

Experimental Investigation of a LiCl-Water Open Absorption System for Cooling and Dehumidification

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In earlier work, a novel open absorption cycle was proposed, capable of producing both cooling and dehumidification for air conditioning, utilizing low-grade heat. The system, referred to as DER (Dehumidifier-Evaporator-Regenerator), uses ambient air in conjunction with an absorbent solution; the air is dehumidified and then employed to produce chilled water in an evaporative cooler. Alternatively, a portion of the dehumidified air may be used directly for air conditioning purposes. The system thus has the potential to provide both cooling and dehumidification in variable ratios, as required by the load. Computer simulations and theoretical investigations were carried out to analyze and predict the performance of the system. The objective of the present study has been to construct a laboratory system to test the concept, identify problems and carry out preliminary design optimization. The characteristic performance of individual components, analyzed theoretically in the simulation, was studied experimentally. Measurements have provided much-needed realistic data about heat and mass transfer coefficients. The performance of the system has been studied under varying operating conditions. The paper describes the operation of the experimental system and presents the measured data and the resulting transfer coefficients. [DOI: 10.1115/1.1643075]

Keywords: Dehumidifier, Evaporator, Regenerator, Absorber, Cooling, Air Conditioning

Introduction

This paper is concerned with an experimental system constructed to test the concept of a novel open-absorption heat pump, referred to as DER (Dehumidifier-Evaporator-Regenerator). The DER cycle has been identified as a basis for promising cooling systems capable of exploiting low-grade heat sources such as solar heat, waste heat and the like as its source of power. The cycle may be considered a combination of conventional evaporative cooling system operating with ambient air and conventional absorption heat pump cycle driven by low-grade heat; it is regarded as a powerful alternative to both of them. The system configuration, proposed in the literature some 60 years ago [1], was claimed to allow for operation at extremely small temperature differences between the heat source and heat sink, but without any support by experiments or calculations, or details about its performance characteristics. It has for the first time been subject to a detailed analysis regarding its potential and performance characteristics in studies by Hellmann and Grossman [2,3]. The immediate application of the system is for cooling and air conditioning, but other applications in the industrial area may be envisaged.

A schematic description of the DER cycle is given in Fig. 1. It consists of six major components, marked by roman numerals: an indirect contact evaporative cooler (I), an air dehumidifier or absorber (II), a solution regenerator or desorber (III), two air/air heat exchangers (IV,V) and a solution/solution heat exchanger (VI). Ambient air entering the absorber at state 7 is brought into contact with a strong (concentrated) absorbent solution entering the unit at state 16 and leaving it at state 17. Water vapor is removed from the air stream by being absorbed into the solution stream. The heat of condensation and dilution Q_A released during this absorption process is rejected to a stream of cooling water or ambient air

(states 3-4). Dehumidified air leaving the absorber at state 8 is drawn through an air/air heat exchanger (IV) into the evaporative cooler. By evaporating externally-supplied water into the air stream the heat Q_E is removed from the chilled medium (states 1-2) at a low temperature. The air leaves the evaporative cooler at state 10 humid, but with a lower temperature than at its inlet, and passes again through the heat exchanger (IV) where it provides pre-cooling for the air stream 8-9 in order to allow for a low cooling temperature. After being pre-heated in the second air/air heat exchanger (V) by exhaust air, the air stream enters the regenerator at state 12 where it serves to re-concentrate the weak absorbent solution (state 18). The heat of desorption Q_D required to remove water vapor from the solution is provided by an external heat source, e.g. solar-heated water (states 5-6). The exhaust air leaves the regenerator (state 13), pre-heats the air stream 11-12 in the air/air heat exchanger (V) and is rejected into the environment. The solution/solution heat exchanger (VI) facilitates pre-heating of the weak solution (states 17,18) and pre-cooling of the strong solution (states 15,16).

The advantages of open-cycle DER absorption chillers over closed-cycle systems are evident from the above description: (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 can be operated at atmospheric pressure. (3) Efficient utilization of very low heat source temperatures is possible. The open-cycle DER chiller can be operated with any heat source temperature only slightly above the heat rejection temperature. This is particularly favorable for the efficient utilization of heat sources with varying temperatures such as solar heat from flat-plate collectors. Compared to conventional evaporative cooling, the main advantage lies in a significant reduction of the cooling limit even below the wet bulb temperature of the ambient air. Alternate configurations of the system make it possible to produce both chilled water and cold and dehumidified air in variable proportions, as required by the load.

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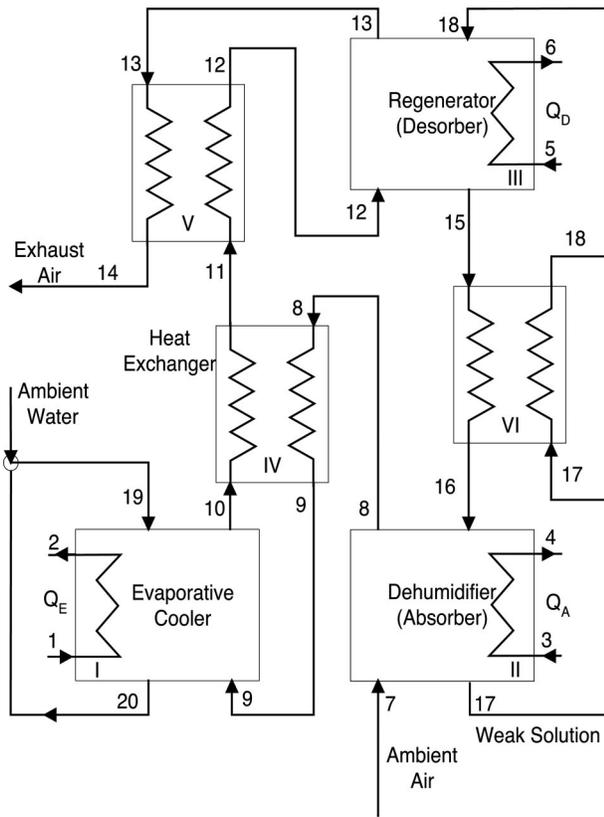


Fig. 1 Schematic description of the DER system

A thorough performance analysis of the DER absorption cycle was performed [2,3] in order to evaluate its potential quantitatively and obtain a fundamental understanding of its characteristic advantages and disadvantages. An aqueous solution of LiBr was employed as absorbent in the simulation; LiCl was also investigated as an alternative. The computer code ABSIM [4] developed

for modular simulation of closed-cycle absorption systems with varying cycle configurations and with different working fluids, was employed in the study, with some modifications and adaptations. The results have proven the advantages listed above and motivate further research on this system.

In conducting the simulation, estimates had to be made of some parameters crucial to predicting the performance—mainly heat and mass transfer coefficients. The primary objective of this work was to conduct experiments for generating realistic data on these coefficients. Other objectives of the experiment were to test the concept, identify possible problems and study the characteristic performances of individual components analyzed theoretically in the simulation.

Experimental System Description

In order to proceed systematically with the development of the DER, an open absorption system having most of the DER components was constructed and tested. The chilled-water-producing evaporative cooler and its associated air-to-air heat exchanger (I and IV in Fig. 1) have been omitted here for simplicity, as it was judged that adding them at a later stage would be relatively straightforward. As such, this experimental system represents an open-cycle absorption device that can be operated with low-grade heat. The system employs an aqueous solution of lithium chloride as the absorbent, in direct contact with the air. A schematic description of the final design version of the system is shown in Fig. 2. It consists of five major components: an air dehumidifier or absorber, a solution regenerator or desorber, two water-to-solution heat exchangers, and an air-to-air heat exchanger. Arabic numerals indicate working fluid state points at specific locations. Air flow is represented by thick solid lines, solution flow by thin solid lines and water flow by dashed lines. It should be clearly emphasized that the system in Fig. 2 is not the same as the one in Fig. 1, and the state point numbers differ from one to the other.

The dehumidifier consists of a packed tower and operates in an adiabatic mode, for reasons which will be explained further below. Ambient air at state 14 enters the bottom of the dehumidifier packed section and is brought into contact with a concentrated absorbent solution entering the unit at the top (state 11). Water vapor is thus removed from the air stream by being absorbed into

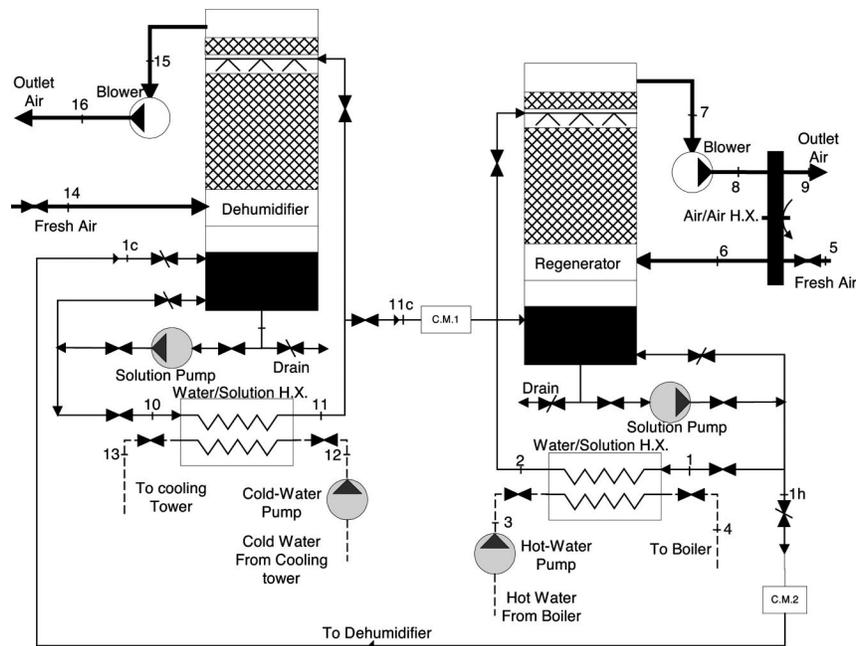


Fig. 2 Schematic description of the experimental system

the solution stream. The dehumidified and warm air leaving the absorber at state 15 passes through the blower and leaves the system (intended for the air-conditioned space or for an evaporative cooler) at state 16. The blower controls the flow of air, while raising its temperature slightly. Solution is pumped from the dehumidifier pool at the bottom of the tower into the plate heat exchanger (state 10), where it is cooled by water from a cooling tower. The solution leaving the heat exchanger (state 11) then proceeds to the distributor at the top of the packing, from where it trickles down in counterflow to the air stream and collects in the pool. A separate stream of solution from the dehumidifier pool is split off to go to the regenerator, as shown (state 11c). The return stream from the regenerator (1h-1c) goes directly into the dehumidifier pool. The heat of condensation and dilution, Q_A , released during the absorption process is hence rejected to the streams of solution and dehumidified air. The portion of heat absorbed by the solution is finally removed to the cooling water (12-13) by pumping the solution from the dehumidifier pool (state 10) through the solution-to-cold water heat exchanger.

As evident, the regenerator device is very similar to the dehumidifier, and so is the flow system and associated components. The solution is heated in the liquid-to-liquid heat exchanger by hot water from the boiler, simulating the external source of solar heat (states 3-4). Ambient air at state 5 is pre-heated in the air-to-air heat exchanger by recovering heat from the exhaust air leaving the regenerator (state 8). After pre-heating, the air stream (state 6) enters the regenerator at the bottom and flows up, in counterflow to the solution entering the top at state 2. This air thus serves to re-concentrate the solution which trickles down to the pool at state 1. The exhaust air leaves the regenerator (state 7), passing through the blower, pre-heats the entering air stream and is rejected to the environment.

Design Considerations for the System Units

The absorber (air dehumidifier) and the desorber (solution regenerator) are the two key components in the system, and their design has been the most challenging task in developing the experimental system. Both are based on the same principle of operation (counterflow heat and mass transfer in a packed tower) and involve the same types of streams of air and liquid desiccant. The main difference between them is their opposite directions of heat and mass transfer. Taking this into consideration, two prototypes of one unit that could serve as either absorber or desorber have been designed and built. This approach saved much time and resources in the design stage. Following the experience gained by testing the two aforementioned prototypes, the third prototype that included the two major units (absorber and desorber) has been designed and built. These two units were connected together and operated as a complete system for long test periods until steady operation conditions were reached. They formed the main part of the experimental system shown in Fig. 2.

A variety of aqueous salt solutions have been proposed as possible working fluids. The three most common desiccants, employed in the majority of the studies on open-cycle systems are lithium bromide (LiBr)/water, lithium chloride (LiCl)/water, and calcium chloride (CaCl_2)/water. Their thermo-physical properties that are required during cycle simulation are available in the literature. Prices for CaCl_2 are a factor of 20 to 60 below the prices of LiCl, while LiBr costs about twice as much as LiCl. LiBr/water is often chosen due to its superior thermodynamic properties leading to better cycle performance. LiCl/water is said to have a better long-term stability than LiBr/water in the regeneration process under atmospheric conditions, but no quantitative information on the difference and on the possible influence of pollutants is presented in the literature. In our study LiCl/water was chosen. It is preferred over LiBr/water for reasons of cost and long-term stability, while it is preferred over CaCl_2 /water for its better hygroscopic properties leading to better cycle performance.

Heat and mass exchangers of various types are employed abundantly in the chemical industry. The chemical engineering source books of McCabe et al. [5], Treybal [6] and Sherwood and Pigford [7] were consulted for general information about existing types of absorption columns and their designs. Examples of commercial open absorption systems are available. Kessling [8] surveys open-cycle absorption systems from the point of view of their energy storage capacity. The idea is to store the regenerated, highly concentrated salt solution in a tank and to employ it at a later time when no regeneration heat is available. This form of energy storage only works if the concentration difference between the solution at the inlet and at the outlet of the regenerator is considerable. Therefore, a low-flowrate solution stream has to be utilized and cooling is required in the absorber. Besides discussing theoretically the characteristics and models of cooled and adiabatic absorbers, as well as their energy storage potential, Kessling [8] also describes experiments on a plastic absorber with internal cooling, which he designed especially for low-flowrate solution streams.

A major initial design consideration was whether the LiCl solution should be cooled continuously inside the absorber during absorption, or outside, prior to the absorption process. The same question pertains to heating in the regenerator. If the solution is cooled continuously during absorption, a more effective absorption process can be achieved, since the heat of absorption is transferred to the cooling stream as it is generated. This is the way it is done in closed-cycle absorption systems. These considerations together with the findings of Kessling [8], who describes the design of a cooled dehumidifier, led to the decision in favor of an absorber with an integrated heat exchanger in our first prototype. While the thermodynamic considerations were correct, this concept did not function well for practical reasons, mainly due to the presence of air that differentiates the present system from a closed absorption system. Our first heat exchanger was a bundle of copper tubes mounted inside the column, with solution distributed over its top and dripping down from one row of tubes to the next, and air flowing upward through the spaces between the tubes. This copper tube bundle suffered from severe corrosion and was replaced by a similar bundle made of polypropylene tubes. With this new heat exchanger corrosion was no longer a problem, but heat transfer was degraded considerably due to inadequate wetting of the tubes.

On the other hand, an absorber without internal cooling is much easier to design and maintain. A typical adiabatic absorber consists of a tower filled with either structured or loose packing material over which the falling solution is distributed. It was learned that dehumidifiers currently employed by leading manufacturers of liquid desiccant systems do not employ internal cooling heat exchangers, while they used to be employed in earlier models. The second prototype in our study was hence adiabatic, which ultimately proved to be the favorable design concept. The heat exchanger for the unit serving alternately as absorber or desorber was submerged in the solution collection pool located at the bottom of the tower. This design did away with the problem of solution wetting at the heat transfer surface. The corrosion problems were less severe than with the first prototype, since the heat exchanger material was no longer exposed to the flow of air, but were still there due to air dissolved in the solution surrounding the submerged heat exchanger. Another problem was that under the desorber operation mode, the submerged heat exchanger transferred heat to the entire amount of solution in the pool and the solution temperature in the pool was relatively high, leading to heat losses to the regeneration air and to the surroundings.

To correct these problems, a different design (third prototype) was developed. Here, two external (non-submerged) plate heat exchangers were used in order to transfer heat from the hot water and cold water to the solution at the desorber and absorber, respectively. Each of the two heat exchangers is made from 20 plates of 316 stainless steel. These heat exchangers are more re-

sistive to corrosion than the original copper tube bundles, and have high heat transfer effectiveness (80-85% near the design point) and low pressure drop. The solution heat exchanger on the regenerator side was connected in such a way (Fig. 2) that enables heat transfer only to the portion of solution going up the desorber tower for regeneration.

To reach low air humidity in the adiabatic absorber design, high flow rate of solution, pre-cooled by means of an external heat exchanger at the entrance of the absorber, is required. While high solution flow rates can solve partially the wetting problem of the packing material as well as increase the heat and mass transfer rates, such high flow rates at the same time increase the circulation losses in the cycle. To solve this problem, splitters had to be added at the absorber and desorber solution outlets, enabling circulation at high flowrate through the absorber and regenerator, while transferring a low flowrate between the two. The arrangement is illustrated in Fig. 2. The splitter at the absorber outlet controls the flow rate of weak solution from the absorber to the desorber. The splitter at the desorber outlet controls the flow rate of strong solution from the desorber to the absorber. This arrangement was employed in our third prototype.

In an open system of the type considered here, attention must be given to the prevention of solution carryover. Experiments have shown that the LiCl solution should not be sprayed, but rather dripped over the packing, with the drops large enough not to be carried away by the air stream. Different solution distributor designs were experimented with; the final design involved a horizontal header with distribution holes drilled into it, which gave satisfactory results. In addition, mist eliminators were incorporated in each of the packed towers, above the solution distributors, as shown in Fig. 2. These consisted of packing material which intercepts and removes any remaining liquid from the leaving air stream.

In the early prototypes, it was found that most of the heat used in the desorber for regeneration was carried out by the air stream. The addition of an air-to-air heat exchanger in the third prototype for heat recovery in the desorber side contributed significantly to the correction of this problem.

Instrumentation and Measurements

Keeping measurements reliability in mind, two electronic humidity meters (Testo 635) were used to provide precise humidity measurements with improved dynamic response. The temperatures at the various locations in the system were measured by type T thermocouples with 22-bit data acquisition system. Air quantities were measured using precise analog differential pressure gauges. This type of measurement enables to evaluate the velocity and flow rate from the static and dynamic pressures. Also, it helps evaluate the pressure losses through the towers for calculating the parasitic power consumption. To ensure the reliability of the air flow measurements, an insertion mass flow transmitter (Kurz model 454FT) was used to measure the flow rate of air in parallel with the flow measurements by means of pressure differentials. The temperature, density, and flow rate of the weak and strong solution exchanged between the towers were measured by high accuracy Coriolis-type mass flow meters (ABB K-flow Inc.). Two meters were installed in the two lines for solution exchange between the two towers (at points 1h and 11c in Fig. 2). Knowing the temperature and density of the solution enables the evaluation of solution concentration and makes it possible to perform a computerized simulation for calculation of the heat and mass transfer coefficients.

Measured quantities included as follows (state point numbers are indicated with reference to Fig. 2): Solution temperatures at regenerator and dehumidifier outlet and inlet (1,2,10,11); heating and cooling water flowrates (3,12) and temperatures at respective heat exchanger inlet and outlet (3,4,12,13); relative humidity and dry bulb temperature of fresh air and of exhaust (outlet) air at inlets and outlets of air-to-air heat exchanger (5,6,8,9), and of

exhaust air from regenerator (7); relative humidity and dry bulb temperature of fresh air and of exhaust (outlet) air at dehumidifier inlet and outlet (14,15) and of exhaust air from blower (16); K-flow temperature, flowrate and density reading at dehumidifier and regenerator outlets (11c, 1h); pressure differential (total minus static) at regenerator and dehumidifier outlets (7,15). Table 1 lists several data sets taken during a typical day of measurements. These measured quantities enabled calculation of solution concentration at regenerator and dehumidifier outlets (1,10); solution enthalpy at regenerator and dehumidifier inlets and outlets (2,1,11,10); water content in air at regenerator and dehumidifier inlets and outlets (5,9,14,16); air densities, velocities and flowrates in regenerator and dehumidifier.

The pressure drop (pressure losses) through the regenerator and dehumidifier units was evaluated as the pressure difference between the total outside pressure (the ambient pressure) and the total pressure at the unit outlet. Typical data is shown below:

Average air flow rate through absorber:	$F_{Aa} = 0.315$ kg/sec
Average air flow rate through desorber:	$F_{Da} = 0.16$ kg/sec
Pressure drop through absorber:	$\Delta P_{Aloss} = 325$ Pa.
Pressure drop through desorber:	$\Delta P_{Dloss} = 142$ Pa.

The Fan power (parasitic power) needed to overcome these pressure drops (assuming a realistic fan efficiency of 70% and an electric motor efficiency of 80%) are estimated at 160 Watts and 35 Watts for the absorber and desorber units, respectively. During the design phase of the experimental system, with the pressure losses still unknown, oversize fans of 400 Watts and 100 Watts respectively were selected in order to have enough leeway in the experiments.

From heat balances based on measured data on the overall system and the desorber side, the error was found to be within 10% of the total supplied heat. On the absorber side the error was within 20% of the absorber power. The total supplied heat by the hot water boiler was approximately 8 kW. These errors are quite acceptable for this kind of measurements. The three major error sources were: (1) temperature measurement accuracy of $\pm 0.5^\circ\text{C}$; (2) heat losses to the surroundings; (3) the process was not completely at steady state conditions.

The heat transfer effectiveness of the three heat exchangers used in the experiments were calculated using the measurements data. The calculated effectiveness of the liquid-to-liquid plate heat exchanger on the desorber side was in the range of 65-72% and for the similar heat exchanger on the absorber side—in the range of 55-70%. For the air-to-air heat exchanger the calculated effectiveness was in the range of 75-78%.

Data Reduction and Analysis

The performance analysis of the DER absorption cycle performed by Hellmann and Grossman, [2,3] was intended to evaluate its potential quantitatively and obtain a fundamental understanding of its characteristic advantages and disadvantages. Due to the lack of literature data on heat and mass transfer coefficients, estimated values were employed. One of the main objectives of this experimental work has been to calculate realistic values of the heat and mass transfer coefficients from the measured data, to be used in the system simulation. As it turns out, the values used by Hellmann and Grossman, [2,3] were too optimistic.

The computer code ABSIM [4] developed specifically for simulation of absorption systems in flexible and modular form, was employed for the task of data reduction through simulation of the present system. Using the experimental data (measured and calculated as described in the previous section), the main units—dehumidifier (absorber) and regenerator (desorber)—were simulated. Both are described in the program by the modular component of air/solution contactor (the reader is referred to the original simulation by Hellmann and Grossman, [2,3] for details and terminology). In the regular mode of using the program, the user input contains data on unit characteristics—size, heat and mass transfer behavior and the like—and state point data at

Table 1 Typical sets of measured data (state point numbers are indicated with reference to Figure 2)

Experiment No.: Time:	Date: May 10, 2001						
	1 12:31	2 13:17	3 14:11	4 14:39	5 14:47	6 15:01	7 15:20
Solution Temperature at Regenerator outlet T_{Do} [°C] (1):	50.4	49.9	49.9	57.1	50.5	49.5	49.0
Solution Temperature at Regenerator inlet T_{Di} [°C] (2):	54.9	54.3	54.3	60.1	55.4	54.4	53.6
Heating water Temperature at heat exchanger inlet T_{hwi} [°C] (3):	63.8	63.1	62.9	65.2	65.2	64.0	62.8
Heating water Temperature at heat exchanger outlet T_{hwo} [°C] (4):	55.2	54.6	54.5	59.9	55.4	54.3	53.6
Relative Humidity of air to Regenerator at air H.X. inlet Φ_{XiDi} [%] (5):	55.0	47.6	52.0	— — —	54.0	54.0	54.1
Dry bulb Temp. of air to Regenerator at air H.X. inlet DBT $_{XiDi}$ [°C] (5):	22.7	23.3	23.3	24.4	23.0	23.1	23.2
Dry bulb Temp. of air to Regenerator at air H.X. outlet DBT $_{XoDi}$ [°C] (6):	22.4	24.0	23.4	— — —	22.8	22.8	22.9
Dry bulb Temp. of air at Regenerator outlet DBT $_{Do}$ [°C] (7):	41.9	41.7	41.6	38.8	41.5	41.6	41.2
Dry bulb Temp. of air from Regen. at air H.X. inlet DBT $_{XiDo}$ [°C] (8):	51.9	51.3	51.3	38.4	52.4	51.2	50.6
Dry bulb Temp. of air from Regen. at air H.X. outlet DBT $_{XoDo}$ [°C] (9):	49.4	49.1	48.9	28.2	49.2	48.9	48.4
Relative Humidity of air from Regen. at air H.X. outlet Φ_{XoDo} [%] (9):	28.4	28.6	28.4	27.4	28.8	28.6	28.4
K-flow Temperature reading at Regenerator outlet T_{KDo} [°C] (1c):	29.0	29.3	29.0	— — —	29.1	29.2	28.9
K-flow Flow reading at Regenerator outlet F_{KDo} [kg/sec] (1c):	73.0	71.0	73.0	— — —	95.0	85.0	85.0
K-flow Density reading at Regenerator outlet D_{KDo} [gr/cm ³] (1c):	48.5	48.1	48.0	52.5	49.2	47.9	47.2
Heating water flow rate F_{hw} [kg/sec] (3):	0.05	0.05	0.05	0.05	0.05	0.05	0.05
Pressure difference (ambient-static) at Regen. outlet ΔP_{Dsa} [inch WG] (7):	1.264	1.259	1.257	1.225	1.233	1.236	1.238
Pressure difference (total-static) at Regen. outlet ΔP_{Dst} [inch WG] (7):	0.22	0.22	0.22	0.22	0.22	0.22	0.22
Solution Temperature at Dehumidifier outlet T_{Ao} [°C] (10):	0.78	0.78	0.78	— — —	0.76	0.76	0.76
Solution Temperature at Dehumidifier inlet T_{Ai} [°C] (11):	0.185	0.185	0.185	— — —	0.19	0.19	0.19
Cooling water Temperature at Heat Exchanger inlet T_{cwi} [°C] (12):	29.7	29.5	30.7	29.4	30.4	30.1	30.3
Cooling water Temperature at Heat Exchanger outlet T_{cwo} [°C] (13):	26.9	26.9	28.4	27.4	28.2	28.0	28.1
Relative Humidity of air at Dehumidifier inlet Φ_{Ai} [%] (14):	22.4	22.4	22.5	22.6	22.4	22.4	22.4
Dry bulb Temperature of air at Dehumidifier inlet DBT $_{Ai}$ [°C] (14):	26.2	26.3	28.3	27.1	28.0	27.9	28.0
Dry bulb Temperature of air at Dehumidifier outlet DBT $_{Ao}$ [°C] (15):	56.7	53.3	50.0	— — —	55.0	55.0	58.0
Relative Humidity of air at Dehumidifier Fan outlet Φ_{AFo} [%] (16):	23.7	24.2	24.6	27.2	24.3	24.3	24.3
Dry bulb Temperature of air at Dehumidifier Fan outlet DBT $_{AFo}$ [°C] (16):	23.5	23.2	24.0	— — —	23.1	24.0	23.3
K-flow Temperature reading at Dehumidifier outlet T_{KAo} [°C] (11c):	27.6	27.5	28.8	28.4	28.8	28.5	28.5
K-flow Flow reading at Dehumidifier outlet F_{KAo} [kg/sec] (11c):	18.3	19.0	19.4	— — —	23.0	26.6	26.8
K-flow Density reading at Dehumidifier outlet D_{KAo} [gr/cm ³] (11c):	29.1	29.1	30.3	23.5	30.2	30.0	29.9
Cooling water flow rate F_{cw} [kg/sec] (12):	28.9	28.8	30.0	— — —	29.9	29.6	29.6
Press. diff. (ambient-static) at Dehumidifier outlet ΔP_{Assa} [inch WG] (15):	26.3	26.2	27.5	26.5	27.3	27.2	27.3
Press. diff. (total-static) at Dehumidifier outlet ΔP_{Ast} [inch WG] (15):	0.05	0.05	0.05	0.05	0.05	0.05	0.05
Ambient pressure P_{amb} [kPa]:	1.248	1.246	1.246	1.226	1.225	1.224	1.225
	0.333	0.317	0.167	0.167	0.167	0.183	0.183
	1.98	1.98	1.98	— — —	2.00	2.00	2.00
	0.64	0.64	0.64	— — —	0.63	0.63	0.63
	97.7	97.7	97.7	97.7	97.7	97.7	97.7

(*) Double rows of temperature data indicate temperature reading by two instruments simultaneously.

inlets—specified temperatures, flowrates etc. The results of the calculations yield the parameters at the output state points. In the present case, input and output state points are given from measurements and the units characteristics, heat and mass transfer coefficients, are unknown and have to be calculated. Note that the data sets collected represent an array of typical and specific working conditions where quasi-steady state was reached after a long period of operation. With the way the experimental system was designed, without controls at this stage, it was not possible to maintain certain parameters constant and vary others systematically. The approach was hence to reduce the collected data sets to calculate heat and mass transfer coefficients, then investigate systematic behavior through simulation using these coefficients.

Data sets similar to the one of Table 1 were generated in multiple series of experiments. As evident, the data sets contain the key parameters characterizing the performance of the system—some measured, other calculated—as described in the previous section. Using the measured data with ABSIM for estimating the mass transfer coefficients has yielded mass transfer rates of 0.9 kg/s (interface-air) and 0.3 kg/s (interface-solution). The effective surface area of the packing material was 44.5 m² (the volume of the packing material used in the experiments was 0.156 m³ and the effective surface area of the packing is 285 m² per m³). This data yields mass transfer coefficients of $\sigma=0.02$ kg/m²s (interface-air) and $\beta=0.0067$ kg/m²s (interface-solution). Note

that the heat transfer coefficient on the air side is determined by the assumption that the Lewis number equals unity, and on the solution side by the assumption of negligible temperature difference between interface and bulk (these assumptions are explained in detail in the articles by Hellmann and Grossman [2,3]). The early simulations of the dehumidifier and the regenerator in the DER cycle described in these articles used mass transfer coefficients of $\sigma=0.76 \text{ kg/m}^2\text{s}$ (interface-air) and $\beta=0.21 \text{ kg/m}^2\text{s}$ (interface-solution); the results of the present experiments show these values to have been unrealistically high. Kessling [7] found in his experiments effective mass transfer values between $0.007 \text{ kg/m}^2\text{s}$ and $0.02 \text{ kg/m}^2\text{s}$.

The transfer coefficients measured in this work were subsequently employed in a separate study [9] to simulate the performance of the open absorption system under varying operating conditions. Effects of inlet air humidity, temperature and flowrate, solar-heated water temperature and desiccant flowrate on performance were studied. As expected, the humidity of ambient air is found to have a dominant influence on performance.

Conclusion

The experimental system constructed and operated in this study has yielded important information about practical design aspects of the key components—dehumidifier and regenerator—as well as quantitative data about their performance. The operation of the system was studied under varying operating conditions. Measurements have provided much-needed realistic data about heat and mass transfer coefficients. In comparing these results with the assumed values employed in the earlier DER simulation of Hellmann and Grossman [2,3] it is clear that the assumed values were

too optimistic. A simulation of the open absorption system has now been conducted using the measured values, to determine the effect of operating parameters on performance.

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