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Experimental investigation of a liquid desiccant system for solar cooling and dehumidification

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Abstract

Growing demand for air conditioning in recent years has caused a significant increase in demand for primary energy resources. Solar-powered cooling is one of the environmentally-friendly techniques which may help alleviate the problem. A promising solar cooling method is through the use of a liquid desiccant system, where humidity is absorbed directly from the process air by direct contact with the desiccant. The desiccant is then regenerated, again in direct contact with an external air stream, by solar heat at relatively low temperatures. The liquid desiccant system has many potential advantages over other solar air conditioning systems and can provide a promising alternative to absorption or to solid desiccant systems.

Earlier work by the authors included theoretical simulations and preliminary experiments on the key components of the liquid desiccant system. The objective of the present study has been to construct a prototype system based on the knowledge gained, to monitor its performance, identify problems and carry out preliminary design optimization. A 16 kWt system was installed at the Energy Engineering Center at the Technion, in the Mediterranean city of Haifa. The system comprises a dehumidifier and a regenerator with their associated components operating together to dehumidify the fresh (ambient) air supply to a group of offices on the top floor of the building. LiCl-water is employed as the working fluid. The system is coupled to a solar collector field and employs two methods of storage – hot water and desiccant solution in the regenerated state. The performance of the system was monitored for five summer months under varying operating conditions. The paper describes the operation of the experimental system and presents the measured data and the calculated performance parameters.

Keywords: Solar cooling; Solar air conditioning; Dehumidification; Liquid desiccant; Open absorption

1. Introduction

Growing demand for air conditioning in recent years has caused a significant increase in demand for energy resources. Global warming, now an undisputed fact, has ushered in the air conditioning demand not only in hot and humid climates such as in Mediterranean countries, but also in European countries with no air conditioning tradition. Electric utilities have their peak loads in hot summer days, and are often barely capable of meeting the demand, with brown-out situations. With suitable technology, solar cooling can help alleviate, if not eliminate the problem. It is a good application for solar energy due to the fact that the greatest demand for air conditioning occurs during times of highest insolation (Grossman, 2002, 2004).

Conventional closed-cycle absorption chillers require heat source temperatures that are significantly higher than the temperatures of corresponding heat sinks. Thus, they have to be operated with high-grade heat extracted from natural gas, steam, concentrating solar collectors and the like. A promising alternative is the use of an open absorption system, where humidity is absorbed directly from the process air by direct contact with the absorbent. The

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absorbent is then regenerated, again in direct contact with an external air stream, at relatively low temperatures of the heat source. The entire operation takes place at atmospheric pressure, thus eliminating the need for vacuum vessels and the like.

Earlier work has been conducted on liquid desiccant systems for cooling and dehumidification, using solar energy for regeneration (Oberg and Goswami, 1998). A survey of early projects may be found in the review article by Grossman and Johannsen (1981). In several cases, direct regeneration of the desiccant in the sun has been considered, using a special type of collector. More recently, Wood and co-workers at Arizona State University (Ameel et al., 1995; Nelson and Wood, 1989a,b,c) have constructed and tested a full-scale liquid desiccant system, employing LiCl as well as a mixture of LiCl and $CaCl_2$ as liquid sorbents. Kessling (1997) studied a LiCl-water system operating at a large concentration difference between the strong and weak desiccant, to facilitate cold storage by means of a regenerated solution. Kakabaev et al. (1976, 1977, 1981) report on the operation of a large scale air conditioning system employing LiCl-water, where both direct regeneration in open collectors and cold storage in the form of regenerated solution have been attempted. Some investigators attempted the use of organic absorbents, such as Tri-Ethylene Glycol (TEG) to reduce the corrosion problem associated with the inorganic salts (Grossman and Johannsen, 1981). Noteworthy are also the liquid desiccant system analyses of Collier (1979), Haim et al. (1992), and Gandihdasan and Al-Farayedhi (1995).

The objective of the present project has been to design, build, test and evaluate a fully-instrumented liquid desiccant cooling system, to provide a demonstration and supply operating data under varying conditions. The system is capable of using as its source of power low-grade solar heat from low-cost flat plate collectors and has the potential to provide both cooling and dehumidification in variable ratios, as required by the load. While many of its characteristics are similar to those in earlier liquid desiccant systems, it possesses several advanced features such as automatic controls and the ability to store cooling capacity in regenerated desiccant. The significance of this work lies in the potential to provide solar-powered cooling, dehumidification and air conditioning for residential and commercial applications.

As part of the design phase of the liquid desiccant system, a complete system simulation was conducted, in order to predict trends and attempt a preliminary optimization. The lack of reliable data on heat and mass transfer coefficients in the absorption and desorption processes had been a serious impediment in earlier simulation studies to obtaining a good prediction of the system's performance. Particularly critical are the performances of the dehumidifier (absorber) and regenerator (desorber), forming the two key components of the liquid desiccant system. Such data has now been obtained through the experimental work described by Gommed et al. (2004). This made it possible to conduct an extensive parametric study of the overall system behavior (Gommed and Grossman, 2004). The computer simulation yielded the temperature and humidity of the air at the system outlets as well as heat duties of the various system components as functions of the specified conditions at inlets and other operating conditions.

2. Project description

The system under consideration is designed to air-condition a group of offices on the top floor of the Energy Research Center building at the Technion – Israel Institute of Technology (Haifa, Israel). The originally-selected conditioned space[1] consists of three offices, with one north-facing exterior wall each (including a window). The total floor area of the conditioned space is 35 m^2. The walls and roof are made of 8 in. (20 cm) prefabricated low-weight concrete, not insulated, with cement plaster. Each office serves two people and their computers; hence total occupancy is six persons.

The city of Haifa is an ideal site to test such a system. Located on the Mediterranean coast at 33° north latitude, it has the typical climate of Mediterranean cities. Outside summer conditions (typical for design) are 30 °C and 70% relative humidity. Room design conditions have been selected at 24 °C and 50% relative humidity.

A load calculation for the three typically staffed and equipped offices shows about 4.2 kW with a room sensible heat factor (RSHF) of 0.92. At 30 cfm (51 m^3/h) of ambient air per occupant (ASHRAE air quality recommendations), the additional fresh air–associated load is about 3.0 kW, most of which (2.4 kW) is latent. Thus, the total cooling capacity required is 7.2 kW, with a grand sensible heat factor (GSHF) of 0.62. The total supply air circulation needed (based on 12 air changes per hour) is 0.4 kg/s (720 cfm). The desired conditions of the supply air are 14.7 °C and 86% relative humidity.

The desiccant solution is regenerated by solar heat, supplied by flat-plate solar collectors of conventional design, of the type widely employed in Israel for domestic water heating, but with better than average quality to enable higher efficiency at high temperatures. The solar collectors and remaining parts of the system are located on the roof immediately above the top floor. Solar-heated water serves as the heat carrier. The option of heating the regenerated solution directly, by exposing it to the sun and to ambient air simultaneously, had been explored but found to be somewhat problematic. The advantages of the current option are simpler construction technology, simpler storage capability, dirt control and simpler ability in using an air-to-air heat exchanger for heat recovery. With the total latent heat load of 2.75 kW, the solar energy demand was estimated to be 4.77 kW. Assuming 10 h of continuous

[1] The size of the liquid desiccant system ultimately built was larger than originally planned. Cooling/dehumidification capacity was measured at about 16 kW (average).

operation daily, and taking a small safety factor, the solar collector area was selected at 20 m^2. Solution storage in the amount of 120 l of LiCl solution at 43% concentration and a 1000 l hot water tank added to the system make it possible to operate for a total of 4 h with no insolation – a typical situation in the summer during the morning hours.

3. Description of the liquid desiccant system

The liquid desiccant system is designed to serve as an open-cycle absorption system that can operate with low-grade solar heat. A schematic description of the final design version of the system is given in Fig. 1. The system consists of six major components: an air dehumidifier or absorber, a solution regenerator or desorber, two water-to-solution heat exchangers, a solution-to-solution heat exchanger, and an air-to-air heat exchanger. Arabic numerals indicate working fluids state points at specific locations. Air flow is represented by thick solid lines, solution flow by thin solid lines and water flow by thin dashed lines.

The dehumidifier (absorber) consists of a packed tower and operates in an adiabatic mode. Solution is pumped from the absorber pool at the bottom of the tower into the plate heat exchanger (state 7), where it is cooled by water from a cooling tower. The solution leaving the heat exchanger (state 8) then proceeds to the distributor at the top of the packing, from where it trickles down in counter-flow to the air stream and collects in the pool. Ambient air at state 13 entering the bottom of the absorber packed section is brought into contact with a concentrated absorbent solution entering the unit at state 8. Water vapor is removed from the air stream by being absorbed into the solution stream. The dehumidified warm air leaving the absorber passes through the blower and leaves the system toward the air-conditioned space at state 14. The blower controls the flow of air, while raising its temperature slightly. A controlled solution stream is transferred from the absorber pool to the regenerator, as shown (state 11). The return (pumped) stream from the regenerator (10) goes directly into the absorber pool.

As evident, the regenerator (desorber) device is very similar to the dehumidifier, and so are the flow system and associated components. The solution is heated in the liquid-to-liquid heat exchanger by solar-heated water (states 1–2). Ambient air is pre-heated in the air-to-air rotary heat exchanger (H.X.) by recovering heat from the

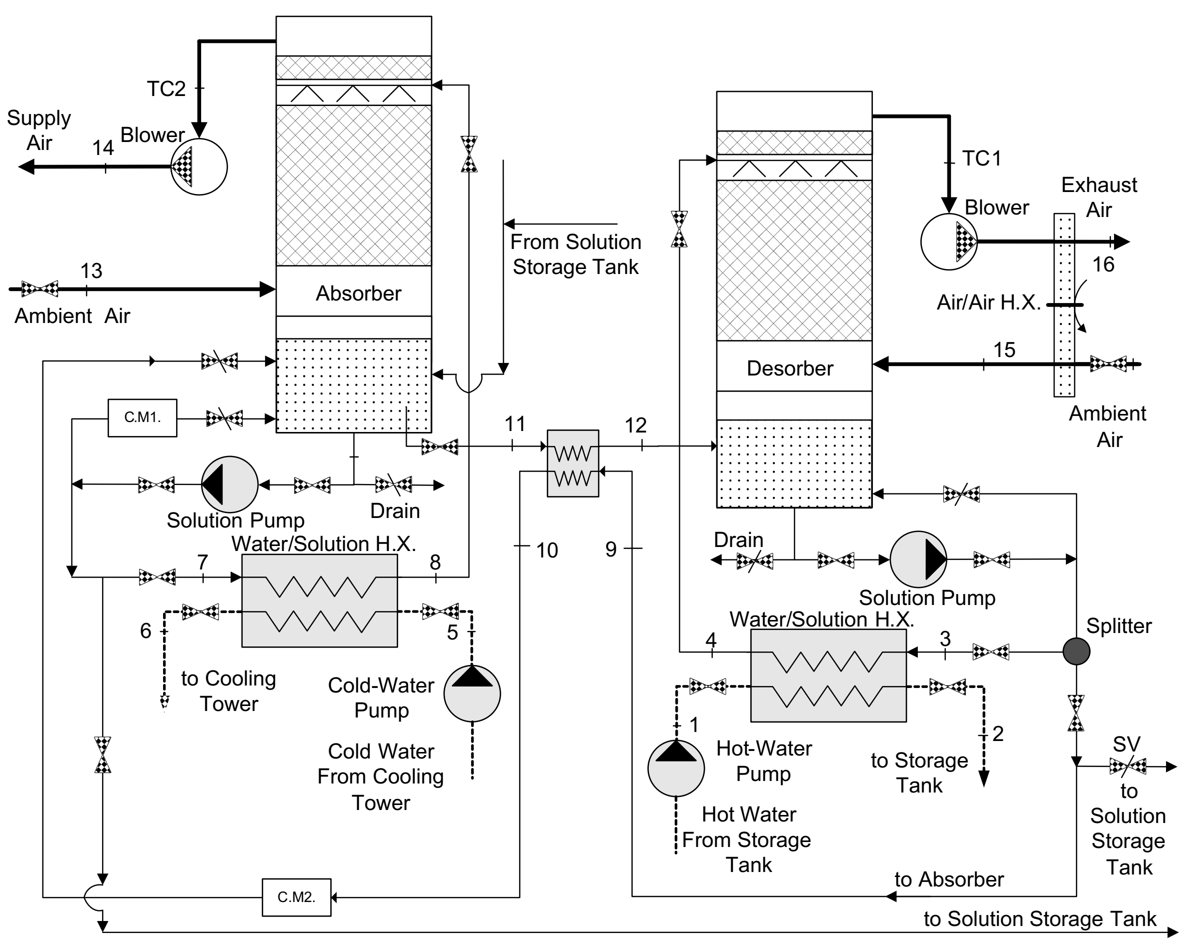

Fig. 1. Schematic description of the liquid desiccant system: Solid thick lines indicate air flow; solid thin lines indicate solution flow; dotted thin lines indicate water flow.

exhaust air leaving the desorber. After pre-heating, the air stream (state 15) enters the desorber where it serves to re-concentrate the solution. The exhaust air leaves the desorber, passing through the blower, then pre-heats the entering air stream and is rejected to the environment. The solution-to-solution heat exchanger facilitates pre-heating of the weak solution leaving the dehumidifier (states 11–12) and recovers heat from the hot strong solution leaving the regenerator (states 9–10).

The above brief description of the system already reveals a number of advantages of this system over conventional absorption heat pump cycles: (1) The number of main components is reduced by one by transferring condensation of the refrigerant from a condenser to the environment. (2) Capital-intensive pressure-sealed units are avoided as the whole system operates at atmospheric pressure. (3) The amount of refrigerant (water) evaporated in the regenerator does not require an evaporator elsewhere in the system, thus providing greater flexibility. (4) Efficient utilization of very low heat source temperatures is possible.

Fig. 2 is a photograph of the liquid desiccant system, showing its main components. The conditioned space is located to the right of the system, and the air ducts connecting the two are visible in the picture. The solar collector field is visible from its back. Other components forming part of the system and not shown in the picture are the cooling tower and the solution storage tank.

In the overall setup, the liquid desiccant system is connected in a flow arrangement allowing storage of concentrated solution and a capability to work in three different modes. The first is a manual mode used for testing individual components of the system. The other two modes are automatic, as may be selected by the user. One automatic mode is for full operation of the system (FOP) and the second is for regeneration only (REG). In the automatic FOP mode, all system components operate, including the solution storage circuit. In this case, the absorber solution pump may supply the dehumidifier with solution from both the absorber pool and from the solution storage tank, in parallel. Thus, dehumidification can continue independent of regeneration. If the solar collectors cannot supply water at sufficiently high temperature, or if the concentration of the solution in the storage tank and/or the regenerator pool rises above a set limit, the regeneration side of the system will shut down for a certain time. In the REG mode, only the regenerator (desorber) side of the system operates. The system shuts down automatically when the concentration of the solution in the storage tank reaches a certain high value or when the temperature of the hot water drops below a certain limit. At the end of days of high insolation, when a large amount of solar heat has been collected and stored in hot water, the user can set the system to operate in the automatic REG mode before leaving the site.

Fig. 2. Photograph of the liquid desiccant system: 1 – absorber/dehumidifier; 2 – desorber/regenerator; 3 – air ducts; 4 – fan; 5 – rotary air/air heat exchanger; 6 – control cabinet; 7 – solar collector field; 8 – hot water storage tank.

4. System monitoring

The liquid desiccant system has been fully operational since April 2003, and its operation was monitored throughout the summer of 2003 (27 May through November 6). Numerous experimental runs were conducted prior to this period, to test various components and instruments and to make final adjustments in the control system. As it turned out, weather during the spring of 2003 in Haifa was relatively dry, and the desiccant system could not perform in a meaningful way until the end of May.

The complete set of data collected during the monitoring period contains 30955 lines, each representing a record of the system's performance at a particular time. Data was taken at 1 min intervals. Each record contains 28 instrument readings of temperatures, humidities, flowrates, pressures, etc. at various parts of the system. A computerized data acquisition system was used, showing the location and type of the various sensors and measuring points in the system. The temperature uncertainty of the PT100 sensors is 0.2 °C and the Relative Humidity uncertainty is 2%. Fig. 3 is a screen view of the data acquisition system with the main components discussed earlier. A logbook of events was maintained during the monitoring period and all irregular events were listed. These include electric power interruptions, failed equipment etc.

For the purpose of the following discussion, three typical days were selected from the large volume of data – one for July, one for August and one for September. These are days during which the system operated rather smoothly, representing what may be expected ultimately after all the various operational and control problems (leakages, overflows, control problems causing overheating, etc.) have been fixed. Examination of these data has shown consistent similarity among the three days. Characteristic measured data representing the total monitoring period is given in Table 1. In view of the wealth of data, the following discussion concentrates on what the authors have considered most meaningful in terms of the system operation.

Fig. 4 illustrates the variation of the absolute air humidity as a function of time during a typical day (21 August 2003): ambient air (*Win*), desorber outlet air (*Wout1*) and absorber outlet air (*Wout2*). Note that the absorber outlet air humidity is also that of the supply air to the conditioned space. As evident, the outside air humidity has remained approximately constant during the whole day, at about 16 g/kg, with a slight increase toward the evening. The absorber outlet humidity was equal to that of the outside air when operation began at about 10:00, and was reduced to about 8 g/kg within 20 min. The machine was able to keep this humidity steady throughout the day. The desorber outlet air humidity, which is the control parameter,

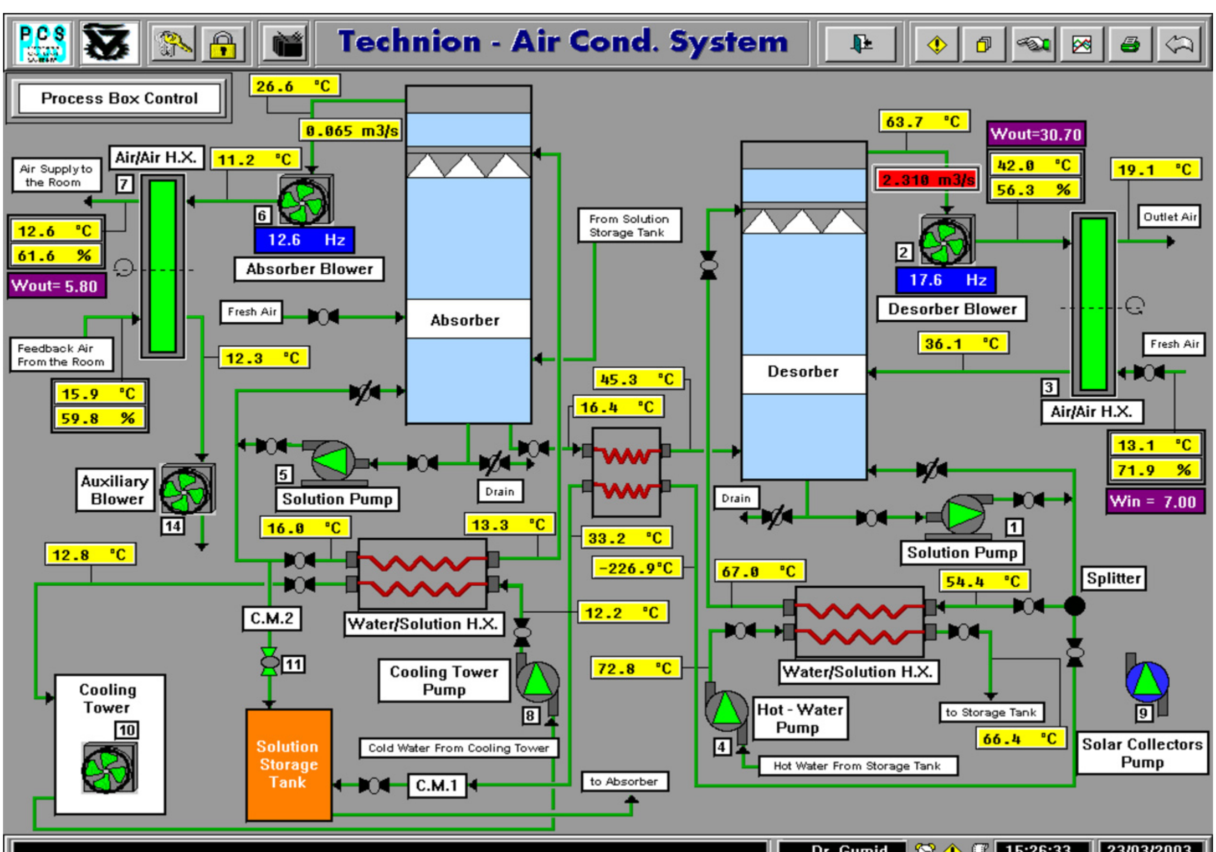

Fig. 3. Screen view of the data acquisition system. An additional rotary heat exchanger, not shown in Fig. 1, is included on the absorber side. (Note that numbers in this Figure correspond to the various components, not to streams, and hence bear no relation to the state point numbers in Fig. 1.)

Table 1
Characteristic measured data representing the monitoring period

Heating water temperature	65–100 °C
Heating water flow rate, $\dot{m}_{hw}$	0.24 kg/s
Cooling water temperature	22–27 °C
Cooling water flow rate, $\dot{m}_{cw}$	0.5 kg/s
Air flow rate through desorber, $\dot{m}_{da}$	0.1–0.32 m^3/s
Air flow rate through absorber, $\dot{m}_{aa}$	0.1–0.40 m^3/s
Pressure drop through desorber/absorber towers	180 Pa
Pressure drop through entire duct pass including air/air heat exchanger	120 Pa
Total heat supplied to desorber	up to 20 kW
Maximum solution concentration	43%
Minimum DPT	4 °C
Air flow rate through desorber/absorber fans (m^3/s)	0.008 × Freq. (Hz)
Average air density through absorber	1.15 kg/m^3
Average air density through desorber	1.05 kg/m^3
Desorber fan power @ 32 Hz/40 Hz	280/500 W
Absorber fan power @ 20 Hz/40 Hz	60/350 W
Air/Air Heat exchanger power	80 W
Solar collectors circulation pump power	50 W
Solution pump power on absorber side	180 W
Solution pump power on desorber side	250 W
Hot water pump power	290 W
Cooling tower fan power	520 W
Cooling tower (cold water) pump power	400 W
Total parasitic power, W_{par}	2700 W

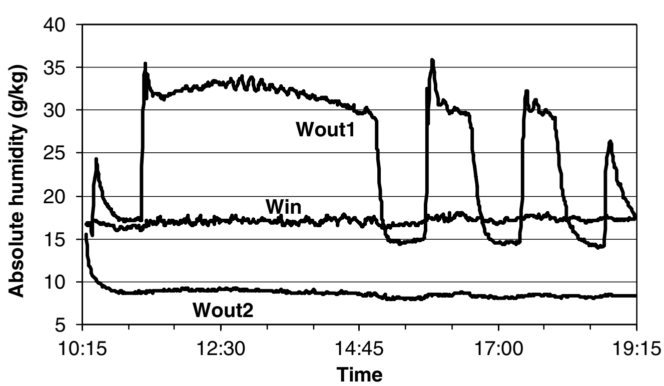

Fig. 4. Absolute humidities (g water/kg dry air) of outside air (*Win*), desorber outlet air (*Wout1*) and absorber outlet air (*Wout2*) as functions of time during a typical day (21 August 2003).

shows considerable variations. The control system shut the desorber off about 5 min after the start of operation, when the temperature of the hot water did not reach the minimum certain limit; it turned the desorber back on and then turned it off when its outlet air humidity dropped below 30 g/kg; this is an indication that the solution in the desorber becomes too concentrated in LiCl, which may lead to crystallization. The same sequence continues several times during the day. The on-off cycling of the desorber makes it possible to maintain the supply air humidity at the desired and steady values.

Fig. 5 shows the specific humidity of the dehumidified air as a function of that of the ambient inlet air. The figure contains 390 data points representing the three days, and

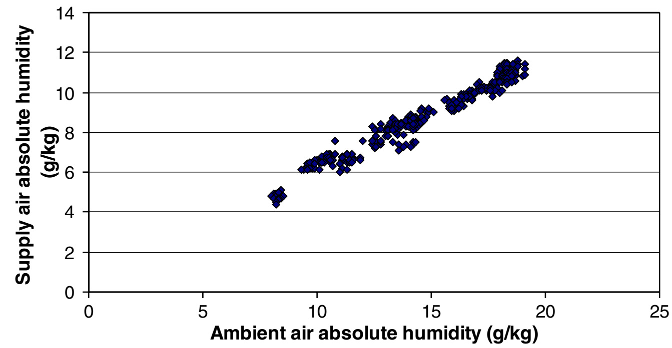

Fig. 5. Supply air absolute humidity (g water/kg dry air) as function of ambient air absolute humidity (g water/kg dry air).

shows a clear trend, indicating an almost linear relation between the two humidities. This trend is in agreement with the one obtained from the simulation study (Gommed and Grossman, 2004). Applying a least-square analysis to the measured humidity data indicates that the average supply air humidity content at the absorber outlet (*Wout2*) could be estimated by the following empirical formula:

$$Wout2 \approx 0.6Win \tag{1}$$

In view of the ambiguity often encountered in the literature regarding the role of parasitic losses (from fans, pumps and anything else that is not an integral part of the thermal system) in the performance of desiccant systems (and other heat pumps and HVAC systems), four types of coefficient of performance have been introduced here. In each, the numerator contains the useful cooling/dehumidification produced, whereas the denominator differs, as follows:

(1) *COP1* is a strictly thermal COP, not including parasitic losses; the denominator contains the thermal energy supplied, in this case from the solar collectors.
(2) *COP2* includes in the denominator the sum of (solar) heat input and parasitic losses.
(3) *COP3* is the same as *COP2*, but the parasitic losses in the denominator are converted to their equivalent heat value, assuming power plant electricity generation at 40% efficiency.
(4) *COP4* is the ratio of latent heat removed from the process air to the electric power equivalent of the input energy, consisting of (solar) heat and parasitic power.

Fig. 6 is of equal interest, describing the Thermal COP as a function of the ambient air specific humidity. This COP is calculated from the measured data (see below) and averages about 0.81, almost independent of the humidity of the entering air. Studies of similar liquid desiccant system report thermal COP's in the range 0.6–0.8.

Fig. 7 describes the four COP's as functions of time, for the data of 21 August 03, discussed previously. Clearly, *COP4* shows the highest values and *COP3* the lowest,

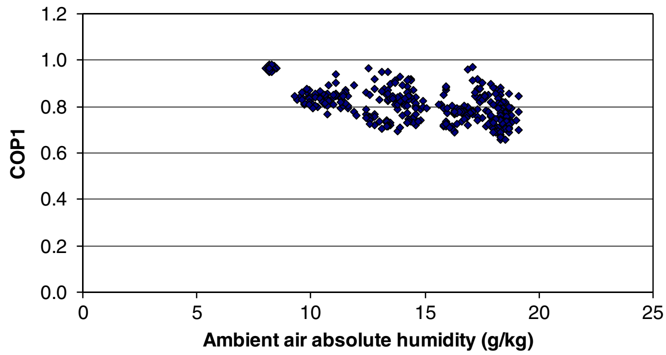

Fig. 6. Thermal COP (*COP1*) as function of ambient air absolute humidity (g water/kg dry air).

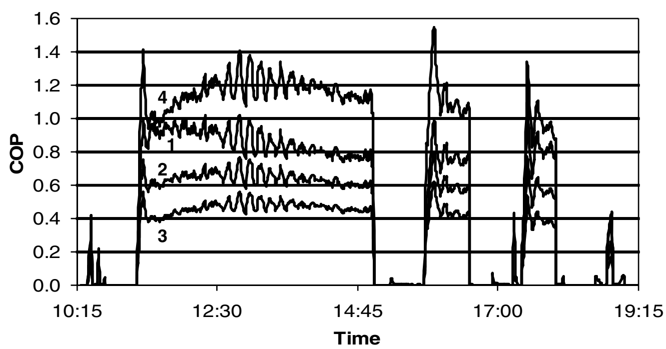

Fig. 7. Four COP's of the system as functions of time during a typical day – 21 August 2003. 1 – *COP1*, 2 – *COP2*, 3 – *COP3*, 4 – *COP4* (refer to text for exact definitions).

based on the above definitions. Except for local fluctuations, all COP's seem relatively steady at periods of operation. The on–off operation of the desorber is clearly reflected in Fig. 7.

The measured data as presented in Table 1 were employed to calculate not only the COP's but also other system parameters of interest. A summary of the calculated data representing the total monitoring period is given in Table 2. The heat balance errors for the absorber and desorber are rather low (although not insignificant) considering the complexity of the system at hand. The heat and mass transfer coefficients, estimated from the data using the computer code ABSIM (Grossman and Zaltash, 2001) are one of the most important results of the study, in view of the lack of reliable data on these coefficients in the literature. With this information, further systems may be designed.

Table 2
Summary of calculated data representing the total monitoring period

Average heat balance error (desorber side)	10%
Average heat balance error (absorber side)	20%
Heat transfer effectiveness:	
Desorber solution/water heat exchanger	60%
Absorber solution/water heat exchanger	40%
Solution/solution heat exchanger	51%
Desorber side air/air heat exchanger	76%
Mass transfer coefficients:	
Solution-interface	0.3 kg/s (0.0067 kg/m^2 s)
Air-interface	0.9 kg/s (0.0200 kg/m^2 s)
Average thermal *COP1*	0.81

5. Conclusion

The objective of this project has been to construct a solar-driven liquid desiccant system for cooling, dehumidification and air conditioning – to test the concept, identify problems, carry out preliminary design optimization and measure performance. The prototype system, built at the Technion campus in Haifa, is designed to air-condition a group of offices on the top floor of the Energy Engineering Center. The design process involved initial measurements to determine unknown parameters, along with extensive performance simulations. With many unknown factors, the initial design of the system underwent several changes during the development period. The characteristic performance of individual components, analyzed theoretically in earlier simulation, was studied experimentally. Measurements have provided much-needed realistic data about heat and mass transfer coefficients. Important information was obtained about practical design aspects of the key components – dehumidifier and regenerator – as well as quantitative data about their performance. The final prototype, including controls, has been fully operational since April 2003. The system functioned well, with 16 kW (average) dehumidification capacity, and its performance was monitored throughout the summer of 2003 – from the end of May till the beginning of November. The data analysis indicates a thermal COP of about 0.8, with parasitic losses on the order of 10%.

The COP calculations performed on the monitoring data have yielded satisfactory results, particularly with regard to the thermal COP. By more elaborate design in the future, it is anticipated that parasitic losses could be minimized and better overall COP's could be achieved.

Acknowledgement

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