motor efficiency, 73W are needed to pump solution to the collector. Given a maximum of 6.5 hours of operation, this totals 1.7 MJ/day, or less than 2% of the lowest daily latent load. Most months, the average daily operating time would be less. Calculations using flow rates given by Ji and Wood lead to lower pumping costs. The parasitic costs for pumping cooling tower water (3) and blowing process air (4) are considered to be less than required for a conventional vapor compression system.

Load Calculation for Conventional Vapor Compression System

For comparison purposes, a load profile for a conventional vapor compression system is estimated. Note that this is not the vapor compression system referred to above (and shown as "air cooling unit" in the Open Cycle schematic). It is the system that would be needed if the open cycle system were not installed. It will be referred to as the "conventional" system. Also, please note that this conventional system is not the alternate desiccant system that was described in the original report. That system was described for cost comparisons only, at the request of the DOE contractor. That cost comparison showed that a closed cycle desiccant dehumidification system using flat plate collectors was more costly than the open cycle desiccant dehumidification system.

The load profile for the conventional system is developed using a calculation procedure similar to that used for the open cycle system. This conventional system meets the same indoor space specifications for outside air as those given for the open cycle system. Note that this is the load profile for the outside air requirements only; recirculation air flows are not considered in this

simulation. The load profile shown in the Summary Table represents only the outside air load. Conventional design procedures generally specify a vapor compression unit sized large enough to meet both recirculation load and outside air load, or specify two units in a dual path system.

It is evident that cooling with the vapor compression system requires much more energy than cooling with the open cycle system. This is because of the different psychrometric paths taken to reach the same goal. The vapor compression system sensibly cools outside air past the dew point to condense water vapor and reduce the humidity ratio. The air must then be sensibly reheated to meet indoor air requirements. The open cycle system removes the humidity first. Then a vapor compression system sensibly cools the air to meet indoor air requirements. It is evident that this vapor compressor can be significantly smaller than the one without desiccant dehumidification. In fact, the load is less than 7% of that of a conventional system. This is not including the reheat load, which sometimes is met with waste heat. If electrical reheat is required, the savings with the open cycle desiccant system is even greater.

Part of the conventional system's relatively poor performance is demonstrated by the "Sensible Heat" column under the Open Cycle system. Even in Miami, there is a heating load. However, even though the dry bulb temperature is colder than the required indoor temperature, the humidity ratio is almost always higher. Consequently, the conventional system must cool the air further to condense the water vapor out and reduce the humidity ratio. Then, the air must be reheated past its original outside temperature to the required indoor temperature. Only in December is there a heating load that does not also require dehumidification.

Cost Comparison of Open Cycle System to Conventional Vapor Compression System

In order to develop a feel for the costs of the Open Cycle System as compared to a conventional system, a rough estimate of system sizing follows. Please note that, because the TMY data are average data rather than maxima, the system sizing estimate is not fully accurate. An accurate sizing estimate would include load calculations for all 8760 hours in a year plus optimizations of the desiccant size versus the vapor compression size. However, since both the Open Cycle System and the Conventional System are evaluated in the same manner, the sizing estimate is adequate for comparison purposes.

The size of the desiccant part of the Open Cycle System has already been established. The vapor compression unit located downstream of the desiccant tower must meet the daily September sensible cooling load (756 MJ) plus the 971 MJ ($4387 - [7.98 \times 428]$) that the desiccant system does not meet, or a total of 1,727 MJ. (Note that the maximum sensible cooling load is in August but the total load is only 1,234 MJ.) This would require a 6 ton unit, costing about \$2,400. This should be added to the Open Cycle System cost of \$75,064 for a total of \$77,464. Assuming a SEER of 9 and electricity cost of \$0.05/kWh, the annual electrical cost of the vapor compression system is \$673.96/yr. The cost of pump power should be added to this cost.

The maximum load for the conventional system is 7,831 MJ/day. A 26 ton unit should meet this load, at a purchase price of \$10,400. Reheat equipment must be added to this cost. Assuming a SEER of 9 and electricity cost of \$0.05/kWh, the annual electrical cost of the conventional system is \$9,884.24/yr. The cost of reheat must be added to this cost.

If electric reheat is used, allowing a cost of \$1000 for the equipment, the additional electrical cost is \$13,657.18/yr, and the simple payback time is under 3 years. As noted previously, reheat energy may sometimes be obtained from sources such as waste condenser heat. Ignoring pump power in the Open Cycle system and reheat cost in the Conventional system, the simple payback time is under 8 years.

Cost Comparison of Open Cycle System to Alternate Closed Loop System

The original report contained a cost comparison of the Open Cycle System to an Alternate Closed Loop System. This comparison was made at the request of the DOE contractor. The cost comparison was meant to evaluate the cost effectiveness of desiccant dehumidification with the open cycle system as opposed to desiccant dehumidification with conventional flat plate collectors and an additional packing tower for regeneration. The alternate closed cycle system is not being proposed by FSEC, as believed by the reviewers of the original report. It is simply described, as requested, for cost comparison.

The cost estimate for the Open Cycle System was derived from the original cost estimate for a system with 404 m² collector area. The original estimate for the collector was drawn up by a Building Contractor certified by the State of Florida. The contractor's estimate was then broken down into both cost per square meter and cost per additional 32'x4' panel. Two additional panels were added to make the current estimated collector area of 428 m². The absorption dehumidifier tower cost is based on a cost breakdown given by Robison *et al.* for a similar system designed for a student center. Costs are in 1984 dollars and are adjusted to 1993 dollars using the Historical Cost Adjustment method as shown in R.S. Means and Engineering News Review. Other costs are based on 1993 Means Mechanical Cost Data. The cost breakdown is an estimate only, and is subject to actual costs at time of purchase, plus changes in design that may be necessary as a result of lab scale testing.

The ALTERNATE Closed Loop Flat Plate System is sized to meet a similar load as that of the Open Cycle System. The cost estimate for the ALTERNATE Closed Loop Flat Plate Collector System is based on 1993 Means Mechanical Cost Data. Some assumptions made are in favor of the Flat Plate System. For example, a very high collector efficiency is assumed, no collector support structure is included, and copper tubing length estimates are underestimated. The cost breakdown is an estimate only, and is subject to actual costs at time of purchase, plus changes in design that may be necessary as a result of lab scale testing.

As shown in the Open Cycle cost estimate, the installed cost for the Open Cycle System with collector area of 428 m² is \$75,064. As shown in the Alternate Closed Loop cost estimate, the

Break down of Cost Estimate for 404 m2 (32'L x 136'W) Collector for Open Cycle System

Item	Estimate basis	Orig. estim. 404.3 m2	\$/m2	\$/addtl panel (32'x4')	Current 428 m2
Framing	204 - 2x6x16's @ \$560/1000 LF	\$1,827.84	\$4.52	\$26.88	
Plywood baselayer	136- 4x8x1/2 shts, @ \$380/1000 SF	\$1,653.76	\$4.09	\$48.64	
Membrane	\$6347 SF @ \$1.80/SF	\$11,424.00	\$28.26	\$230.40	
Flexiglass	136-4x8x1/4 shts, @ \$85 ea.	\$11,560.00	\$28.59	\$340.00	
Caulk	102 tubes @ \$1.85ea	\$188.70	\$0.47	\$5.55	
Adhesives	4352 SF @ 6c/SF	\$261.12	\$0.65	\$7.68	
Splines	138- 2x2x16's @ \$373/1000LF	\$824.32	\$2.04	\$23.87	
Z-bars	408LF @ 10c/LF	\$40.80	\$0.10	\$1.20	
Bottom braces	26- 2x6x16's @ \$560/1000 LF	\$232.96	\$0.58	\$6.72	
Edge braces	17- 2x4x16's @ \$373.33/1000 LF	\$101.54	\$0.25	\$2.99	
Hurricane Clips	690 @ 25c ea.	\$172.50	\$0.43	\$5.07	
Roof support system	x = 28.5'; y = 17" to 16' (14.58'); 27.1deg	\$1,828.60	\$4.52	\$56.56	
Fasteners		\$393.25	\$0.97	\$11.57	
PVC distribution pipe	136 LF, 2" Sch 40 @ \$1/LF	\$136.00	\$0.34	\$4.00	
PVC collection pipe	136 LF, 4" Sch 40 @ \$2.25/LF	\$306.00	\$0.76	\$9.00	
Collector Air Ventilation:					
650 cfm fan (I.B. est.)	(x 404 m2 / 1920 ft2)	\$809.20	\$2.00	\$53.95	
air manifold (galv. steel)	5#/LF x 272' x \$.43	\$584.80	\$1.45	\$8.60	
Subtotal, Materials		\$32,345.39	\$80.00	\$842.68	
Subtotal, Assy Labor	60% matls(incl facil. tools, ...) & manifold	\$22,116.35	\$54.70	\$545.45	
Transport Allowance	simple guess, distance to site unknown	\$500.00			
Site Installation	Labor 4 workers, 2.5 days	\$800.00			
	Supervision	\$200.00			
	Crane @ \$75/hr, 2.5 days	\$1,500.00			
Total, roof-installed collector		\$57,461.74	\$142.13	\$1,388.12	\$60,238

Break down of Cost Estimate for BOS of Open Cycle System

Item	Estimate basis	Cost	
Absorption Dehumidifier Tower			
	Robison actual cost w/ Celdek, 708 l/s, 1984	\$960.34	
	Flowrate increase, 708 l/s to 1784 l/s	\$2,419.84	
	Means Historical Cost Index Adjustment	\$2,969.13	
	Assembly labor, @ 60% materials	\$1,781.48	\$4,751
Storage Tanks	2-1500 gal fiberglass (based on Means O&P)		\$7,075
Piping and fittings			\$2,500
Controls			\$500
Total, 428 m2 Open Cycle Collector System:			\$75,064

ALTERNATE Closed Loop Flat Plate Collector System

	Unit Materials	Unit Labor	Total
Collector Array (248 m ²) (to provide 2891 MJ/day in Oct, given 14.59 MJ/m ² day insolation, with 80% collector efficiency)			
127 units, 3'x7'	\$405.00	\$63.00	\$59,436.00
Roof clamps (not including support rigging to provide tilt on flat roof)	\$11.00	\$8.55	\$2,482.85
Cu tubing (per LF; 2.5" Type M, 35' to roof, 127 x 3', not incl. O&P)	\$6.50	\$6.15	\$5,705.15
Fittings			\$3,000.00
Urethane pipe insulation, 1" wall, 2" pipe	\$2.38	\$2.09	\$2,015.97
Valves (fill/drain, isolation, relief, check, tempering, riser, ...)	\$2,779.50	\$642.25	\$3,421.75
Expansion tank	\$90.00	\$20.80	\$110.80
Pressure gage, 0-60 psi	\$19.70	\$20.80	\$40.50
Support Rigging			\$2,000.00
Collector circulator pump (2.5", 3 hp, bronze impeller)	\$900.00	\$197.00	\$1,097.00
Heat exchanger (bronze, 10 gpm)	\$1,477.50	\$79.00	\$1,556.50
Site Installation (Transport, Supervision & Crane only)			\$2,200.00
Regenerator Tower	\$2,969.13	\$1,781.48	\$4,750.61
Absorption Dehumidifier Tower	\$2,969.13	\$1,781.48	\$4,750.61
Storage Tanks			\$7,075.00
Controls			\$500.00
Total, ALTERNATE Closed Loop, Flat Plate system			\$99,642.74

installed cost of the ALTERNATE Closed Loop Flat Plate Collector System is \$99,643. In spite of the cost reductions taken in favor of the ALTERNATE Closed Loop Flat Plate Collector System, it is evident that the Open Cycle System is a much more cost-effective way to dehumidify air with liquid desiccants.

References

Calcium Chloride, Technical and Engineering Service Bulletin No. 16, Allied Chemical Company, Industrial Chemicals Division, P.O.Box 1139R, Morristown, NJ 07960.

Collier, R.K., 1979. "The Analysis and Simulation of an Open Cycle Absorption Refrigeration System", Solar Energy, Vol. 23, pp. 357-366.

Conlan, D., Certified Building Contractor, State of Florida, Engineering Manager, Integri Homes, Inc.

Gandhidasan, P., 1983. "Thermal Performance Predictions and Sensitivity Analysis for a Parallel Flow Solar Regenerator", Journal of Solar Energy Engineering, Vol 105, pp. 224-228.

Ji, L-J, Wood, B.D, 1993. "Performance Enhancement Study of Solar Collector/Regenerator for Open-Cycle Liquid Desiccant Regeneration", "Solar 93" ASES Annual Conference Proceedings, S. Burley, M.E. Arden, eds. Washington, D.C., April 25-28, 1993.

Means Mechanical Cost Data, 1993. R.S. Means Co., Construction Publishers and Consultants, 100 Construction Plaza, P.O.Box 800, Kingston MA 02364

Means Construction Cost Data, 1993. R.S. Means Co., Construction Publishers and Consultants, 100 Construction Plaza, P.O.Box 800, Kingston MA 02364

Robison, H, Griffiths, W, and Arnas, O, 1984. "Liquid Desiccant/Vapor-Compressor Hybrid Air Conditioner And Energy Storage System For Load Management", "Solar in the Southeast", Conference Proceedings, North Carolina Solar Energy Association, Wrightsville Beach, NC, April 27-29, 1984.

AIM

Association for Information and Image Management

1100 Wayne Avenue, Suite 1100
Silver Spring, Maryland 20910

301/587-8202

Centimeter

Inches

MANUFACTURED TO AIM STANDARDS
BY APPLIED IMAGE, INC.

**DATE
FILMED**

6 / 2 / 95

END