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Experimental Testing of a Prototype AirGreen Liquid-Desiccant Air-Conditioner

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Final Report

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Executive Summary

A prototype liquid-desiccant air-conditioner manufactured by AirGreen Corporation was installed at Queen's University's *Thermal Systems Test Facility* located in Kingston, ON and was tested during September 2014. The goals of the tests were to evaluate the thermal and functional performance of the prototype AirGreen system. For testing, the unit was installed outdoors and was subject to variable atmospheric conditions during operation. Thermal input to the system was provided by a hot-water circulation-loop heated to a nominal set-point temperature of 135ºF (57.2ºC) by a natural-gas fired boiler. Monitored results for eight test days are analyzed and summarized in this report. Average performance data is reported over quasi-steady-state test periods and included: latent and sensible cooling rates, thermal and electrical energy consumption and functional performance. The system was installed under the supervision of AirGreen personnel and Queen's staff. It was operated by Queen's personnel during the tests and was observed to function as intended, producing 1300 CFM of conditioned air at an average total cooling rate of 2.4 to 5.6 tons (8.4 to 19.7 kW). Since testing was carried out towards the end of the summer, the total air-conditioning load (i.e., dehumidification and sensible cooling load) was moderate and the AirGreen unit was occasionally operated at part-load (i.e., below its rated capacity). Total air-conditioning rate was also observed to depend on the temperature of cooling water supplied to unit, and the (outdoor) air temperature and humidity, with values ranging from 2.4 to 3.4 tons of latent cooling (i.e., dehumidification) and -0.3 to 2.5 tons of sensible cooling (with the negative value representing an increase in the temperature of the conditioned air). The test results indicated that, as the temperature of the cooling water (used for heat rejection) increased, latent cooling capacity (i.e., dehumidification capacity) remained fairly constant, but sensible cooling capacity (i.e., indicated by the temperature reduction in the process-air stream) decreased, consistent with expectations. Due to facility constraints, maintaining cooling water temperature during long test periods was difficult, however, performance metrics evaluated during test periods with stable cooling water inlet temperatures indicated: a thermal COP of 0.58; an electrical COP of 4.7 (including power consumption for the unit's pumps and fans); and, a cooling water effectiveness of 0.78 (i.e., total air-conditioning rate/heat rejection rate). The unit's airflow to total cooling rate ratio (i.e., airflow/air-conditioning rate) ranged from 230 to 540 CFM/ton. In summary, the experimental results clearly indicate that the prototype unit was able to reliably condition an outdoor airstream while using only low temperature heat and a small amount of electricity as energy inputs.

Introduction

A prototype liquid-desiccant air-conditioner was installed and instrumented at the test site in Kingston, Ontario for testing during September, 2014. This device was designed to operate as a dedicated outdoor air system (DOAS) with the principle goal of dehumidifying a supply air stream. The system used a liquid desiccant solution of lithium bromide and a small amount of lithium chloride with water to dehumidify the air, taking advantage of the hygroscopic nature of the salts.

Objectives of the Tests

The primary objectives of the test program were to evaluate the thermal and functional performance of the prototype AirGreen system. Specifically, tests were undertaken to:

1. verify the unit's ability to reduce relative humidity below 40%;
2. verify the regeneration the desiccant solution with hot water at temperatures as low as 135ºF;
3. determine the cooling capacity of the system as a function of the airflow rate through the unit;
4. monitor the system for ease of operation and robustness; and,
5. record the thermal and electrical COP values obtained during the system's operation.

Description of Unit Tested

The AirGreen unit is a thermally driven air-conditioning unit that utilizes a concentrated liquid desiccant solution to dehumidify and cool a process air-stream in a proprietary conditioner section. During operation, the liquid desiccant absorbs moisture from the process-air (humidity), diluting the desiccant solution. To facilitate this process, cooling water is supplied to the unit's conditioner section from an available source. After absorbing moisture in the conditioner, the resulting dilute, low-concentration desiccant solution is pumped to a proprietary regenerator where heat is added to drive off the absorbed moisture. This re-concentrates the desiccant solution before being returned to the conditioner section in a continuous process. Heat delivered to the regenerator can typically be supplied from relatively low temperature sources (e.g., 135ºF or 57.2ºC) including: absorption and vapor-compression refrigeration and heat pumps, waste heat recovery units; or renewable energy systems. Moisture rejected in the regenerator section is transferred to a separate "scavenging" air-stream. A typical arrangement would be to use building exhaust-air as the source of the scavenging air-stream. The unit tested contained fans to move both the process and scavenging air-streams, as well as, internal pumps to transport the desiccant solution between the conditioner and regenerator sections of the unit. Both the regenerator and conditioner were "direct contact" type mass and energy transfer devices arranged in a proprietary configuration as described in a provisional US patent (Application #61/979882).

Experimental Description

The AirGreen unit was installed and operated during September 2014. Heating water was supplied via a two-stage natural gas boiler while the cooling water was supplied from a 1600 gallon (US) storage tank. An evaporative cooling tower was run overnight to cool the water for use during testing the following day. Depending on flow rate, thermal stratification in this tank allowed for one to two hours of operation at relatively constant cooling-water temperatures. During the tests the cooling tower was bypassed and water was returned directly into the top of the tank. The system schematic is shown in Fig. 1.

The desiccant solution was stored in a 25 gallon (US) drum. The conditioner and regenerator each had their own additional 12 gallon (US) sumps. To promote stratification in the desiccant drum, weak solution from the conditioner sump was delivered to the top of the drum at a constant flow rate, and strong solution from the regenerator was pumped to the bottom of the drum at a constant flow rate. A floating barrier provided physical separation of the two layers in the storage drum. Two submersible pumps in the drum circulated strong desiccant from the bottom of the drum to the conditioner sump and weak desiccant from the top of the drum to the regenerator.

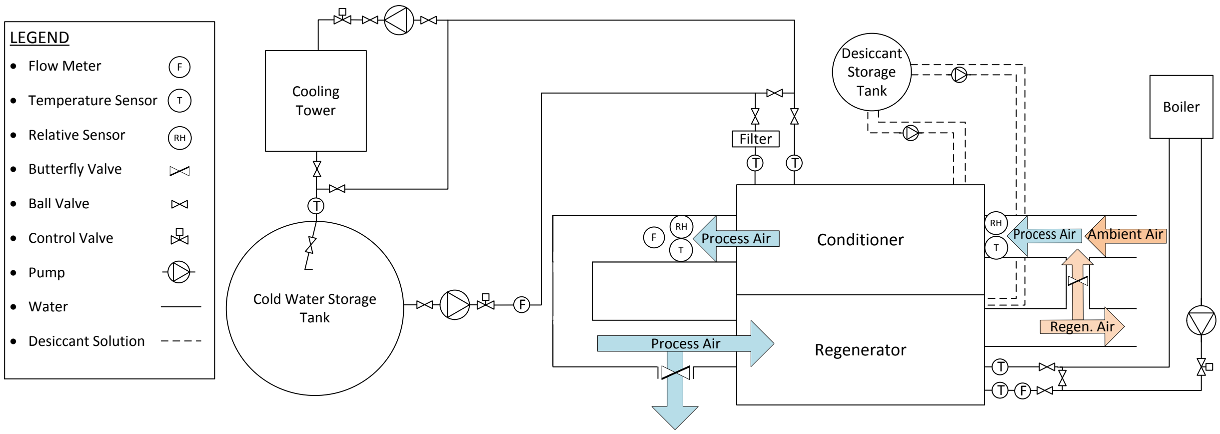

Fig. 1. System layout showing location of sensors and airflow

The system was instrumented to obtain the total cooling capacity[1] of the system, the cooling water effectiveness (ε_{CW}), and to determine the thermal and electrical coefficients of performance (COP_T and $COPe$ respectively). These performance metrics are defined in Eq. 1, 2 and 3.

$$\varepsilon_{CW} = \frac{\text{Total cooling rate}}{\text{Heat rejection rate to cooling water}} \quad \textbf{Eq. 1}$$

$$COP_T = \frac{\text{Total cooling rate}}{\text{Heat input rate to regenerator}} \quad \textbf{Eq. 2}$$

$$COP_e = \frac{\text{Total cooling rate}}{\text{Rate of electrical power consumption}} \quad \textbf{Eq. 3}$$

The total cooling rate was determined by comparing the temperature and relative humidity of the process air-stream at the conditioner's inlet and outlet. These values were determined by a pair of identical temperature and humidity sensors (see Appendix B for information on sensors). A pressure transducer and an accompanying pitot tube array was used to determine the air flow rate.

The rate that heat was rejected to the cooling water was found by recording the water temperature at the inlet and outlet of the unit, as determined by thermistors, and the flow rate of cooling water, as determined by a magnetic flow meter. The heat input rate to the regenerator was calculated in the same fashion, using thermistors at the heating water inlet and outlet of the regenerator and an impeller flow meter.

The electrical power consumption was monitored by a power transducer. This included all electricity consumed by the desiccant and water pumps, as well as, the regenerator and conditioner fans. The PLC system controlling

[1] In this report "total cooling capacity or rate" is used to represent total air-conditioning capacity or rate and includes both latent cooling (i.e., dehumidification) and sensible cooling rates consistent with ANSI/ASHRAE Standard 174-2009 [1].

the boiler and the data logger were also included in the power draw total but these only consumed approximately 50 W of power. The cooling tower pump and fan were only run at night to cool the water in the storage tank and did not contribute to the reported power consumption. The cooling water pump was a high-efficiency variable-speed pump, but the heating water pump was a single speed unit rated at two horsepower. This was considered to be oversized for this particular application. Photos of the system as installed are shown in Fig. 2, Fig. 3, and **Fig. 4**.

Fig. 2. Side view of the AirGreen unit and testing apparatus

The system was operated under variety of atmospheric conditions due to the variable weather conditions experienced during the test periods. The temperature of the cooling water was dependent on the temperature distribution in the supply water storage and was influenced by the previous night's ambient air conditions. Due to the stratification of the tank, it was possible to keep the cooling water temperature effectively constant for periods of one to two hours (depending on the flow rate of cooling water) and allowed "quasi-steady-state" performance measurements to be taken. After these periods, however, the cooling water temperature increased over the course of the day which primarily impacted the sensible cooling capacity of the machine.

The conditioner inlet-air conditions were also dependent on the ambient-air conditions. For testing, the air exiting the conditioner section of the unit was directed into the regenerator to represent the conditioned air exiting a building. Due to the time of year, both ambient-air and humidity levels were moderate and did not represent full-load conditions for the unit. In an attempt to increase the effective air-conditioning load, a fraction of the exit-air from the regenerator was recirculated into the inlet of the conditioner. A butterfly valve was used to modulate the amount of recirculation air. This arrangement increased both the inlet-air temperature and the absolute humidity of the process air-stream significantly but tended to reduce the inlet relative humidity.

During operation, the hot water temperature and flow rate were manually set to maintain the concentration of the desiccant solution at the regenerator outlet at constant levels. The specific gravity of the solution was monitored at thirty minute intervals throughout the test using a hydrometer. This hydrometer was checked against an Anton Paar DMA 4500 density meter (appendix B) to ensure its accuracy. The solution mass fraction at the outlet of the regenerator was typically in the range of 0.50 to 0.54 (calculated from a specific gravity ranging from 1.53 to 1.60 using correlations from literature [2]. To further increase the load on the machine, a variable-speed controller was used to reduce the regenerator fan speed; increasing the humidity level of regenerator exhaust-air supplied to the conditioner. The flow rate of the desiccant solution was set between one to two liters per minute as determined by quantitative methods (i.e., graduated cylinder and stopwatch).

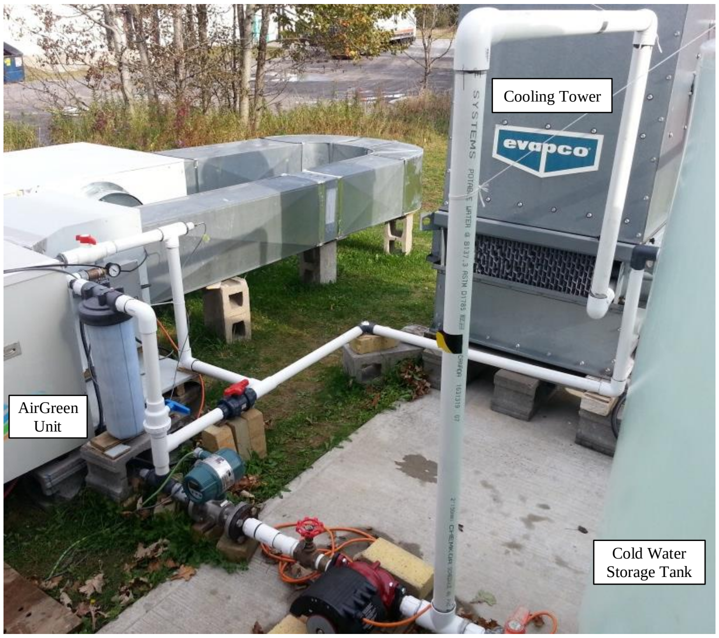

Fig. 3. View of the cooling water system with edge of tank on far right side

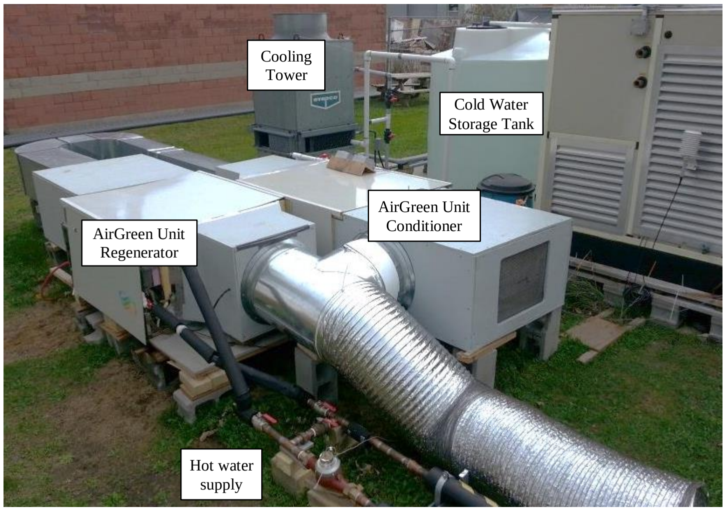

Fig. 4. View of the heating water supply and the AirGreen unit

Test Results

During the eight test days, data obtained during quasi-steady-state periods[2] indicated total cooling rates rate of between 2.4 to 5.4 tons depending on test conditions. Values of 2.4 to 3.4 tons of latent cooling and -0.3 to 2.5 tons of sensible cooling were measured at an air flow rate of approximately 1300 CFM. Fig. **5** shows how the cooling rate values varied with changing cooling water temperature. To determine the performance metrics (i.e., Eq.'s 1 to 3), instantaneous data recorded over these quasi-steady-state test periods was averaged for each day, see Appendix A. To arrive at an overall estimate of the unit's performance, the average of the daily values was calculated and indicated an *overall average* COP_T of 0.58, a COP_e of 4.7, and a cooling water effectiveness of 0.78. The system delivered an *overall average* "airflow to total air-conditioning rate ratio" of approximately 350 CFM/ton, with values ranging between 230 and 540 CFM/ton.

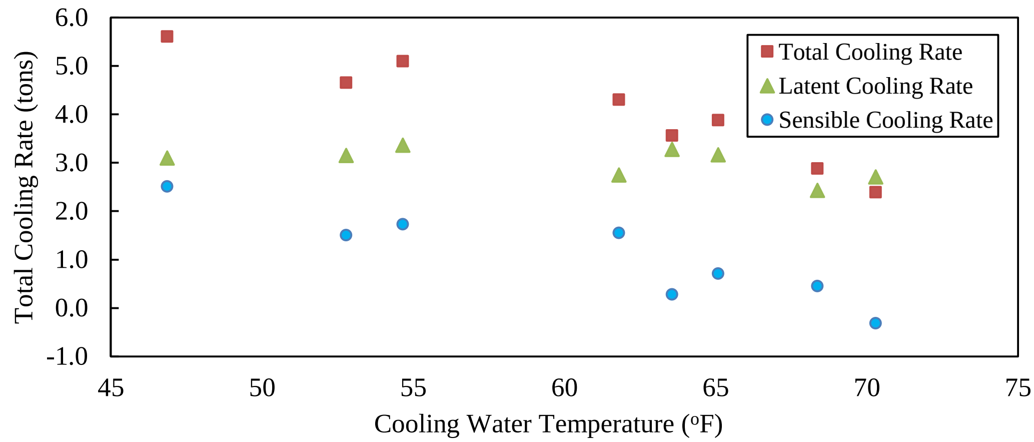

Fig. 5. Average total, latent and sensible cooling rate at different cooling water inlet temperatures

Each data point in Fig. 5 represents an average of the data recorded over each quasi-steady-state test period (typically of 30 to 90 minutes duration) when the cooling water temperature was coldest. Each data point used approximately the same cooling water flow rate of 10 GPM (US). The inlet-air conditions, however, often varied over the course of test period and from day to day (e.g., from 80°F to 92°F and relative humidity 36% to 70%) as listed in Appendix A. The variation in humidity caused some data points have lower than expected total or latent cooling rates. For example, during the day with 55°F cooling water, the total cooling rate was higher than the day with cooling water at 53°F. The average inlet-air conditions for the 53°F day were lower in both temperature and relative humidity, leading to reduced cooling rates. A plot of a typical air-conditioning process as recorded on September 16th is displayed on a psychrometric chart in Appendix C.

Fig. **5** shows that lowering the cooling water temperature increased the total cooling rate. This increase was primarily caused by the increase in sensible cooling. The cooler water temperature did not show a large impact on the latent cooling rate. This is due to the reduced impact of temperature on the water vapor partial pressure of the desiccant at lower temperatures as indicated in Appendix D for $LiBr$-H_20 solutions [2].

To illustrate the effect of process inlet-air conditions, recorded data was divided into three different 10 minute test periods based on inlet-air conditions. Fig. **6** shows results of inlet-air temperature on cooling rate for a 55°F cooling water at 10 GPM (US) flow rate.

[2] For the purpose of this report quasi-steady-state periods are considered to be when the cooling water delivery temperature was within 2°F of its initial temperature at the start of the test period.

These results showed that increasing the temperature of the inlet-air increased sensible cooling rates, while increasing the humidity level increased the latent cooling rates. Appendix A contains further data on performance during these conditions.

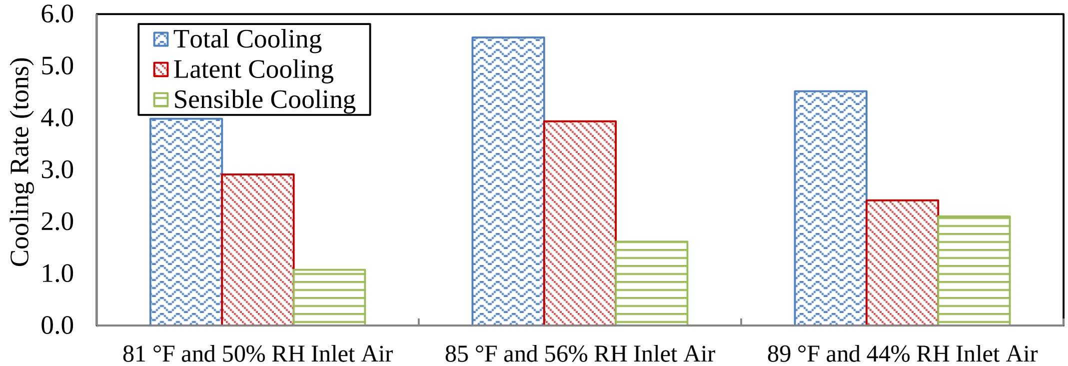

Fig. 6. Effect of inlet air conditions for cooling water at10 GPM (US) and 55ºF

The results of a full test run are displayed in Fig. 7, **Fig. 8**, and Fig. 9 to show the performance of the unit on September 10th including a typical test period (used to calculate average performance) in which the cooling water was effectively constant (i.e., approximately 12:30 to 14:00), as well as, the values recorded as the cooling water temperature increased over the balance of the day.

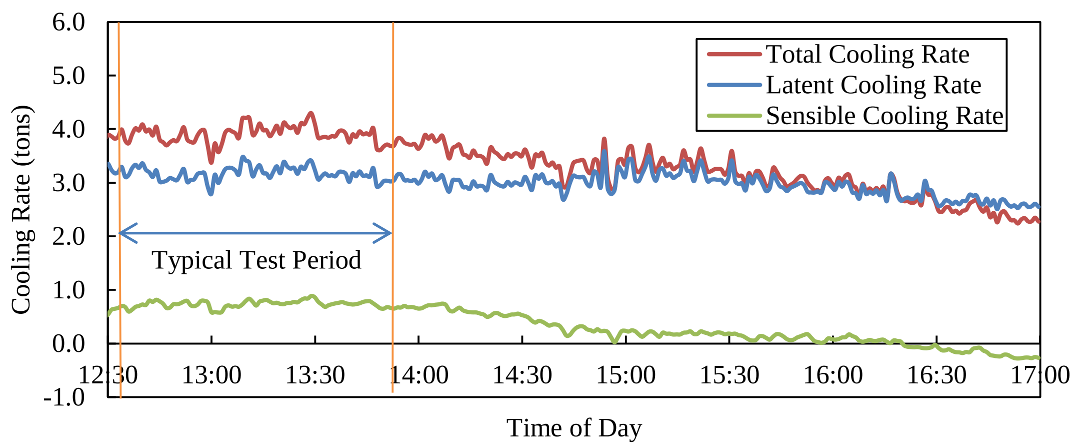

Fig. 7. System air-conditioning rates on September 10

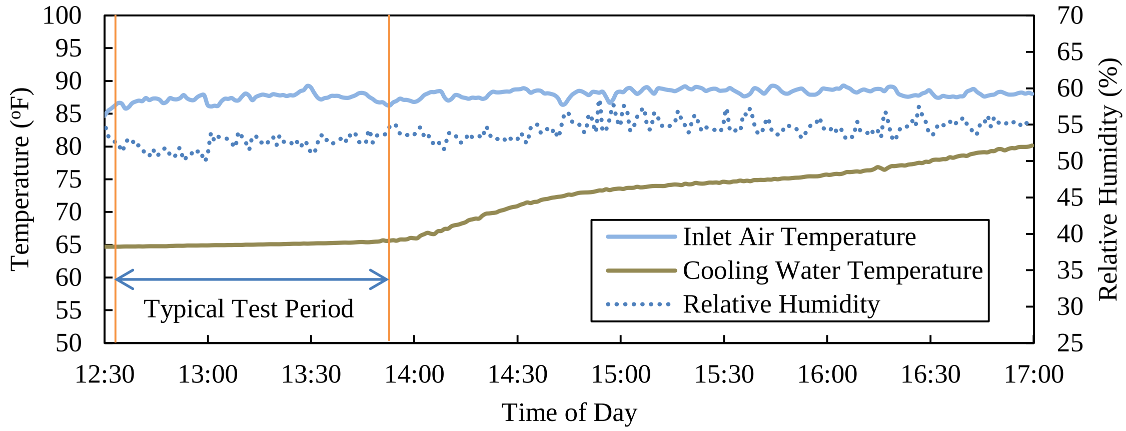

Fig. 8. Inlet air conditions and cooling water temperature on September 10

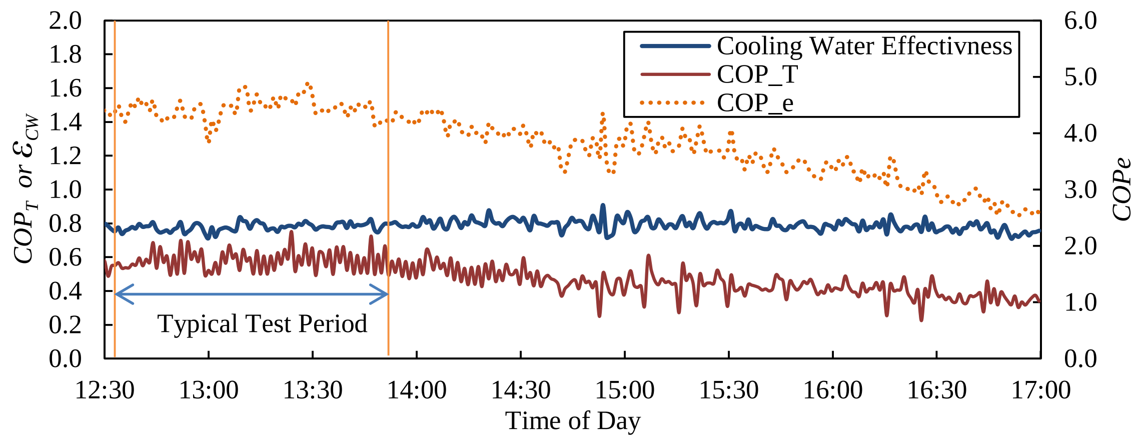

Fig. 9. Performance of the unit on September 10

Fig. **9** shows that the cooling water effectiveness remained relatively constant, even while the cooling water temperature was increasing over the course of the test period. The COP_T also remained relatively constant, with a slight decreasing trend as the cooling water temperature increased. The rapid fluctuations in COP_T were due to the hot water boiler cycling between off, low-fire, and high-fire modes as it attempted to maintain the set point temperature. From the plot it is evident that COP_e depends on total cooling rate for the configuration tested as the electrical consumption of the unit remained constant at roughly 3 kW during the experiment, while the total air cooling rate varied. It is probably, that with modulating fans and pumps, this effect can be minimized.

Other operational factors may have also influenced the results. For example, the desiccant concentration affects the latent cooling rate (e.g., a more concentrated solution will absorb more moisture). The flow rates and temperatures of the heating water and the temperature, humidity and flow rate of the air in the regenerator affect this concentration, however, it was not in the scope of this study to identify optimal operational configurations. Rather, the goal focus of this study was to focus on the "Objectives of the Study" as described above.

Discussion of Results

From an analysis of the experimental data it was possible to evaluate the prototype unit in relation to the objectives of this brief study. In particular it was observed that:

i. during all test sequences the unit was able to significantly reduce the humidity level in the process air stream, delivering conditioned air at low relative humidity levels (Appendix A);
ii. the unit achieved good performance when driven at relatively low water temperatures, (i.e., 135ºF);
iii. a maximum total cooling rate of 5.6 tons was recorded at an airflow rate of approximately 1300 CFM, achieving an airflow to total cooling rate ratio of 230 CFM/ton;
iv. the installation, instrumentation and commissioning of the unit was relatively simple and the unit was operated reliably for 8 days.

It was also noted that the system was capable of sensibly cooling air even during periods of high latent loads which can be difficult to achieve in desiccant systems. Periods of negative sensible cooling were also recorded but these occurred only when the cooling water temperature increased during the latter part of the test sequences.

In general, the unit was easily set up and operated reliably. A few minor start-up issues were quickly rectified during the test period, however, these were primarily due to the fact that the test unit was a prototype.

With regard to study objective 5: analysis of the results indicated an *overall average* COP_T of 0.58, a COP_e of 4.7, and a cooling water effectiveness of 0.78. These values are considered to be relatively high when compared to other desiccant systems on the market operating from a comparable low temperature source. It should be noted as well, that the COP_e was negatively impacted by the use of an oversized, single-speed hot water pump for the tests. This oversized pump was a part of the test apparatus and a higher efficiency smaller pump would have further increased the COP_e.

AirGreen personnel, have indicated that there is further potential for improvement to the system by refining the specifications of some of the unit's other subcomponents, e.g., heat exchangers. Also, as previously noted, the system was frequently operated below its full capacity due to the fact that tests were conducted outdoors, late in the summer season. Increasing the load on the machine should further improve the thermal and electrical COP's of the system.

Conclusion

A prototype liquid desiccant air-conditioner was provided by AirGreen Inc. and was installed, instrumented and run outdoors under a simulated load during the fall season of 2014. The unit was designed for use as a "dedicated outdoor air system", intended to condition make-up air supplied to a building while rejecting moisture to a building's exhaust air-stream. Test results confirmed that unit was able to condition air at an average rate of 2.4 to 5.4 tons, with latent cooling rates represented 2.4 to 3.4 tons and the sensible cooling rates ranging from -0.3 to 2.5 tons. Averaged performance metrics indicated a COP_T of 0.58, COP_e of 4.7, and a cooling water effectiveness of 0.78. Finally, although further testing under extreme humidity and temperature conditions would be of value, the thermal, and the functional performance, of the prototype AirGreen liquid desiccant air-conditioner was demonstrated.

References

[1] ANSI/ASHRAE Standard 174-2009. Method of Test for Rating Desiccant-Based Dehumidification Equipment. American Society for Heating, Refrigerating and air Conditioning Engineers, Inc., 1791 Tullie Circle NE, Atlanta, GA 30329, (www.ashrae.org)

[2] Patek, J., and Klomfar, J., "A computationally effective formulation of the thermodynamic properties of LiBr-$H_2$0 solutions from 271 to 500 K over full composition range", International Journal of Refrigeration, Vol. 29, pp. 566-578, 2006.

Appendix A – Data tables

Table A1. Inlet and outlet conditions for system at different cooling water temperatures

	Power Draw	T Air In	T Air Out	RH Air In	RH Air Out	Air Flow	Cooling Water Flow Rate	Hot Water Flow Rate	T Cooling Water In	T Cooling Water Out	T Hot Water In	T Hot Water Out	Humidity Ratio in	Humidity Ratio Out
	kW	ºF	ºF	%	%	CFM	GPM	GPM	ºF	ºF	ºF	ºF	kg $H_2$0/kg dry air	kg $H_2$0/kg dry air
Sept. 04	3.05	82.7	80.9	59.0	34.3	1307	14.7	20.2	63.6	72.4	126.9	120.4	0.0141	0.0076
Sept. 05	2.89	81.1	84.4	70.3	42.4	1295	11.1	18.2	70.3	78.4	135.4	129.7	0.0160	0.0107
Sept. 09	3.16	90.7	78.0	41.0	35.3	1345	11.0	16.3	61.8	73.6	131.9	119.9	0.0124	0.0071
Sept. 10	3.10	87.1	81.9	52.7	37.5	1394	10.6	14.8	65.1	76.3	130.8	119.7	0.0145	0.0086
Sept. 11	3.13	84.6	81.3	53.8	39.9	1348	11.0	15.3	68.4	76.9	131.6	119.7	0.0138	0.0091
Sept. 15	3.01	82.5	70.3	48.8	35.3	1337	9.7	13.2	52.8	65.8	129.7	113.4	0.0115	0.0055
Sept. 16	3.01	85.7	72.3	52.0	44.5	1391	10.1	13.2	54.7	67.7	134.3	118.8	0.0136	0.0074
Sep. 17	3.06	90.5	69.9	36.7	34.5	1344	10.5	16.8	46.9	63.1	136.9	123.1	0.0112	0.0053

Table A2. System performance at different cooling water temperatures

	Moisture Absorption Rate	Total Cooling	Latent Cooling	Sensible Cooling	ε_{CW}	COP_T	COP_e
	lb/hr	tons	tons	tons	~	~	~
Sept. 04	36.6	3.56	3.28	0.28	0.67	0.68	4.20
Sept. 05	30.2	2.39	2.71	-0.32	0.64	0.68	2.91
Sept. 09	30.7	4.31	2.75	1.55	0.78	0.56	4.81
Sept. 10	35.4	3.88	3.17	0.71	0.78	0.57	4.41
Sept. 11	27.1	2.90	2.4	0.5	0.73	0.39	3.24
Sept. 15	35.2	4.66	3.15	1.51	0.90	0.53	5.43
Sept. 16	37.6	5.10	3.37	1.73	0.97	0.61	5.99
Sep. 17	34.6	5.61	3.10	2.51	0.79	0.61	6.46

Table A3. Inlet and outlet conditions for system at different inlet air conditions

	Power Draw	T Air In	T Air Out	RH Air In	RH Air Out	Air Flow	Cooling Water Flow Rate	Hot Water Flow Rate	T Cooling Water In	T Cooling Water Out	T Hot Water In	T Hot Water Out	Humidity Ratio in	Humidity Ratio Out
	kW	ºF	ºF	%	%	CFM	GPM	GPM	ºF	ºF	ºF	ºF	kg $H_2$0/kg dry air	kg $H_2$0/kg dry air
81F 50%RH	3.00	80.6	72.3	49.8	33.9	1370	9.5	12.9	54.3	66.1	133.8	120.5	0.0110	0.0056
85F 56%RH	3.01	85.1	72.5	56.3	42.5	1371	10.1	12.7	54.5	68.2	134.5	118.3	0.0145	0.0072
89F 44%RH	3.03	88.7	71.8	44.3	50.2	1373	10.7	14.0	54.9	67.1	134.2	118.7	0.0128	0.0083

Table A4. System performance at different inlet air conditions

	Moisture Absorption Rate	Total Cooling	Latent Cooling	Sensible Cooling	ε_{CW}	COP_T	COP_e
	lb/hr	tons	tons	tons	~	~	~
81F 50%RH	32.4	3.98	2.91	1.07	0.86	0.56	4.66
85F 56%RH	43.9	5.55	3.93	1.62	0.96	0.65	6.48
89F 44%RH	26.8	4.51	2.41	2.10	0.81	0.51	5.24

Appendix B – Error analysis and Instrument Specifications

Table B1. Manufacturer Reported Sensor Accuracy

Location	Sensor Type	Manufacturer Reported Sensor Accuracy
Power Draw	Continental Control Systems. WattNode Pulse WNB-3D-240-P	0.5 % of reading
T Air In	Vaisala. HMT 333 Series Humidity and Temperature Transmitters.	±0.2ºC
T Air Out	Vaisala. HMT 333 Series Humidity and Temperature Transmitters.	±0.2ºC
RH Air In	Vaisala. HMT 333 Series Humidity and Temperature Transmitters.	±1 % RH
RH Air Out	Vaisala. HMT 333 Series Humidity and Temperature Transmitters.	±1 % RH
Air Flow	Ruskins. AMS 810 Pressure Transducer	3 % of reading
Cooling Water Flow Rate	Yokogawa. AXF Magnetic Flowmeter and AXFA14G/C Magnetic Flowmeter Remote Converter	±0.5 LPM
Hot Water Flow Rate	Elster. M190 Multi-Jet Impeller Hot Water Flowmeter	±5 LPM
T Cooling Water In	10 k ohm Thermistor (4159-1/8-6-25-TH44036-FEP)	±0.2ºC
T Cooling Water Out	10 k ohm Thermistor (4159-1/8-6-25-TH44036-FEP)	±0.2ºC
T Hot Water In	10 k ohm Thermistor (4159-1/8-6-25-TH44036-FEP)	±0.2ºC
T Hot Water Out	10 k ohm Thermistor (4159-1/8-6-25-TH44036-FEP)	±0.2ºC

The experimental measurement errors of the measurement system were calculated using the root mean squared (RMS) Method. It is assumed that all errors are random and independent for this analysis. Samples of error analysis on the results from Fig. 5 are given in Table B2. The data-logger polled the sensors every 5 seconds, and the results were averaged and saved every minute. Large percent errors are seen for some data on sensible cooling. This is due to the near zero value of sensible cooling during these experiments.

Table B2. Average error over reported test times

	Moisture Absorption Rate	Moisture Absorption Rate Error	Total Cooling	Total Cooling Error	Latent Cooling	Latent Cooling Error	Sensible Cooling	Sensible Cooling Error	ε_{CW}	ε_{CW} Error	COP_T	COP_T Error	COP_e	COP_e Error
	lb/hr	%	tons	%	tons	%	tons	%	~	%	~	%	~	%
Sept. 04	36.65	3.5	3.56	2.1	3.28	3.5	0.28	61.7	0.67	2.8	0.68	7.1	4.20	2.1
Sept. 05	30.23	3.8	2.39	3.0	2.71	3.8	-0.32	-41.3	0.64	3.7	0.68	8.6	2.91	3.0
Sept. 09	30.74	3.9	4.31	2.0	2.75	3.9	1.55	11.8	0.78	2.7	0.56	8.7	4.81	2.1
Sept. 10	35.40	3.7	3.88	2.1	3.17	3.7	0.71	20.4	0.78	2.8	0.57	9.4	4.41	2.2
Sept. 11	27.13	4.0	2.9	2.6	2.4	4.0	0.5	100.3	0.73	3.3	0.39	9.2	3.24	2.6
Sept. 15	35.21	3.4	4.66	1.7	3.15	3.4	1.51	9.0	0.90	2.5	0.53	10.2	5.43	1.8
Sept. 16	37.62	3.5	5.10	1.7	3.37	3.5	1.73	8.7	0.97	2.4	0.61	10.3	5.99	1.7
Sep. 17	34.61	3.6	5.61	2.1	3.10	3.5	2.51	5.8	0.79	2.3	0.61	8.2	6.46	1.7

Appendix C – Sample of select data on psychrometric chart

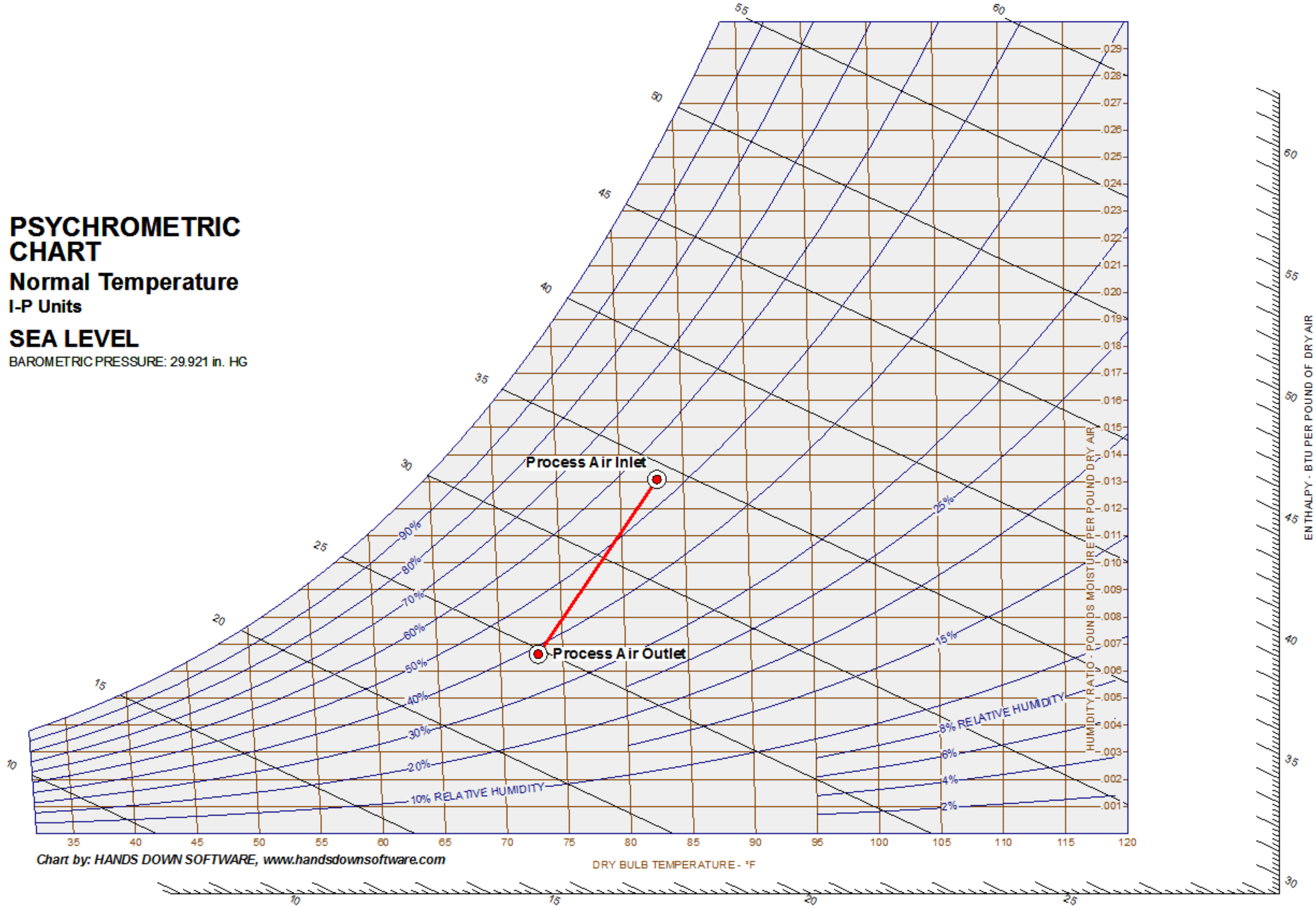

Fig. C1. Psychrometric chart showing average experimental data for the process air inlet and outlet conditions on September 16, from 11:20 to 11:29, for a cooling water temperature of 54.5ºF

Appendix D – Water vapor partial pressure of $LiBr\text{-}H_20$ solutions

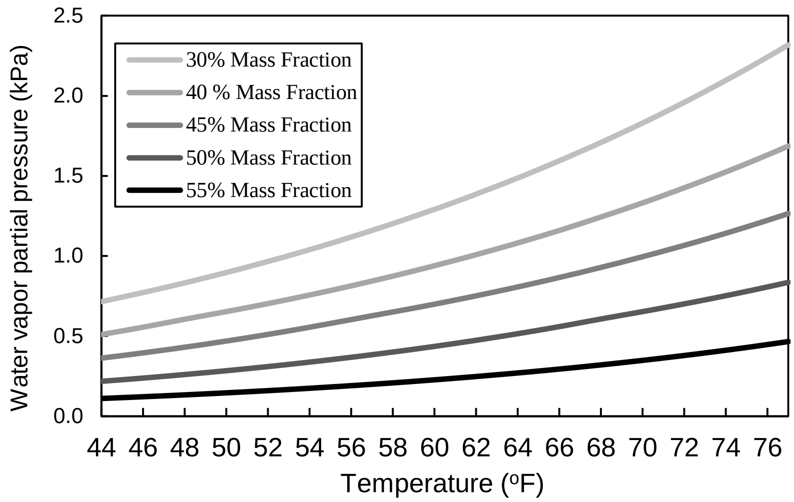

Fig. D10. Water vapor partial pressure of lithium bromide at different mass fractions and temperatures, (calculated according to [2])