Achieving 50% Energy Reduction with Liquid Desiccant DOAS

Philip C. Farese, Ph.D. **Rachel Ellman, Ph.D.** **Aaron Meles** **Kris Toivola**

ABSTRACT

Liquid desiccants have long promised energy reductions for applications requiring delivery of air in the dewpoint range of 35-55ºF (2–13ºC), but innovators have struggled to develop a packaged product capable of realizing these promises and operating reliably in the field. We report on a prolonged field campaign comprising over 30,000 operating hours across three climates and five sites. Sites were selected in Florida, Texas, and Michigan, allowing operation across a wide range of ambient/inlet conditions. A performance model was developed in Engineering Equation Solver (EES) to design and predict the performance of these Liquid Desiccant Dedicated Outdoor Air System (LD-DOAS) units and compared to conventional direct-expansion-only DOAS (DX-DOAS) units. The units were built and tested in the laboratory environment demonstrating an ISMRE2 (Integrated Seasonal Moisture Removal Efficiency) of 9.5 lbs./kWh (4.3 kg/kWh). The units were installed and started up between April, 2023 and March, 2024. To date 31,673 hours of data have been collected; operations continue. The units have demonstrated 97% system uptime with uptime of 99% for the liquid desiccant subsystem. Unit efficiency has met design with the moisture removal efficiency measured at 101% +/- 4% of the modeled efficiency. Accordingly, field operations verify the modeled ISMRE2 of 9.5 lbs./kWh (4.3 kg/kWh) and demonstrate an energy savings compared to a ASHRAE 90.1 rated unit of 53% +/- 5% and compared to an advanced DX-DOAS of 32% +/- 5%. The field campaign remains ongoing.

INTRODUCTION

Liquid desiccants have long promised energy reductions for applications requiring delivery of air in the dewpoint range of 35-55ºF (2–13ºC), but innovators have struggled to develop a packaged product capable of realizing these promises and operating reliably in the field. Desiccant systems, both solid and liquid, deliver energy savings in packaged equipment through several straightforward mechanisms. First and foremost, they enable delivery of air at relative humidity below 100% without reheating. Conventional vapor compression (VC, or direct expansion, DX) systems require cooling the process air to a dry bulb temperature approximately equal to the dew point (i.e., ~100% relative humidity). By combining a DX subsystem with a desiccant subsystem, air can be cooled to the target enthalpy, typically at saturation, rather than the target dew point by the DX subsystem. The desiccant subsystem can then dry, and typically, reheat the air to reach the desired supply dry bulb and dew point. This allows the system to "cut the corner" as shown in the psychrometric chart, Figure 1, below. Additional energy savings are realized because this target enthalpy is reached without requiring as low an evaporator temperature in the DX system resulting in a higher coefficient of performance for the DX compressor. Often this is offset by the need to harvest high-quality (i.e., high temperature) heat from the DX's compressor in sufficient quantity to regenerate the desiccant. This requirement can push the DX operation to a higher-than-required condensing temperature; liquid desiccant systems, however, require lower quality heat than solid desiccant systems and therefore are far less susceptible to this requirement. Accordingly, liquid desiccants **require no external heat input for regeneration of the desiccant:** this is why they can provide the most efficient delivery of dry air (i.e., more than double current AHSRAE 90.1 code levels) for applications requiring air in the dewpoint range of 35-55ºF (2–13ºC).

Philip Farese is the CEO of Mojave Energy Systems (Mojave). **Rachel Ellman** is the Vice President of Engineering at Mojave. **Aaron Meles** is the Vice President of Product at Mojave. **Kris Toivola** is a Senior Reliability Engineer at Mojave.

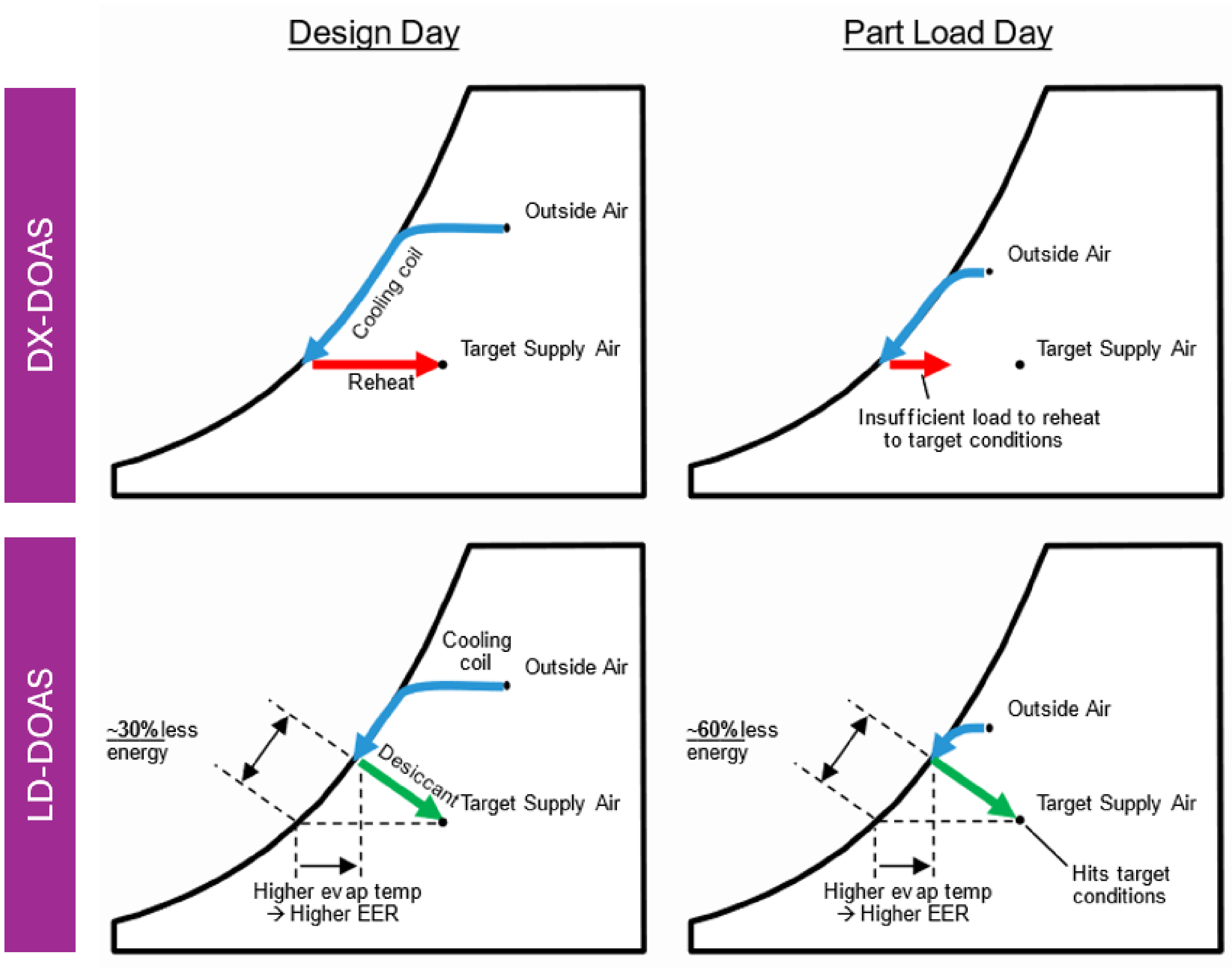

Figure 1: Desiccant saves energy by "cutting the corner" on the psychrometric chart to reduce energy use: they stop overcooling and improve compressor COP.

Multiple systems have been designed deploying solid desiccant with these energetic benefits in mind. Thousands of units have been deployed in various applications and have met with some success. Liquid desiccants promise to be even more energy efficient as the airside pressure drop, regeneration energy needs, and parasitic energy (e.g., motors) of solid desiccant systems are reduced or eliminated in a liquid desiccant system. Further, challenges of working with solid desiccants, and the resulting reliability concerns, have led innovators to explore liquid desiccant systems. Liquid desiccants eliminate two key challenges encountered with solid desiccants:

- Liquid desiccants require lower maintenance: the desiccant lasts the 15-year life of the unit and requires no additional maintenance beyond filter servicing. This contrasts with solid desiccants that typically require wheel replacement after several years of operation and annual replacement of expensive belts.
- Liquid desiccants operate consistently in all conditions: because of the more moderate regeneration requirements, liquid desiccant systems can continue to reliably deliver air at the desired supply conditions at times where solid desiccants have traditionally struggled (e.g., the more mild but damp shoulder seasons of spring and fall where conditions are often ~100% RH at cooler dry bulbs of 65-76ºF or 18-24ºC).

These benefits have led multiple innovators to pursue liquid desiccant systems.

Developing a liquid desiccant system that can work reliably in commercial applications requires addressing three fundamental challenges: liquid desiccant handling, control systems design, and air-liquid interface design (i.e., prevention of carryover of the desiccant into the airstream). Detailed discussion of these topics is beyond the scope of this paper. However, to briefly address these points: prior to the field campaign described herein, the system underwent rigorous accelerated lifetime testing; control development and laboratory evaluation; and design and testing of the air-liquid interfaces. These efforts demonstrated a unit that, in the laboratory, showed reliable operation across a range of environmental conditions simulated in a psychrometric chamber with zero carryover. Further, this unit demonstrated sufficient energy savings with a measured ISMRE2 of over 8 lbs./kWh (3.6 kg/kWh): more than double the current ASHRAE 90.1 standard of 3.8 lbs./kWh (1.7 kg/kWh) for air-cooled systems without energy recovery. Having developed a packaged product capable of realizing the energy efficiency potential of a liquid desiccant system and of operating reliably in a laboratory system, the system was moved to field testing.

The liquid desiccant system tested represents a packaged unit with an integrated heat pump and liquid desiccant system packaged into a single unit requiring only a single point electrical connection. No building utilities are required for cooling the air (i.e., chilled water) or regenerating the desiccant (i.e., hot water or natural gas). Figure 2 details the system architecture.

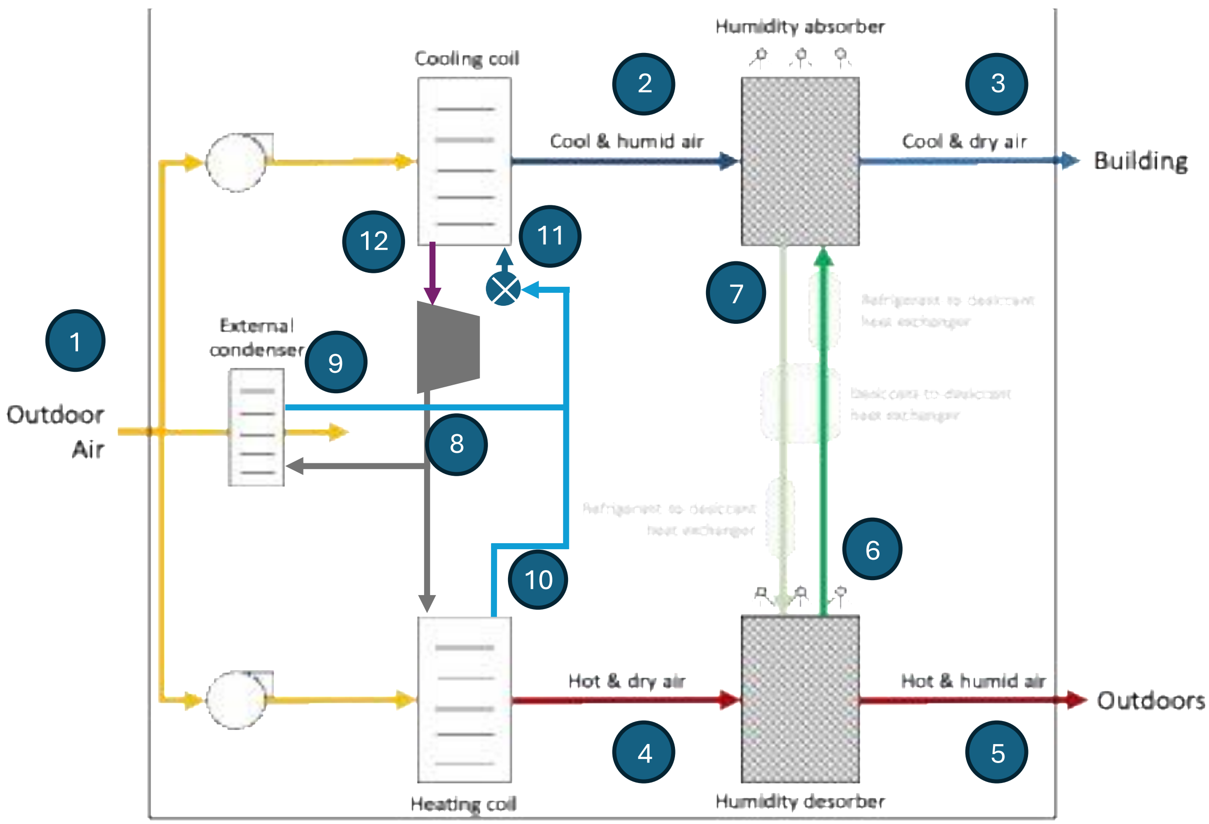

Figure 2: System architecture showing air, refrigerant, and liquid desiccant handling systems inside packaged unit.

Outdoor air enters the unit [1] and is passed through both the process side (top airstream) and regeneration side (bottom air stream) in appropriate amounts. The process air is delivered to the building at a flow rate that is set to meet the design conditions and can be fixed or variable as required by the customer. The regeneration air stream flow rate is set by the system controller to regenerate the desiccant at the desired rate.

The process air first passes through the cooling coil where it is cooled, typically to saturation (100% relative humidity, RH) and some moisture extracted [2]. The supply air stream then passes through the humidity absorber where it is dehumidified by the liquid desiccant. The dehumidification process is adiabatic, thus the air increases in temperature as the latent heat is converted to sensible heat. It will exit the absorber [3] at the desired supply air conditions.

The regeneration air passes through the heating coil where it is heated typically to 100-120^{o}F (40-50^{o}C) [4] using heat from the refrigerant that was picked up in the evaporator coil. This temperature is set by the controller to achieve the required desiccant regeneration rate in the humidity desorber. **No additional heat input is required for regeneration: the desiccant is entirely regenerated with waste heat from the condenser.** The air then passes through the humidity desorber where it heats the desiccant and evaporates off the collected moisture from the desiccant leaving the unit to the outside [5] carrying away the heat and humidity that has been extracted from the supply air. The refrigeration loop is nearly standard, though compressor discharge [8] gas passes to two condensers where required heat is added to the regeneration air [10] and excess heat is rejected [9]. Expansion of gas [11] and cooling the process air in the evaporator [12] are standard.

CAMPAIGN GOALS, SITE SELECTION, AND MODELING

The campaign goals were to demonstrate greater than 90% uptime with 90% of rated energy performance across a wide variety of conditions, ideally providing some hours at all psychrometric conditions under which the units would be expected to operate. Rated energy performance was evaluated using two metrics: Integrated Moisture Removal Efficiency measured in the lab and calculated as per ASHRAE 90.1 and comparison between actual field performance at measured ambient conditions and the performance of a digital twin at those conditions. ISMRE represents the Moisture Removal Efficiency (MRE) measured at four conditions and weighted as per AHRI Standard 920:

$$MRE = \frac{Moisutre\ remove\ (lbs\ or\ kg)}{Total\ Enegy\ Input\ (kWh)}$$

To these ends, each liquid desiccant air conditioner (LDAC) was configured with inlet and outlet air sensors (ACH A/RH2-1K-2W-O) and three-phase power monitors (Phoenix Contact Empro EEM-MB370) to allow continuous performance monitoring and modeling. Ideally, conventional DX or solid desiccant systems would be co-located and similarly monitored to allow comparison of conventional units under identical conditions, but site requirements prevented us from doing so. Accordingly, the field campaign was designed to test unit performance along the following dimensions:

1. Energy Use/ Efficiency: The energetics of each LDAC was compared against the modeled Moisture Removal Efficiency (MRE) and energy use.
2. Reliability: Unit reliability was measured as uptime percentage. This was defined as the number of hours the unit was able to operate divided by the number of hours the building owner/operator called on the unit to operate.

Site selection

Site selection was driven primarily by logistic requirements: identifying a site that was available and willing to support a novel product while trying to subject the units to a wide variety of conditions. All units were configured as 100% outdoor air units. For the initial field campaign, comprising three 1000 CFM engineering validation units (EV-1, EV-2, and EV-3), three sites were identified: two offices in Florida and one maintenance facility in Michigan. For the extended campaign, two production design units (PV-A and PV-B) were installed: one in a warehouse in Texas and one in a university laboratory in Florida. Details of the installations, including leaving air temperature (LAT) and dew point (LADP) are shown in Table 1. Note that these conditions are met without requiring hot gas reheat as the liquid desiccant concentration is set in unit operations to provide the target leaving air temperature and dew point without using hot gas reheat.

Table 1. Field Installations

Unit	Location	Application	Refrigerant	Start	CFM	LAT, °F (°C)	LADP, °F (°C)
EV-1	Tampa, FL	Office	R-410A	4/12/23	1000	65 (18)	49 (9.4)
EV-2	Midland, MI	Maintenance shop	R-410A	6/2/23	1000	70 (21)	45 (7.2)
EV-3	Tampa, FL	Office	R-410A	8/18/23	1000	69 (21)	50 (10)
PV-A	Houston, TX	Warehouse	R-454B	12/13/23	3000	70 (21)	54 (12)
PV-B	Orlando, FL	Laboratory	R-454B	3/22/24	5000	65 (18)	52.5 (11)
PV-B					4000	60 (16)	47.5 (9)

Performance modeling methodology

Performance calculations require careful measurement and a basis of comparison. At deployment, careful airflow measurements are taken at each site to +/-10% accuracy; as is typical in HVAC, airflow measurements represent the greatest area of uncertainty. The temperature/ humidity sensors indicated above present uncertainties better than +/- 5% as do the power meters deployed. Data are gathered using cellular data-enabled IoT logging platform with timesteps of 10 minutes typically used for analysis. Additional operating parameters are also gathered including refrigeration system parameters (evaporator and condenser temperature set points and measurements; suction superheat; expansion valve opening percentage; variable compressor speed), air parameters (fan speed settings, speed measurements, and power draws), and desiccant parameters (pump speeds and power draw, desiccant quantity). These were used to confirm unit operations and performance.

A digital twin performance model programmed in Engineering Equation Solver (EES) was used to develop the system that represented our basis for comparison. This physics-based model allows predictions of capacity and energy use for a given set of ambient and supply air conditions (i.e., temperature and humidity). It incorporates the specifics of the air flow, VC system (i.e., compressor, evaporator and condenser coils), and desiccant system (i.e., absorber and regenerator geometries, desiccant chemical properties and flow rates). This model was used to evaluate cost and performance tradeoffs in system development and to predict installed efficiency and unit health in the field.

The accuracy of the digital twin model was confirmed and improved through rigorous lab testing across a wide range of operating conditions in our psychrometric chamber: the resultant model predicts performance to within +/- 5% of the measured laboratory results. Across the five units in the field campaign, the rated ISMRE2 ranged from 8.3 lbs./kWh (3.8 kg/kWh) to 10.7 lbs./kWh (4.9 kg/kWh), with an average of 9.5 lbs./kWh (4.3 kg/kWh), as shown in table 2 below. (Note that PV-B was tested at both 5000 CFM and 4000 CFM, with results reported for each in Table 2.) Table 2 also shows the annual energy savings calculated by comparing the indicated ISMRE2 to ISMRE2 of a unit that meets the requirements of ASHRAE 90.1 (3.8 lbs./kWh, 1.73 kg/kWh) and to an advanced DX-DOAS as some building owner/operators instal today (5.6 lbs./kWh, 2.5 kg/kWh). This assumes that the ISMRE2 rating accurately reflects the typical annual energy use of each system.

Table 2. Unit Efficiency Ratings and Expected Energy Savings

	Location	ISMRE2		Total capacity	Moisture removal	Energy savings (%) to	
System design:		kg/kWh	Lbs/kWh	Tons	Lbs./hr (kg/hr)	ASHRAE 90.1	Advanced
EV-1	Tampa, FL	3.8	8.3	9	57 (26)	54%	33%
EV-2	Midland, MI	3.8	8.3	9	57 (26)	54%	33%
EV-3	Tampa, FL	3.8	8.3	9	57 (26)	54%	33%
PV-A	Houston, TX	4.3	9.5	25	117 (53)	59%	40%
PV-B, 5000 CFM	Orlando, FL	4.3	9.5	40	234 (106)	51%	26%
PV-B,4000 CFM		4.6	10.7	40	234 (106)	44%	19%

Actual energy savings will depend on the ambient (inlet) and supply conditions presented during the field campaign and are expected to vary from the estimates derived from the ISMRE2 ratings above.

FIELD CAMPAIGN

The initial field campaign began with EV-1 deployment and commissioning in Florida in April, 2023. EV-2 was deployed in Michigan in May 2023. EV-3 was deployed in Florida in August 2023 (see Table 1). EV-2 was deactivated for winter by the building operator on October 11th, 2023 as outdoor conditions dropped to the point that air conditioning was no longer needed. It was subsequently restored to operation on May 30th, 2024 with no issue and continues to operate. EV-1 was deactivated by the building owner on November 20th ahead of the Thanksgiving holiday. Data collection on EV-3 continued until 10,000 hours were accumulated the week of December 7th, 2023.

The extended stage of the campaign continued with these three units operating as requested/ required by the building operators. Added to the fleet were PV-A (deployed in Houston in December 2023) and PV-B (deployed to Florida in March 2024). At the conclusion of this second stage of the field campaign, operating time across the five units represented 31,673 operating hours.

Because the buildings conditioned were occupied or in use by the building owner/ operator, we allowed interruptions to the field campaign if needed. Two meaningful interruptions occurred. First, the occupants of the building where EV-1 was located underwent a business merger that disrupted use of the building and of the field campaign. Accordingly, the customer called on the unit intermittently in the period between December 1st and February 29th and ultimately deactivated the unit. Hours where the customer had deactivated the unit were excluded from the uptime and operating hour data. Second, it was determined that PV-A was slightly undercharged with refrigerant and follow-up service was not performed correctly, leading to overcharge which may have caused the leak in the system that ultimately led to failure. Downtime due to this event is still included in Tables 4 and 5. Figure 3 shows the distribution of the 31,673 hours gathered and the ambient/inlet conditions (temperature and humidity ratio) on the psychometric chart showing that we achieved our goal of operating at all psychrometric conditions under which the unit would be expected to operate.

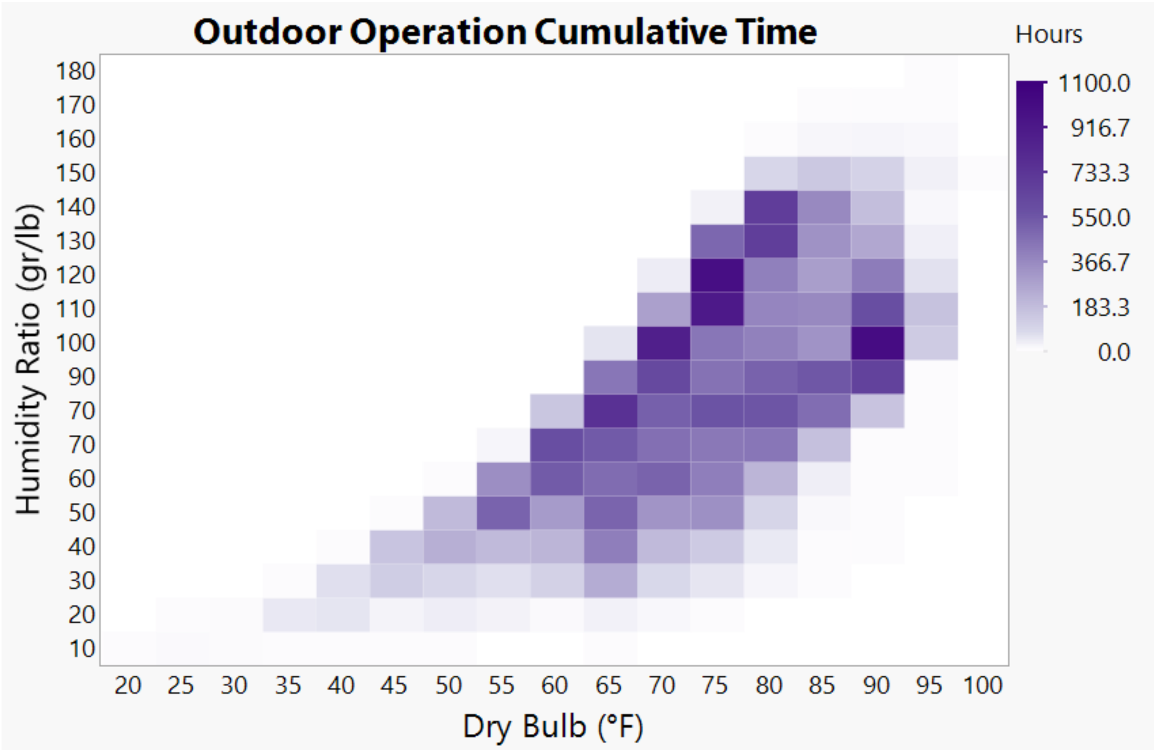

Figure 3: Distribution of operating hours for the full field test shown on the psychrometric chart.

PERFORMANCE ANALYSIS

Overall, the units met all field campaign goals. As detailed in each subsection below:

1. Efficiency was within +/- 4% of our estimates across all operating conditions observed, and
2. The fleet presented 97% uptime, broadly in line with industry expectations; the liquid desiccant subsystem displayed 99% uptime.

Efficiency

Each unit was instrumented as described above to capture power, inlet air conditions, outlet air conditions, and operating indicators using a cellular data-enabled IoT logging platform with data recorded every 10 minutes. The digital twin's modeled performance was calculated given the inlet and outlet conditions for the Mojave unit. The total power consumption data are compared against this model for PV-A in figure 4 below. Across the fleet for the full field campaign, the median ratio of Mojave's total power to the modeled power was 97%, the mean was 101%, and the 25%-75% confidence range was 82-104%. There is no sign of performance degradation over the test period with an upper limit of 8%/year (95% confidence) level. This estimate agrees with the model given the measurement uncertainties in airflow parameters and model fidelity demonstrating the accuracy of the ISMRE ratings measured in the laboratory, namely 9.5 lbs./kWh (4.3 kg/kWh).

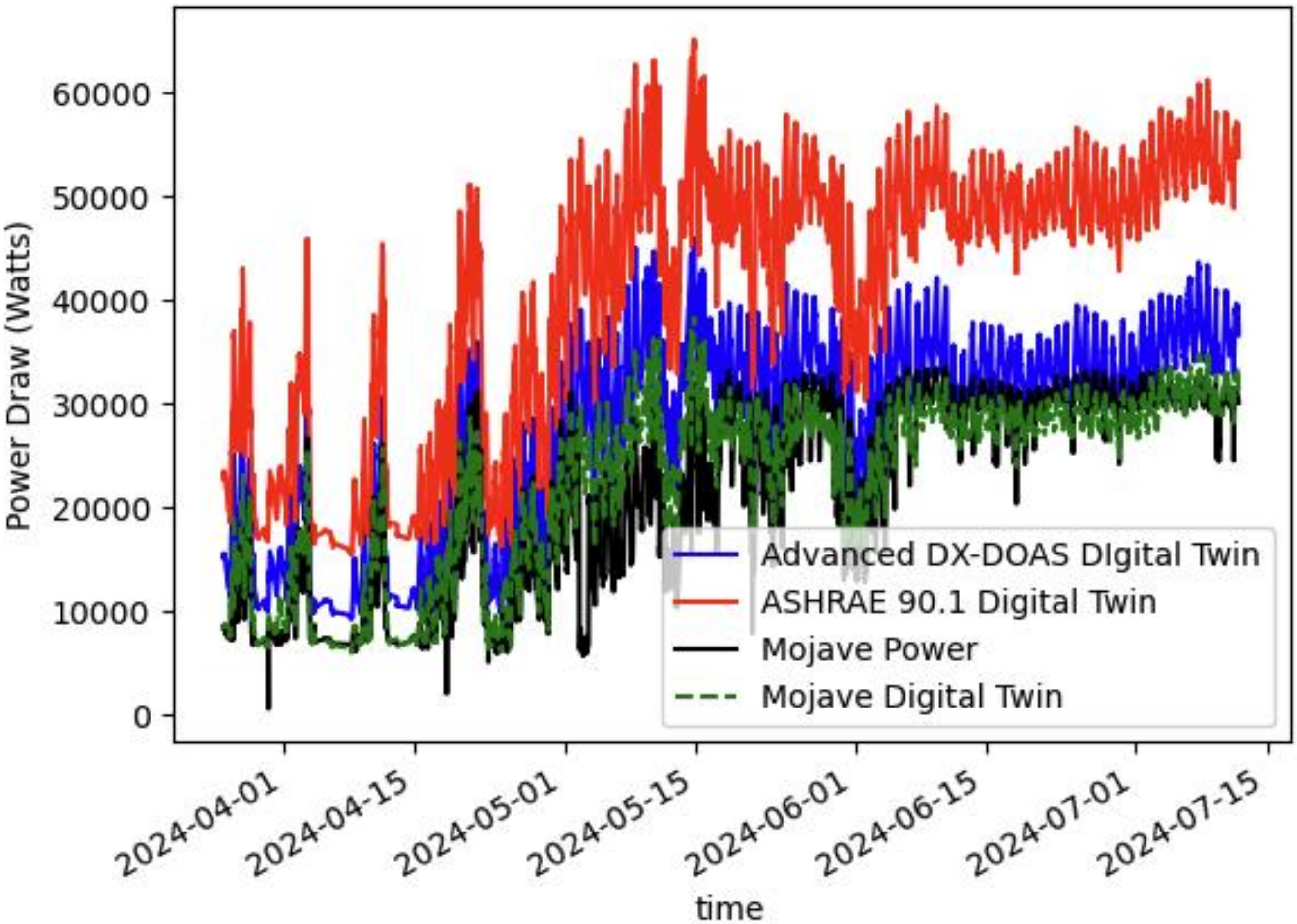

Figure 4: Time series plot of DV-B total power consumption (Watts) compared to modeled power consumption for the PV-A LDAC, and two DX-DOAS units: one at the ASHRAE 90.1 efficiency rating (3.8 lbs./kWh, 1.73 kg/kWh) and one an advanced DX-DOAS efficiency rating (5.6 lbs./kWh, 2.5 kg/kWh).

Digital twin models were used to compare the measured LDAC performance to DX-DOAS. The LDAC model is proprietary but similar to those developed by Woods and Kozubal. The DX physics-based python models select parameters to characterize a DX-DOAS system (e.g., evaporator and condenser approach temperature, compressor coefficient of performance, pressure drop, and fan efficiencies) to match the ISMRE performance desired. Specifically, condenser and evaporator approach temperatures set the high- and low- side of a compressor with a specified efficiency; thermodynamic balance determines the required fan energy to meet conditions. Two models were developed, one representing the performance of a DX-DOAS with an ISMRE2 of 3.8 lbs./kWh (1.73 kg/kWh, i.e., matching ASHRAE 90.1 minimums) and one with an ISMRE2 of 5.6 lbs./kWh (2.5 kg/kWh, i.e., advanced DX-DOAS). Energy use of each actual hour of operation (i.e., entering and leaving air temperature and dew point) is simulated and compared to actual energy use. Performing this analysis for all units over the full campaign and comparing to both the Mojave performance model and the modeled performance for code and typical units provides the results in table 3 below. This table demonstrates that a liquid desiccant air conditioner can realize 48%-60% annual energy savings as compared to a DX-DOAS that meets ASHRAE 90.1. Compared to an advanced DX-DOAS unit, such a LDAC will deliver ~32% +/-5% annual energy savings on average. As shown in figure 1, this is the result of the desiccant system stopping overcooling and improving compressor COP across all operating conditions (i.e., design days and part load days).

Table 3. Energy Savings Estimates from Field Campaign

Unit	Performance ratio (actual to digital twin)	Energy savings (vs. ASHRAE 90.1 unit)	Energy savings (vs. advanced DX-DOAS)
EV-1	0.97		
EV-2	1.04	60%	37%
EV-3	1.02	52%	27%
PV-A	0.97	52%	36%
PV-B	1.04	48%	29%
Average	1.01	53%	32%
Standard deviation	0.04	5%	5%

Reliability

Multiple metrics exist to characterize the reliability of a mechanical system. For simplicity we consider as our primary reliability metric the operating "uptime", defined as follows. Given a building operator (e.g., an occupant, facilities manager, building management system, etc.), we define the number of hours that the building operator called on the unit to operate as "operating hours". Downtime represents the number of hours that the building operator called on the unit to operate but it was unable to do so (i.e., was "down"). The difference between operating hours and downtime we define as "uptime". Uptime divided by the number of operating hours yields the operating hours percentage. Ultimately this represents a metric to characterize the comfort of building occupants. As shown in Table 4 below, the field test demonstrated a system uptime of 97%. Downtime is typically caused by issues or failures with the vapor compression (DX) subsystem, desiccant subsystem, or other subsystems (e.g., air handling, electrical, etc.). As we are ultimately interested in the reliability of the liquid desiccant subsystem specifically, we also calculate the uptime of that subsystem. This includes as desiccant subsystem downtime only issues or failures that prevent the desiccant system from operating to specification. Table 4 also provides the desiccant subsystem uptime: note that the uptime of 98.4% (1.6% downtime) is greater than the system uptime (2.6% system downtime) revealing that desiccant issues and failures represent the minority of downtime hours.

Table 4. Overall Reliability Results from Full Field Campaign

Unit	System Uptime (hours)	Operating (hours)	System Uptime (%)	Desiccant Subsystem Uptime (hours)	Desiccant Subsystem Uptime (%)
EV-1	7,419	7,602	97.6%	7,521	98.9%
EV-2	4,945	5,081	97.3%	4,948	97.4%
EV-3	8,273	8,732	94.7%	8,628	98.8%
PV-A	5,972	6,292	94.9%	6,149	97.7%
PV-B	3,919	3,966	98.8%	3,926	99.0%
Total	30,705	31,673	96.9%	31,172	98.4%

Further, it is instructive to separately look at the initial (or total) number of failures/ issues as well as the number of failures/ issues once a unit has reached steady state because failures typically follow the "bathtub curve". To isolate this, Table 5 shows the reliability data for the final 90 days of the test campaign. As noted above, PV-A experienced a failure of the vapor compression (DX) subsystem; this is the only failure in the past 90 days (8,640 operating hours) and accounts for the majority (73%) of the downtime of the field test over this period. Including this as downtime reveals a system uptime percentage of 97.2%. Calculating the desiccant subsystem uptime, which excludes this DX failure from the desiccant subsystem downtime total, reveals a **desiccant subsystem uptime of over 99%. This demonstrates that with proper engineering and manufacturing a liquid desiccant air conditioner can equal or exceed the field reliability of a conventional vapor compression air conditioner.**

Table 5. Overall Reliability Results from Final 90 Days of Field Campaign

Unit	System Uptime (hours)	Operating (hours)	System Uptime (%)	Desiccant Subsystem Uptime (hours)	Desiccant Subsystem Uptime (%)
EV-1	Not active	Not active	N/A	Not active	N/A
EV-2	2,147	2,160	99.4%	2,147	99.4%
EV-3	2,154	2,160	99.7%	2,154	99.7%
PV-A	1,981	2,160	91.7%	2,158	99.9%
PV-B	2,114	2,160	97.9%	2,114	97.9%
Total	8,396	8,640	97.2%	8,573	99.2%

CONCLUSION

Five liquid desiccant DOAS systems have been field tested in a variety of climates, accumulating over 30,000 hours of field operation. Prior to deployment each unit had its ISMRE measured, demonstrating ISMREs between 8.3 and 10.7 lbs./kWh (3.8 to 4.6 kg/kWh) with an **average ISMRE of 9.5 lbs./kWh (4.3 kg/kWh)**. Once installed in field locations in Florida, Texas, and Michigan, **the units demonstrated 53% energy savings when compared to ASHRAE 90.1 and 32% energy savings when compared to advanced DX-DOAS units**. Additionally, each unit's performance continues to match its digital twin within 4% and shows no evidence of degradation over time (below 8%/year at 95% confidence). Finally, the units operated reliably, with **97% system uptime and steady state desiccant subsystem uptime of over 99%.** All five units are still in operation, and the performance and reliability of these units continues to be monitored.

In this study, the LD-DOAS systems provided substantial energy savings while supplying neutral dewpoints (48-55°F, 9-13°C). Future work will focus on LD-DOAS's ability to supply lower dewpoint (35-45°F, 2-7°C) air than DX-DOAS. As has been previously characterized (Harriman et al, 2001), further benefits to building-wide efficiency and operability can be realized by using DOAS to provide drier-than-neutral air, which removes the need for all other sensible cooling equipment in the building to dehumidify. Demonstrating this building-wide efficiency benefit from LD-DOAS is the scope of future work.

ACKNOWLEDGMENTS

We would like to thank the various organizations that supported this effort through their funding and expertise, including the Department of Energy who supported this work through two grants: DE-EE0009682 and DE-EE0011039. These included the collaboration of the National Renewable Energy Laboratory (NREL) and Palo Alto Research Center (PARC). In particular, we thank Jason Woods and Eric Kozubal of NREL for their ongoing support. We also would like to acknowledge those organizations that participated in field testing, including Insight Partners (now Integrated Cooling Solutions), Tom Barrow HVAC Solutions, Hemlock Semiconductor, the National Aeronautic Space Administration (NASA), Johnson Space Center, and the University of Central Florida.

REFERENCES

Harriman, L.G., G. Brundrett, and R. Kittler. 2001. Humidity control design guide for commercial and institutional buildings. Peachtree Corners, GA: ASHRAE.

Myrefelt, Sonny. (2004). The reliability and availability of heating, ventilation and air conditioning systems. Energy and Buildings. 36. 1035-1048. 10.1016/j.enbuild.2004.06.010.

NIST/SEMATECH e-Handbook of Statistical Methods, http://www.itl.nist.gov/div898/handbook, September 4th, 2024, 8.1.2.4

Air Conditioning Heating, and Refrigeration Institute. 2020. Performance Rating of Direct Expansion-Dedicated Outdoor Air System Units. Arlington, VA

Woods, J and Kozubal, E. 2012. Desiccant Enhanced Evaporative Air Conditioning: Parametric Analysis and Design. Conference Paper NREL/CP-5500-54087