Solar-Powered, Liquid-Desiccant Air Conditioner for Low-Electricity Humidity Control

Energy and Water Projects
Demonstration Plan SI-0822

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Acronyms and Abbreviations

AAHX	air-to-air heat exchanger
AC	air conditioning
AFB	air force base
AFRL	Air Force Research Laboratory
AHU	air-handling unit
AILR	AIL Research
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
Btu	British thermal units
$CaCl_2$	calcium chloride
cfm	cubic feet per minute
CHP	combined heat and power
CPVC	chlorinated polyvinyl chloride
CoC	cycles of concentration
COP	coefficient of performance
DAS	data acquisition system
DOAS	dedicated outdoor air system
DoD	Department of Defense
DOE	Department of Energy
DX	direct expansion
EER	energy efficiency ratio
EISA	Energy Independence and Security Act
E.O.	executive order
EPAct	Energy Policy Act
ESTCP	Environmental Security Technology Certification Program
ft	foot
ft^2	square foot
FY	fiscal year
gal	gallon
GHI	global horizontal irradiance
gpm	gallons per minute
hr	hour
EUI	energy use intensity
HMX	heat and mass exchanger
hp	horsepower
HVAC	heating, ventilating, and air conditioning
LDAC	liquid-desiccant air conditioner
kW	kilowatt
kWh	kilowatt-hour
lb	pound
LiCl	lithium chloride
MBtu	million British thermal units
MEP	Mountain Energy Partnership
NREL	National Renewable Energy Laboratory
PLC	programmable logic controller
PPSU	polyphenylsulfone
PVC	polyvinyl chloride
RH	relative humidity
VFD	variable-frequency drive
W	watt
yr	year

ACKNOWLEDGEMENTS

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EXECUTIVE SUMMARY

Today's air-conditioning (AC) technology is primarily based on direct expansion (DX) or the refrigeration process. It is now so prevalent that it is considered a necessity for the majority of residential and commercial buildings throughout the United States. During the 100-plus years of development, DX AC has been optimized for cost and thermodynamic efficiency, both of which are nearing their practical limits. Nevertheless, AC accounts for approximately 15% of all source energy used for electricity production in the United States alone [nearly 4 quadrillion British thermal units (Btu)], which results in the release of about 343 million tons of carbon dioxide into the atmosphere every year. (DOE 2011)

The Department of Defense (DoD) occupies over 316,000 buildings and 182,000 structures on 536 military installations worldwide, and accounts for about 64% of the energy consumed by federal facilities. This makes the DoD the largest energy consumer in the United States. In Fiscal Year (FY) 2007, the DoD consumed 218 trillion Btu in site-delivered energy, 26.2 trillion Btu for AC alone. This cooling cost equates to an estimated $413 million per year. (Pacific Northwest National Lab, undated)

In hot, humid climates, conventional AC units expend excess energy to sensibly overcool the air for dehumidification. As a result, excess energy must be used to reheat the air to a more comfortable supply temperature (overcool/reheat cycle). The use of desiccant-based AC systems decouples the latent and sensible loads of an airstream, enabling higher efficiency cooling and improved thermal comfort conditions. The following is a list of criteria that can be used to identify feasible sites for liquid-desiccant air conditioner (LDAC) applications:

- Hot and humid climate – latent cooling required most of the year
- One hundred percent outdoor air ventilation requirements
- Significant reheat loads on current heating, ventilation, and AC (HVAC) system
- Heat source available or suitable installation identified for desiccant regeneration
- Current issues with humidity control – comfort, sick building syndrome, mold, etc.

The primary objective of this project was to demonstrate the capabilities of a new high-performance, liquid-desiccant dedicated outdoor air system (DOAS) to enhance cooling efficiency and comfort in humid climates while substantially reducing electric peak demand at Tyndall Air Force Base (AFB), which is 12 miles east of Panama City, Florida. The new type of LDAC invented by AIL Research (AILR) has higher thermal efficiency than any other LDAC on the market today. The technology was recently invented, and only six active units were operating at the time of this report, four of which are demonstration projects funded by the Department of Energy (DOE) with the purpose of demonstrating different applications and resolving new-product technical issues. Broader application is expected soon after technical reliability and manufacturing costs become acceptable. Seeing the technology's potential, Munters Corporation recently purchased AILR's LDAC technology and will commercialize it in areas with low thermal energy costs compared to electricity (e.g., low natural gas cost or waste-heat applications).

This was the first solar-powered demonstration of the technology. The goal of the project was to quantify energy and water consumption, solar energy utilization, and cost savings relative to DX air conditioners. The LDAC system that was installed at Tyndall AFB was a pre-commercial technology, and given that it was the first solar-powered demonstration, a fundamental objective of the demonstration was to evaluate system performance and use the lessons learned to develop design/manufacturing guidance for future commercial LDAC systems. Each demonstration of this new technology is expected to reveal technical issues related to the specific application. This demonstration was also the first to integrate the LDAC as a retrofit into an existing air handler. Lessons learned from these experiences are expected to improve product design and create a methodology for determining suitable retrofit applications.

Performance evaluation of the LDAC began in the summer of 2010. Only 3 weeks of continuous operation were recorded in 2010 due to system malfunctions and limited run-time. Roughly 5 months of operation were recorded between April and September in 2011. Table 1 describes the performance objectives that were evaluated during the demonstration.

Table 1. Performance Objectives

Performance Objective	Metric	Success Criteria	Results
Quantitative Performance Objectives			
Improve humidity control and comfort (energy efficiency)	• Hours outside psychrometric comfort zone • Chiller power • Reheat run-time	• <1% of hours outside ASHRAE summer comfort zone • Reduce chiller/reheat run-time	• Achieved but inconclusive cause • Achieved but inconclusive cause
Provide high-efficiency dehumidification (energy efficiency)	• EER • COP	• >40 (Btu/hr)/W EER • Thermal COP >0.7	• Not achieved • Achieved
Sustain high-dehumidification performance (energy efficiency and maintenance)	• Conditioner heat exchange effectiveness • Desiccant charge • Supply-air pressure drop • Conditioner cooling-water pressure drop • Projected service life	• <Once-per-year desiccant/buffer adjustment • <5% degradation of HX eff. over 3 years • Negligible increase in air/water pressure drop • Above criteria should support >10-yr service life projection	• Achieved; no degradation of desiccant during operation • Duration of performance evaluation too small to determine

Qualitative Performance Objectives			
Maintainability (ease of use)	Ability of an HVAC technician to operate and maintain the technology	A single facility technician able to effectively operate and maintain equipment with minimal training	• Many unforeseen maintenance issues occurred during initial demonstration • Many lessons learned for design and ease of operation

Performance data—including energy efficiency ratio (EER), kilowatt (kW)/ton, and coefficient of performance (COP) for 2010 and 2011—are summarized in Table 2 and Table 3, respectively. It is clear that the electrical and thermal efficiency improved throughout the summer of 2011.

Table 2. Summer 2010 (3 Weeks) Performance Summary

Date	Cooling (ton-hr)	EER [(Btu/hr)/W]	kW/ton	Solar heat (MBtu)*	COP*
7/21/10 - 8/14/10	1982	14.7	0.82	3.1	0.85

*Solar thermal generation only recorded for 3 days (7/21-7/23)

Table 3. Monthly (Averaged) Performance for Summer 2011

Month	Cooling (ton-hr)	EER [(Btu/hr)/W]	kW/ton	Solar heat (MBtu)	COP
April	667	7.8	1.54	18.1	0.44
May	1565	8.2	1.47	39.9	0.5
June	1837	12.4	0.97	35.4	0.62
July	1142	14.6	0.82	19.4	0.71
August	1916	18.8	0.64	32.2	0.71
September	1300	15.1	0.79	26.7	0.73

Table 4 summarizes the displaced load on the existing chiller and the approximate energy and cost savings from the LDAC.

Table 4. Energy and Cost Savings from the LDAC in 2011

Month	Cooling (ton-hr)	Chiller Elec. (kWh)	LDAC Elec. (kWh)	Elec. Savings (kWh)	Elec. Cost Savings ($)
April	667	890	1,026	-137	-14
May	1,582	2,110	2,325	-215	-21
June	1,837	2,449	1,774	676	68
July	1,239	1,652	1,131	521	52
Aug	1,916	2,554	1,223	1,331	133
Sept	1,333	1,778	1,099	678	68

The total cost savings for the 2011 cooling season was $321. The installed costs for the solar thermal system were $170,000, and the installed costs for the LDAC components were $40,000, for a total installed cost of $210,000 and a simple payback of 654 years. Because this was a pre-commercial system, the simple payback is not indicative of the payback period of a commercial system. If the system would have operated per design intent, the cost savings would be substantially higher. In addition, in building types with electric reheat, the zone-level overcooling and reheat savings dwarf the energy savings from the mechanical chiller. Reheat energy use in hospitals, for example, has been documented to account for over 30% of the total energy use. Finally, when the system is coupled with solar thermal, the solar thermal component becomes the most expensive part of the system, and solar incentives or high utility rates are required to offset the increased costs of the solar thermal system.

In general, the LDAC system did not perform as well as expected due to design, installation, and operation issues. Consequently the project's focus was necessarily changed to focus on discovery of technical issues with this new emerging technology. Many of the issues arose because the installation had many unique features including the following:

- The demonstration was the first combination of solar heat with this type of LDAC system.
    - Due to initial budgetary constraints, the LDAC relied solely on solar heat, with no natural gas backup to ensure that the unit operated throughout the cooling season. A properly designed system that uses solar heat will have backup. Due to this, the system did not achieve peak-cooling capacity for significant hours of operation. Because the system largely has static electrical power draw, this resulted in a low average EER.
    - The solar field design and LDAC system design were not tightly coordinated by the prime installation contractor (Regenesys). This resulted in a design that did not consider the frequency and duration of stagnation periods for the solar field. The collector design was not designed to withstand more than about two stagnations per year. Furthermore, the collector system was not initially designed to withstand the massive volume of steam from these collectors when stagnation occurred. The solar field required significant redesign. The end result was

workable for the demonstration, despite being problematic and suboptimal in operation.

- The demonstration was the first to create a split system where the conditioner and regenerator were contained in separate packages and separated by a distance of approximately 120-foot (ft). This technical challenge resulted in a suboptimal pumping design because of the necessary pump size to transfer desiccant this distance. Future designs should reduce the distance from the regenerator and conditioner.

- This demonstration was the first to have 10 hours of desiccant storage using calcium chloride ($CaCl_2$). Tuning the storage to achieve optimal efficiency was required. The desiccant charge and the tank's low and high levels have significant impact on efficiency, capacity, and solar utilization. These variables were tuned as the demonstration progressed.

- This demonstration required the placement of the conditioner unit about 100 feet from the outdoor intake to the building. This required significant fan power to move the air from the mechanical yard to the building. Future designs and applications should consider the duct length reduce the duct run from the conditioner to the outdoor air intake as much as possible.

- The demonstration did not treat 100% of the outdoor air, thus limiting the benefit to energy savings from offset cooling. In order to offset the reheat for such an installation, a system should be designed to ensure that the LDAC meets a significant portion of the latent load. Typically, the LDAC can meet 100% of a building's latent load if designed to treat 100% of the outdoor air.

This report outlines lessons learned that should be applied to future projects in order to ensure successful design, installation, and operation of a solar-powered LDAC system.

At the end of 2011, the LDAC technology was sold to Munters Corporation, one of the largest HVAC manufacturers in the United States. The first demonstration of a commercial LDAC system is being evaluated at the Coral Reef Fitness and Sports Center on Andersen AFB in Guam. A 6,000 cubic-feet-per-minute (cfm) conditioner was designed for this system. The power requirements per ton of cooling for the existing building level chiller and LADC are 1.05 kW/ton and 0.3 kW/ton, respectively. Note that the power requirement of the chiller does not account for the chiller water pumps, so the power requirement may be slightly greater in reality. The system is designed with an evacuated-tube solar thermal field supplying 80% of the thermal power and a backup diesel-powered boiler providing 20% of the thermal power. The system is expected to reduce HVAC energy use by 34% and save $145,395 per year with an estimated simple payback of 11.6 years.

1.0 INTRODUCTION

1.1 BACKGROUND

Today's AC is primarily based on the DX or refrigeration process, which was invented by Willis Carrier more than 100 years ago. It is now so prevalent and entrenched in many societies that it is considered a necessity for maintaining efficient working and living environments. DX AC has also had 100-plus years to be optimized for cost and thermodynamic efficiency, both of which are nearing their practical limits. However, the positive impact of improved comfort and productivity does not come without consequences. Each year, AC accounts for approximately 15% of all source energy used for electricity production in the United States alone (nearly 4 quadrillion Btu), which results in the release of about 343-million tons of carbon dioxide into the atmosphere every year. (DOE 2011)

R-22 (Freon) as a refrigerant for AC is quickly being phased out because of its deleterious effects on the ozone layer. The most common remaining refrigerants used today (R-410A and R-134A) are strong contributors to global warming; their global warming potentials are 2,000 and 1,300, respectively. (Owen 2010) Finding data on refrigerant release rates for air conditioners is challenging as they are generally serviced only when broken, and refrigerant recharge is not accurately accounted for. The limited data that does exist indicates that typical refrigerant release rates for supermarket refrigeration equipment are 10% to 15% per year. (Baxter et al. 1998) A typical residential-size AC unit may contain as much as 13 pounds of R-410A, and a 10-ton commercial AC will contain as much as 22 pounds.

Water is not commonly considered to be a refrigerant, but the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) recognizes it as the refrigerant R-718. Evaporative cooling uses the refrigerant properties of water to remove heat the same way DX systems use the refrigeration cycle. Water evaporates and drives heat from a first heat reservoir; water vapor is then condensed into a second reservoir. The water used in this process is delivered to the building as a liquid via the domestic water supply. Evaporative cooling is so efficient because the Earth's atmosphere and nature cycles, rather than a compressor and condenser heat exchanger, perform the energy-intensive process of recondensing the refrigerant.

The National Renewable Energy Laboratory's (NREL's) thermally activated technology program has been working closely with AILR as an industry partner for more than 15 years to develop an LDAC. The technology uses liquid desiccants to enable water as the refrigerant in lieu of chlorofluorocarbon-based refrigerants to drive the cooling process. The desiccants are strong saltwater solutions. In high concentrations, desiccants can absorb water from air and drive dehumidification processes; thus, evaporative cooling devices can be used in novel ways in all climates. Thermal energy dries the desiccant solutions once the water is absorbed. LDACs substitute most electricity use with thermal energy, which can be powered by many types of energy sources, including natural gas, solar thermal, biofuels, and waste heat. The benefits include generally lower source energy use, much lower peak-electricity demand, and lower carbon emissions, especially when a renewable fuel is used.

The LDAC technology deployed in this demonstration was invented by AILR, and was the result of collaborative effort with NREL, and was funded by DOE. The LDAC technology developed by AILR is the result of a 10-year, $5 million DOE R&D effort to increase the efficiency of the LDAC technology on the market and decrease maintenance concerns related to legacy problems with desiccant carryover into the product airstream. The technology is emerging and at the writing of this report, six active demonstrations had been deployed. Munters Corporation has seen the promise of the technology and has purchased the rights from AIL Research. The six active demonstrations are focused on providing cooling to grocery stores where the benefits from drying the space to sufficient levels reduce refrigeration evaporator-coil frosting due to water condensation and freezing. Energy is reduced by less defrosting and a lower load on the refrigeration system. Munters Corporation has taken on the task to manufacture the LDAC technology. The demonstration to date, including the Tyndall demonstration, has shown a critical level of reliability of the LDAC system and identified points of improvement. The sale of the technology shows that Munters Corporation is satisfied with the current state of reliability and willing to commercialize it.

AC is also the single largest contributor to peak demand on electric grids and is a primary cause of grid failure resulting in blackouts. Power generators and electric air conditioners are least efficient at high ambient temperatures, when cooling demand is highest, leading to increased pollution, excessive investment in standby generation capacity, and poor utilization of peaking assets. This LDAC approach—the result of a 10-year, $5 million DOE R&D effort—increases the efficiency of LDAC technology on the market and decreases maintenance concerns related to legacy problems with desiccant carryover into the product airstream.

1.2 OBJECTIVE OF THE DEMONSTRATION

The primary objective of this project was to demonstrate the capabilities of a new high-performance, liquid-desiccant DOAS to enhance cooling efficiency and comfort in humid climates while substantially reducing electric peak-demand. This was the first solar-powered demonstration of the technology. The goal of the project was to quantify energy and water consumption, solar energy utilization, and cost savings relative to DX air conditioners. The LDAC system installed at Tyndall was a pre-commercial technology and given that it was the first solar-powered demonstration, a fundamental objective of the demonstration was to evaluate the performance of the system and use the lessons learned to develop design/manufacturing guidance for future commercial LDAC systems.

At the end of 2011, the LDAC technology was licensed to Munters, one of the largest HVAC manufacturers in the United States, and the first demonstration of a commercial LDAC system is being evaluated on the Coral Reef Fitness and Sports Center on Andersen AFB in Guam.

1.3 REGULATORY DRIVERS

The DoD Environmental Security Technology Certification Program awarded this new technology demonstration project as a means to identify programmatic changes that could be applied to the design and construction of energy-efficient, DOAS AC systems for humid environments. A new low-energy use LDAC unit could be implemented throughout ASHRAE climate zones 1, 2, and 3 to help the agency meet or exceed the various requirements set forth in Executive Order (E.O.) 13423, the Energy Policy Act (EPAct) of 2005, and the Energy Independence and Security Act (EISA) of 2007.

EPAct 2005 requires the U.S. secretary of energy to ensure that not less than 7.5% of total electricity consumed by the federal government comes from renewable sources in FY 2013; and thereafter, to the extent economically feasible and technically practicable in FY 2013, and thereafter, of the total electricity consumed by the federal government comes from renewable energy. If the thermal portion of the LDAC unit is driven by a solar thermal source, this technology would help DoD meet its renewable energy goals.

The key features of EISA 2007 that pertain to this technology are outlined in section 431 and requires a reduction in energy use intensity (EUI) [1,000 (k)Btu/square foot ($ft^{2)}$/year (yr)] of federal buildings by 3% per year, from a 2003 baseline, resulting in a 30% reduction in EUI by 2015. The EISA 2007 legislation has superseded all previous EUI reduction mandates.

E.O. 13423 provides requirements for water conservation at federal facilities, mandating federal agencies reduce potable water consumption intensity 2% annually through FY 2020. This would result in a 26% reduction by the end of FY 2020, relative to a FY 2007 baseline. E.O. 13514 also mandates a reduction in industrial, landscaping, and agricultural water consumption by 2% annually, or 20% by the end of FY 2020, relative to a FY 2010 baseline.

The LDAC unit can substantially reduce energy use and peak demand, which will help meet EISA 2007 requirements, but it also has the potential to increase potable water consumption, which will be detrimental to the E.O. 13514 requirements. Each DoD base is encouraged to try to identify alternative sources of cooled water for the conditioner, such as geothermal-based cooling.

2.0 TECHNOLOGY DESCRIPTION

2.1 TECHNOLOGY OVERVIEW

Desiccants reverse the paradigm of standard DX AC by first dehumidifying and then sensibly cooling the outside airstream to meet a given cooling load. Desiccant at any given temperature has a water-vapor pressure equilibrium that is roughly in line with constant relative humidity (RH) lines on a psychrometric chart, as shown in Figure 1. The green lines show the dehumidification potential for two common types of liquid desiccants: lithium chloride (LiCl) and $CaCl_2$. If the free surface of the desiccant is kept at a constant temperature, the ambient air will be driven to the dehumidification potential line. If used with an evaporative heat sink at temperatures between 55°F and 85°F, the air can be significantly dehumidified, and dew points less than 32°F are easily achieved. The blue arrow in Figure 1 shows the path of ambient air driven to equilibrium with $CaCl_2$ with the use of an evaporative heat sink. At this point, the air can be sensibly cooled to the proper supply temperature. This type of desiccant AC system decouples sensible and latent cooling by controlling each independently.

During the dehumidification process, the liquid desiccant (about 43% salt concentration by weight in a water solution) absorbs water vapor in an exothermic reaction. The heat released by the desiccant is carried away by a heat sink, usually cooled water from a cooling tower. As water vapor is absorbed from the ambient air, it dilutes the liquid desiccant, and decreases its vapor pressure and its ability to absorb additional water vapor. Lower concentrations of desiccant come into equilibrium at higher ambient air RH levels. Dehumidification can be controlled by the desiccant concentration supplied to the device. The outlet humidity level of the processed ambient air can be controlled by the desiccant concentration and/or the flow of highly concentrated desiccant. The latter allows the highly concentrated desiccant to quickly be diluted and thus “act” as a weaker desiccant solution in the device.

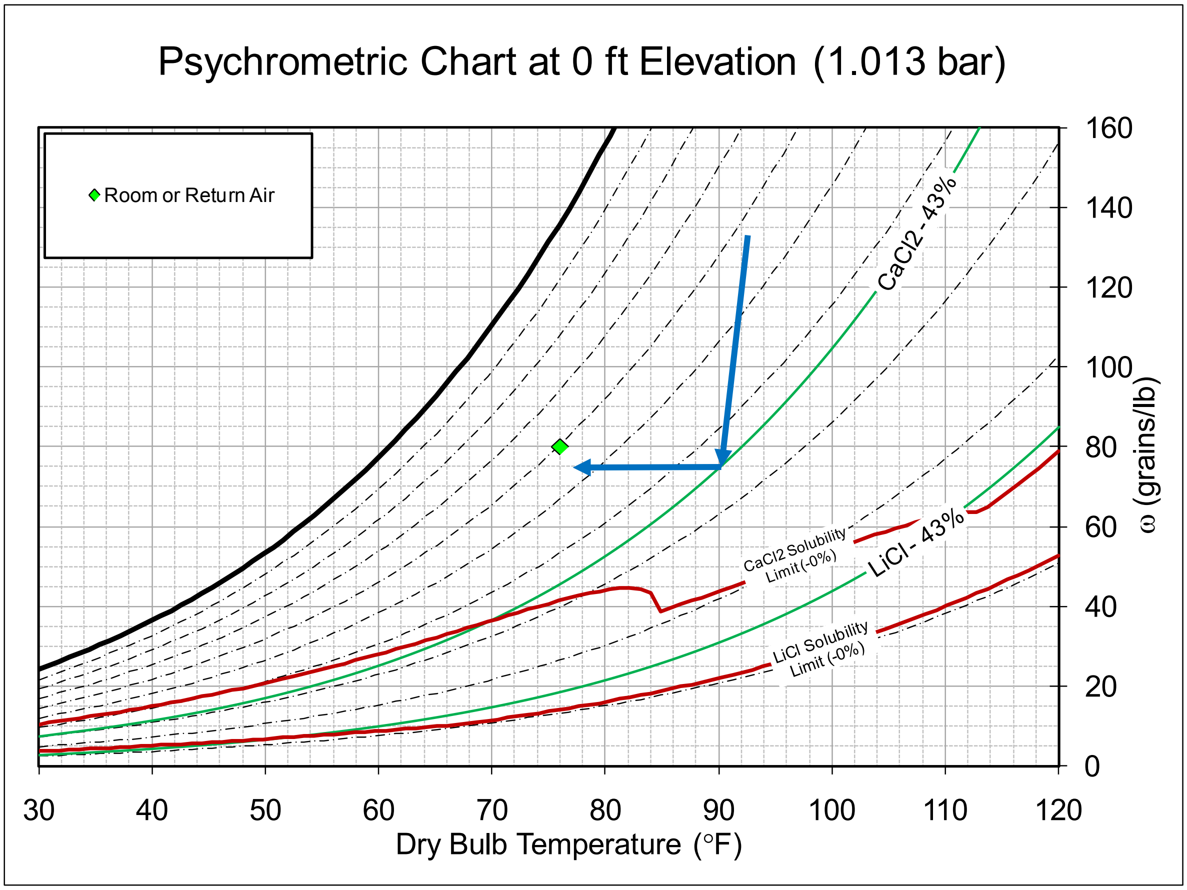


Figure 1. Psychrometric chart showing the dehumidification process using desiccants

Absorption of water vapor will eventually weaken the desiccant solution and reduce its dehumidifying potential; the desiccant must then be regenerated to drive off the absorbed water. Thermal regeneration is the reverse process of vapor absorption. In this process, the desiccant is heated to a temperature at which the equilibrium vapor pressure is above ambient vapor pressure. The water vapor desorbs from the desiccant and is carried away by an airstream (see Figure 2). Figure 2 shows how a scavenging airstream picks up heat and moisture from a regenerator. The green line represents the psychrometric condition of air in equilibrium with a $CaCl_2$ solution at the given temperature. Sensible heat is recovered by first preheating the ambient air using an air-to-air heat exchanger (AAHX). The air comes into contact with the desiccant in the heat and mass exchanger (HMX)—in this example at 190°F—and carries the desorbed water vapor away from the desiccant. Sensible heat is recovered by taking the hot humid air to preheat the incoming air through the AAHX. The change in enthalpy of the airstream as it passes through the regenerator represents the majority of the thermal input.

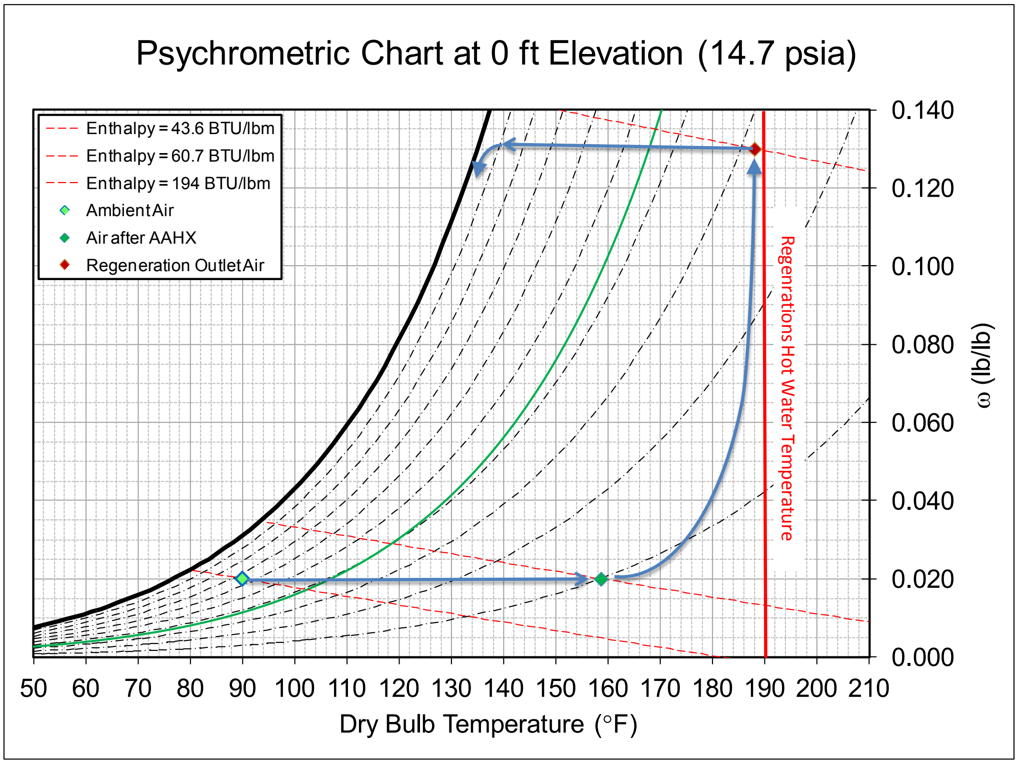


Figure 2. Desiccant reactivation using single-effect scavenging air regenerator

The process uses hot water or steam to achieve a latent COP between 0.8 and 0.94, depending on desiccant concentration. Latent COP is defined as:

$$COP_{Latent} = \frac{(Moisture\ Removal\ Rate) * (Heat\ of\ Vaporization)}{Heat\ Rate\ (Higher\ Heating\ Value)}$$

COP is maximized by maximizing the regeneration temperature and change in concentration while minimizing the desiccant concentration. Including the COP of the water heater (about 0.82), a typical combined latent COP for the LDAC systems is 0.82 × 0.85 = 0.7. If the heating source is derived from solar thermal technologies, the COP of the water heater would be the efficiency of the solar collectors (the benefit here being that there is no fuel cost penalty for the heat conversion efficiency).

The AILR technology innovations result in higher thermal efficiency when compared to other technologies on the market. NREL tests have shown that the Kathabar and other high-flow systems achieve a latent COP of about 0.4 to 0.55.

The LDAC technology developed by AILR uses novel HMXs to perform these two processes as shown in Figure 3, which illustrates the desiccant conditioner and scavenging air regenerator. The liquid desiccant is dispensed over the plates in the conditioner (absorber) where the inlet ambient air is dehumidified. This technology is called *low-flow, liquid-desiccant AC* because the desiccant flow is minimized in the HMXs of the conditioner and regenerator to the flow rate needed to absorb the necessary moisture from the airstream, which eliminates liquid desiccant carryover into the supply airstream. The HMXs must therefore have integral heating and cooling sources; 55°F – 85°F cooling tower water is supplied to the conditioner, and the regenerator uses hot water or hot steam at 160°F – 200°F. The cooling or heating water flows internal to the heat exchange plates while the desiccant flows on the external side of the HMX plates. The plates are flocked, which effectively spreads the desiccant and creates direct-contact surfaces between the air and desiccant as the air passes between the plates.

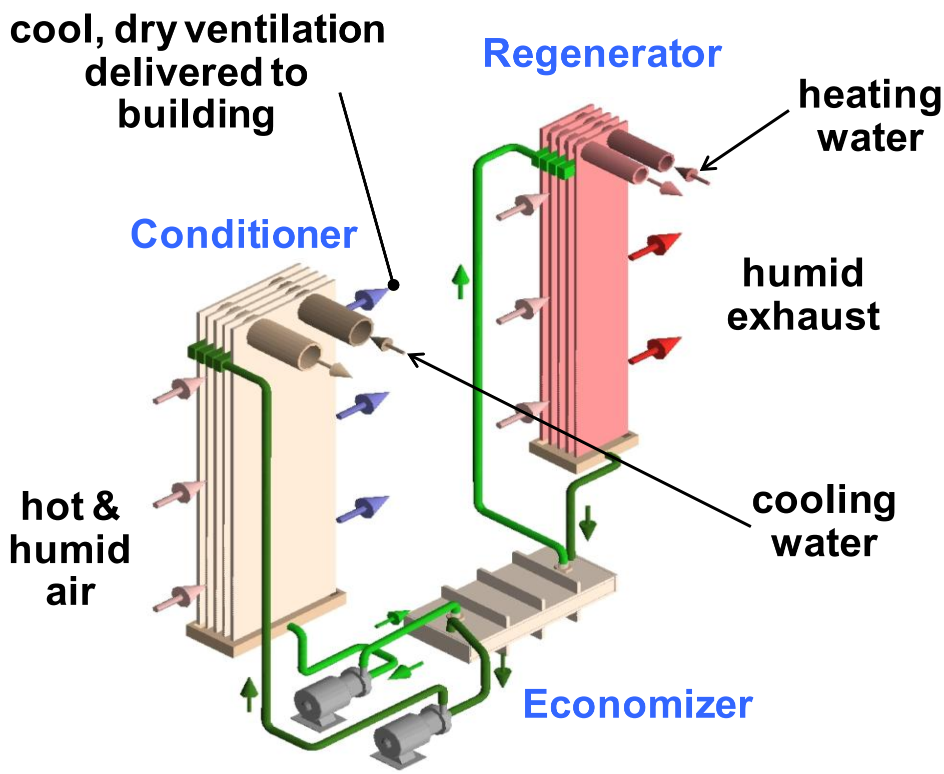


Figure 3. Major components and packaging of the AILR LDAC.

(Illustration by NREL)

2.2 FUTURE DEVELOPMENTS

NREL is working with AILR to develop a double-effect regenerator that expands on the scavenging air regenerator by first boiling the water out of the liquid desiccant solution (250°–280°F) and reusing the steam by sending it through the scavenging air regenerator. This two-stage regeneration system can achieve a latent COP of 1.05–1.2. A typical solar regenerator would consist of a hot water supply and a scavenging regenerator (which would result in a single-effect device that would have about a 60% solar conversion efficiency based on absorber area).

Table 5. Technology Options for Residential and Commercial Buildings*

Regenerator	Performance Rating
Solar Array	60% solar conversion
Single effect using natural gas	COP 0.7

*Based on NREL calculations and laboratory data, available upon request

For the low-flow LDAC, the regenerator and conditioner systems are shown connected in Figure 4, which illustrates the three thermal energy sources that can be used to regenerate the desiccant: solar thermal, traditional water heaters, and a double-effect technology. The water heater or boiler can be fueled by many sources, including natural gas, combined heat and power (CHP), or even biofuels. Also shown is the option for desiccant storage; storage allows an AC system to effectively bridge the time gap between thermal energy source availability and cooling load. A desiccant system that uses strong desiccant at 43% by weight and weakens the desiccant to 35% will be able to achieve storage density of 5 gallons (gal) of desiccant per latent ton-hour·(hr) of cooling provided. This is most easily accomplished using LiCl as the desiccant, as 35% desiccant still retains significant dehumidification potential. Desiccant storage is simple and theoretically lossless. Storage consists of a sealed, uninsulated tank with proper liquid connections. In comparison, ice storage is approximately 13–16 gal/ton-hr (theoretically 10 gal/ton-h, but in practice only 67% of the volume is frozen. (Ice Energy 2012) This storage can be useful to enable maximum thermal use from solar or on-site CHP.

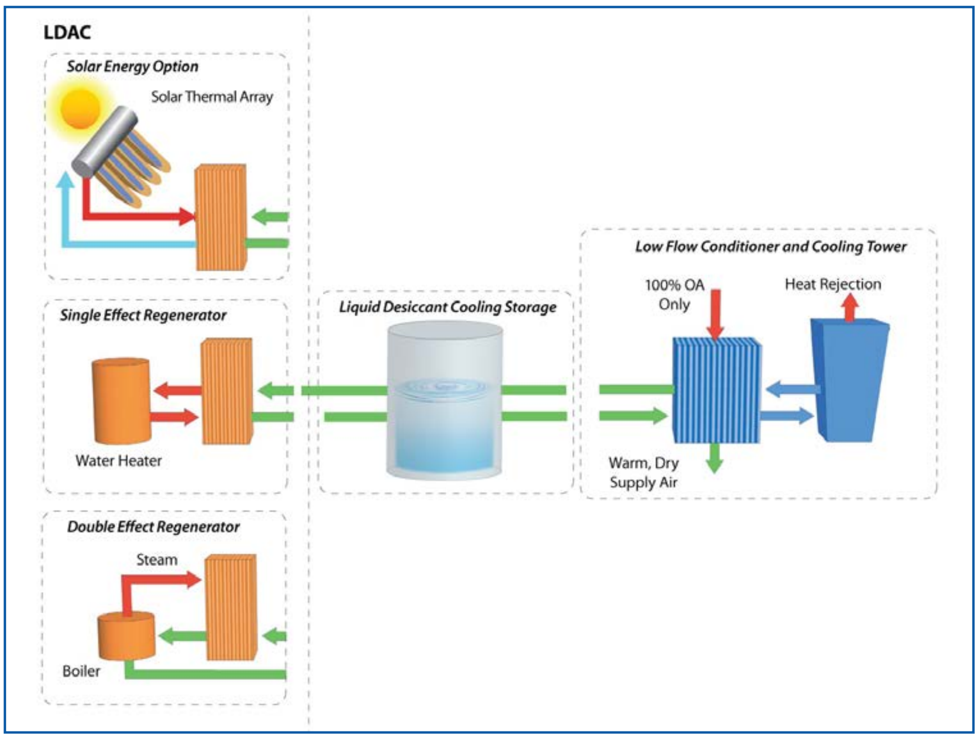


Figure 4. LDAC schematic.

(Illustration by NREL)

2.3 TECHNOLOGY DEVELOPMENT

Since its founding in 1988, the primary mission of AILR has been to develop and commercialize high-efficiency, end-use products for heating and cooling applications. For the past 14 years, AILR has focused on the parallel activities of developing plastic heat exchangers and applying these heat exchangers in HVAC products that use advanced liquid-desiccant technology.

From October 1990 to October 1991, AILR conducted research for the Gas Research Institute on a project entitled, "The Effect of Material Properties on the Performance of Liquid Desiccant Air Conditioners and Dehumidifiers." (AILR, undated) In addition to investigating alternative desiccants to LiCi, AILR studied novel configurations of the regenerator and the conditioner of an LDAC. An important conclusion from this work, which was reported in ASHRAE Paper No. AN-92-3-3, was that the desiccant flow rate in a packed-bed conditioner (which was the dominant technology at the time) is set by the requirement to limit the desiccant's temperature rise. (Lowenstein and Gabruk 1992) By embedding cooling within the conditioner heat and mass exchanger, the desiccant's temperature could be controlled independently of the amount of water absorbed. The desiccant flow rate could then be set by the need to limit the concentration change of the desiccant, a requirement that allows the desiccant flow to be reduced by over an order of magnitude compared to packed-bed conditioner designs.

In 1994, AILR received a patent that covered the low-flow, liquid-desiccant technology. (Lowenstein 1994) The patent was assigned to the Gas Research Institute, the organization that sponsored the research. Shortly after receiving the patent for low-flow, liquid-desiccant technology, AILR began to explore ways to practically capture the benefits of the technology. In September 1998, AILR delivered a 1,000-cfm, liquid-desiccant conditioner to NREL that used low-flow technology. (NREL 1997) The conditioner, which is shown in Figure 5, was composed of 75 polypropylene extruded plates that had been modified so that cooling water made six passes within each plate. A woven cotton fabric sleeve was slipped over each plate to provide a wicking surface for the desiccant.

Figure 5. The first implementation of a low-flow conditioner.

(Photo from AILR)

Following the successful testing of the liquid-desiccant conditioner at NREL (shown in Figure 5), AILR began to develop a manufacturable design for a low-flow, liquid-desiccant conditioner with additional support from NREL. (NREL 1998 and 1999) A low-flow, liquid-desiccant conditioner composed of extruded polyvinyl chloride (PVC) plates bonded to injection-molded manifold pieces was developed in this follow-on work. A 40-plate prototype of this conditioner was successfully tested at NREL in June 2004. This manufacturable design for the liquid-desiccant conditioner, shown in Figure 6, is used in the Tyndall solar LDAC.

Figure 6. The upper end of a manufacturable low-flow conditioner.

(Photo from AILR)

AILR's development of a low-flow conditioner was complemented by a parallel effort to develop a manufacturable, low-flow regenerator. Several approaches to a low-flow regenerator were explored under sponsorship by NREL. (NREL 2001) Prototypes were built using extruded chlorinated PVC (CPVC) plates and coated aluminum plates. The initial operation of both prototypes was good, but within several hundred hours of operation, both prototypes failed. A third prototype composed of extruded polyphenylsulfone (PPSU) plates, which is shown in Figure 7, proved successful operation for thousands of hours. A prototype of the PPSU regenerator was tested by NREL in February 2006. This PPSU regenerator is used in the Tyndall LDAC. PPSU is a plastic that can withstand temperatures as high as 250°F, but is substantially more expensive than other plastics. AILR and NREL continue to investigate regenerator designs with lower cost materials.

Figure 7. A PPSU regenerator (similar to the one installed in the Tyndall LDAC).

(Photo from AILR)

In 2003, AILR built the first prototype of a 6,000-cfm roof-top LDAC under a subcontract to Kathabar, Inc., as part of a larger effort of Oak Ridge National Laboratory. This prototype originally used a CPVC regenerator that failed after several hundred hours of operation.

In 2005, AILR built, installed and operated a second 6,000-cfm LDAC prototype, again under sponsorship of NREL. (NREL 2005) This prototype, shown in Figure 8, was installed on a machine shop in Wrightsville, Pennsylvania, where it successfully processed ventilation for 2 years.

Figure 8. The second prototype of a low-flow LDAC processing ventilation air for a machine shop in Wrightsville, Pennsylvania.

(Photo from AILR)

The 3,000-cfm solar LDAC built for Tyndall AFB was installed in spring 2010 and operated during the summers of 2010 and 2011. The Tyndall LDAC was the first implementation of a low-flow LDAC driven solely by solar thermal energy for regeneration. It was also the first AILR LDAC to operate in the field using a solution of $CaCl_2$ as the desiccant, which is more cost effective than LiCl as a means to store cooling, but it also does not provide the same dehumidification as LiCl, and is thus a compromise.

In May 2009, PAX Streamline (a venture-backed startup company) established a memorandum of understanding with AILR to transfer the low-flow technology to PAX for commercialization. Working together, PAX and AILR built a 6,000-cfm and 3,000-cfm LDAC and installed them on separate supermarkets in the Los Angeles area. Figure 9 shows the 6,000-cfm installation. Unfortunately, PAX Streamline failed in April 2010, despite the successful operation of the two supermarket LDACs.

Figure 9. Commercial LDAC using low-flow technology installed on a Los Angeles supermarket.

(Photo from AILR)

Following the failure of PAX, AILR continued to work with two former employees of PAX to build and install three more supermarket LDACs: two in California and one in Hawaii. These LDACs were installed between October 2010 and April 2011.

In July 2011, all intellectual property and know-how developed by AILR for building liquid-desiccant conditioners and regenerators that use low-flow technology were sold to the Munters Corporation. Munters is now in the early stage of commercializing the technology.

2.4 ADVANTAGES AND LIMITATIONS OF THE TECHNOLOGY

LDACs are a new breed of AC technology that decouples the latent load from the total load (sensible + latent) normally done by a refrigeration or chilled water system. De-coupling of these loads enables independent temperature and humidity control in a space. Also, lower humidities in a space can be achieved more efficiently by avoiding the energy intensive processes. Examples of these avoided processes and systems include:

- Overcooling and then reheating (which reverses the sensible cooling by the refrigeration system, thus lowering efficiency)
- Solid-desiccant wheels with natural gas or condenser heat regeneration. These systems generally increase energy use by the HVAC system due to high air-pressure losses and natural gas use.
- Ultra-low apparatus dew-point temperatures, which increase energy use by the refrigeration system.

LDACs largely switch much of the energy to condition air to thermal sources, such as natural gas, solar thermal, or waste heat. High-density storage can be employed to bridge thermal energy source profiles with cooling profiles, such as the case with solar thermal or even waste heat. Using waste heat is the most cost effective way to power an LDAC unit and should be considered first if a waste heat source is available. Natural gas or propane is economically utilized when dehumidification requirements are high. Supermarkets are a typical case where decreased store humidity drastically improves the efficiency of the food refrigeration systems. Thus store humidity levels are generally kept as low as possible. LDACs enable lower store humidity levels than other available humidity-control technologies and are just now being adopted at major supermarket chains as a result. For example, Whole Foods has recently included the LDAC technology in its HVAC specification in humid climates. Solar energy for LDACs can become economical if the relative cost of solar thermal energy is competitive with natural gas or propane. This is often the case on islands such as Hawaii, Guam, and many other tropical island nations. Solar thermal systems should always be used to offset fossil fuel use but not as the primary source of energy. Designs that attempt to get a solar fraction of 1.0 inherently will have solar fields and desiccant storage tanks that are much larger and more expensive than practical.

LDACs primarily use cooling towers for their cooling sink. If cooling towers are compared to air-cooled AC systems, site water use will increase. However, many chiller systems use cooling towers, and LDAC technology would use about the same amount of water for cooling as these systems do. The electric power grid also uses water to cool thermal power plants. The avoided use of electric power can result in substantial regional water savings. Case-by-case analysis is required to calculate these savings. However, typical thermal power generation station produce about 1.0 to 2.0 kilowatt-hours (kWh) of electricity per gallon of water evaporated (0.5 - 1 gal/kWh).

Water use is dictated by how much energy is removed per pound (lb) of water evaporated. Water's heat of vaporization is about 1,060 Btu/lb, which is equivalent to 1.37 gal/ton-hr of cooling load. However, because evaporative cooling is an open-cycle process where mineral content of domestic water must be removed, the water use will be higher by the cycles of concentration (CoC) required to prevent mineral buildup in an evaporative system. CoC is defined as the ratio of mineral concentration in the blow-down water divided by the initial concentration. CoC is dependent on water quality, but typically range from 2-7 where a CoC of 2 is typically associated with facilities that have extreme water hardness. A typical water-draw rate for a typical cooling tower will be 1.57 to 2.74 gal/ton-hr. In the case of the LDAC technology, a cooling tower must only remove the cooling load. In the case of a water-cooled DX system, the cooling tower must remove the cooling load plus the compressor load. For a DX cycle with a COP of 4, a cooling tower would thus draw 25% more water or typically 1.96 to 3.42 gal/ton-hr.

The preceding analysis does not include the complicated weather effects on a cooling tower, but is approximate for most conditions where the cooling tower's airstream becomes fully saturated and leaves at the same temperature as the inlet air. However, the comparison with DX cooling remains accurate in relative amounts. LDAC technology will, in general, use about 25% less water than a water-cooled DX system. The net regional water impact by using an LDAC system will typically be small or even positive in some cases.

LDACs are now being employed to treat dedicated outdoor airstreams to control humidity in a space. The highest benefit thus will be for humid climates with large yearly humidity loads and applications where reheat energy is high.

High-value applications include buildings with large outdoor air loads that have the highest levels of reheat or benefit from decreased humidity in the space such as the following:

- Hospitals (avoiding massive amounts of reheat)
- Supermarket humidity control
- School buildings in humid regions
- Buildings with waste heat available
- Indoor pools.

LDACs are an emerging technology and have not seen the level of refinement that economy of scale has bestowed upon vapor-compression technology. The technology is still in a pre-commercial state; the systems are more complex than traditional vapor-compression systems and require custom engineering in most new applications. This is a major hurdle that now faces this technology as research funding will inherently drop and market pull must pick up. Thus LDACs will be first introduced in the highest-value applications, where market pull for the benefits is large enough. In its current state the LDAC technology cannot compete in facilities that do not over-cool/re-heat supply air and are also in locations with lower electricity rates.

3.0 PERFORMANCE OBJECTIVES

Table 6 describes the performance objectives, metrics, and data requirements to determine the performance objective results; and the criteria for achieving the objectives.

Table 6. Performance Objectives

Performance Objective	Metric	Data Requirements	Success Criteria
Quantitative Performance Objectives			
Improve humidity control and comfort (energy efficiency)	• Hours outside psychrometric comfort zone • Chiller power • Reheat run-time	• Indoor temp./humidity • Chiller power • Reheat coils on	• <1% of hours outside ASHRAE summer comfort zone • Reduce chiller/reheat run-time
Provide high-efficiency dehumidification (energy efficiency)	• EER • COP	• Supply-air temp./humidity • Supply-air flow rate • Ambient temp./humidity • Power consumption • Heat consumption	• EER >40 (Btu/hr)/W • >0.7 Thermal COP
Sustain high-dehumidification performance (energy efficiency and maintenance)	• Conditioner heat-exchange effectiveness • Desiccant charge • Supply-air pressure drop • Conditioner cooling-water pressure drop • Projected service life	• Supply air temp/hum • Ambient temp/hum • Desiccant chemistry and concentration • Conditioner core-air and water-pressure drop	• <5% degradation of HX eff. over 3 years • <Once-per-year desiccant/buffer adjustment • Negligible increase in air/water pressure drop • Above criteria should support >10-yr service life projection

Qualitative Performance Objectives			
Maintainability (ease of use)	Ability of an HVAC technician to operate and maintain the technology	Standard form feedback from the HVAC technician on usability of the technology and time required to maintain	A single facility technician able to effectively operate and maintain equipment with minimal training

3.1 IMPROVE HUMIDITY CONTROL AND COMFORT

Conventional AC has fundamental limitations regarding humidity control. Mechanical AC provides coincidental drying when it lowers air below its dew-point temperature, condensing water vapor on its cooling coils. In order to provide separate control over temperature and humidity, it must overcool the air and then reheat it so as not to overcool the space. This can be excessively energy intensive. Mechanical AC runs into additional problems with dehumidification when humidity loads are large compared to sensible loads, such as in the spring and fall, and under any conditions where short-cycling occurs, which leads to re-evaporation of water from the cooling coils.

This objective will demonstrate that desiccant drying effectively manages humidity independently of sensible loads throughout the cooling season. A comfort zone of combined temperature and humidity ranges has been established by ASHRAE; a technology's ability to provide comfort is typically represented by the number of hours per year indoor conditions leave this established zone. NREL monitored chiller power and reheat coil run-times both with and without desiccant system operation to show effects on comfort and overcool/reheat. NREL used its testing experience to ensure that sensors were positioned to read representative supply-air temperatures that were not skewed by spatial nonuniformities or radiative heat exchange.

3.2 PROVIDE HIGH-EFFICIENCY DEHUMIDIFICATION

It is critical to establish the efficiency advantages of desiccant dehumidification over conventional mechanical AC. The cooling effect is determined by the change in enthalpy of the supply air relative to ambient, multiplied by the total mass flow of supply air. The range of psychrometric conditions that is monitored allows high-quality temperature and RH sensors to adequately make these enthalpy measurements. Mass flows are measured with pitot tube-in-duct measurements, inferred from fan power draw and checked against pressure drop measurements calibrated at NREL's test facility. Energy balance for the cooling effect can be checked against the measured heat gain of the cooling water flowing through the conditioner.

Power measurements to monitor fan and pump energy are straightforward. Thermal energy consumption is also conveniently measured by the drop in temperature of the solar collector fluid through the regenerator multiplied by its mass flow rate and specific heat. All of these parameters are straightforward to determine in real-time.

3.3 SUSTAIN HIGH-DEHUMIDIFICATION PERFORMANCE

A common concern regarding liquid-desiccant devices is the carryover of desiccant droplets into the supply (or scavenging) airflows, exhibited as decreasing desiccant charge and corrosion of metal ductwork where these droplets settle. One strategy for avoiding this loss of desiccant is the deployment of high-pressure drop demisters in the supply duct. However, this approach is not desirable because it imparts a significant energy penalty in fan power and requires frequent maintenance to remain effective. Alternatively, the low-flow liquid desiccant uses minimal mist eliminators, relying on its low-flow feature to prevent carryover and preserve desiccant charge. This low-flow feature is enabled by the flocked surfaces of its conditioner component. These desiccant-wetted wicks must keep themselves relatively free of dirt buildup to maintain their zero-carryover feature and the effectiveness of the HMX. The conditioner's heat exchange effectiveness and pressure drop were monitored to ensure that ambient dirt was being successfully managed to preserve performance over a projected service life of 10-15 years. In general, because the conditioner is water cooled, mineral fouling of its internal water passages is possible, albeit improbable due to its all-plastic construction.

3.4 MAINTAINABILITY

There are many ways for a building technology to fail if its operation requires more than the current HVAC standard maintenance requirements, which is minimal. The success criteria for this performance objective were determined if the demonstration unit settled into autonomous operation after the first season following Tyndall's standard HVAC maintenance schedule.

4.0 FACILITY/SITE DESCRIPTION

The selected laboratory at Tyndall AFB is located in Panama City, Florida, on the Gulf Coast. Its high temperatures are typically in the 80-89°F range in the summer, rarely above 90°F, with ambient humidity in excess of 0.02 lbs of water per lb of dry air through portions of the cooling season. Its design wet-bulb temperatures are very high, ranging from 70°F to over 80°F, with humidity extremes up to 90%. Figure 10 illustrates the wide range of temperatures and humidity that Tyndall experiences throughout the year.

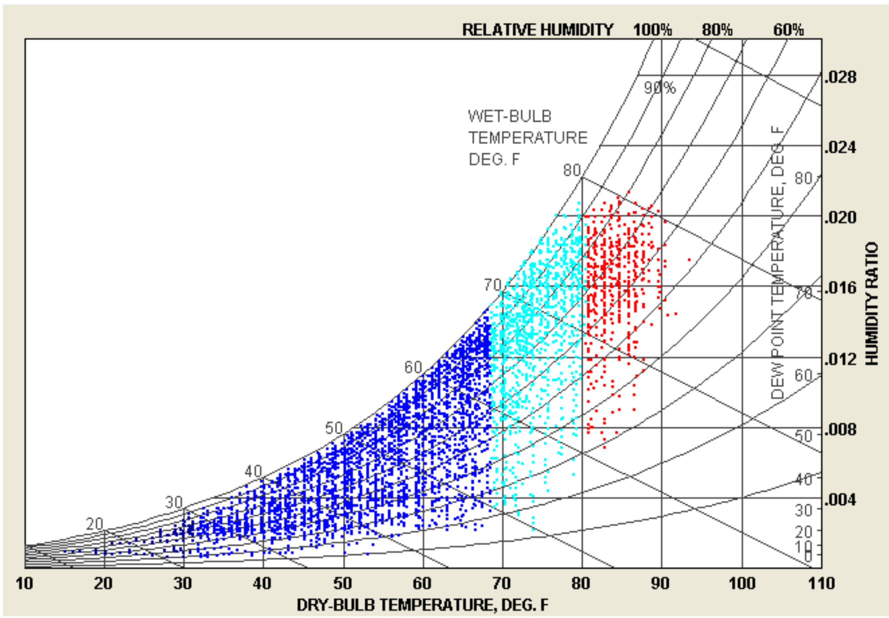


Figure 10. Psychometric plot Tyndall AFB

In addition to the hot and humid summer days, the cool temperature days from the late fall to early spring allow for a robust system performance evaluation through the observation of operation in non-ideal weather conditions and taking proper measures to avoid damage from freezing. Altogether, the site provides the necessary spectrum of ambient conditions to characterize the LDAC performance sufficiently for predicting performance across most, if not all, conditions in the United States, U.S. territories, and countries with active U.S. military operations.

4.1 FACILITY/SITE LOCATION AND OPERATIONS

The Air Force Research Laboratory (AFRL) building at Tyndall AFB is a mix of laboratory and office space. Three main air-handling units (AHUs), serve the laboratory and office space.
A satellite image of Tyndall AFB and the LDAC system is provided in Figure 11.

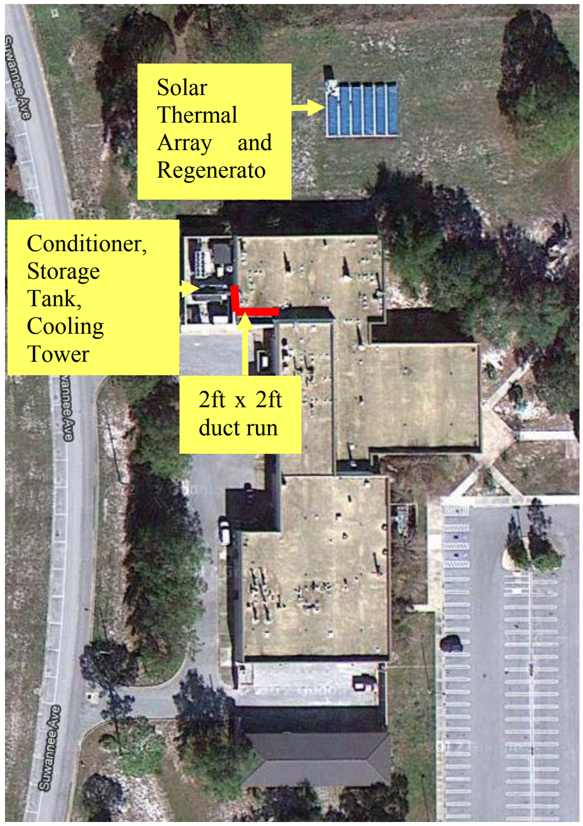


Figure 11. AFRL site.

(© 2012 Google)

Typical configuration may include a single-packaged unit or a split-system arrangement, where the regenerator cabinet is physically split from the conditioner cabinet. The requirement to place the solar system in the nearby field rather than roof mounting meant that the regenerator is best placed in the field adjacent to the solar array for minimizing heat loss. The conditioner, cooling tower, and desiccant storage tank are located in an enclosed HVAC area, so the conditioned air can be supplied into the building via ductwork.

Figure 12 shows the Tyndall AFB building layout, with the space apportioned by office, laboratory, and mechanical rooms. The red box highlights the predominantly laboratory space for which the LDAC system provides ventilation air. Reheat coils in terminal units in each zone activate if the air has been overcooled, and using measured data from AHU #3, the reduction of reheat can be determined.

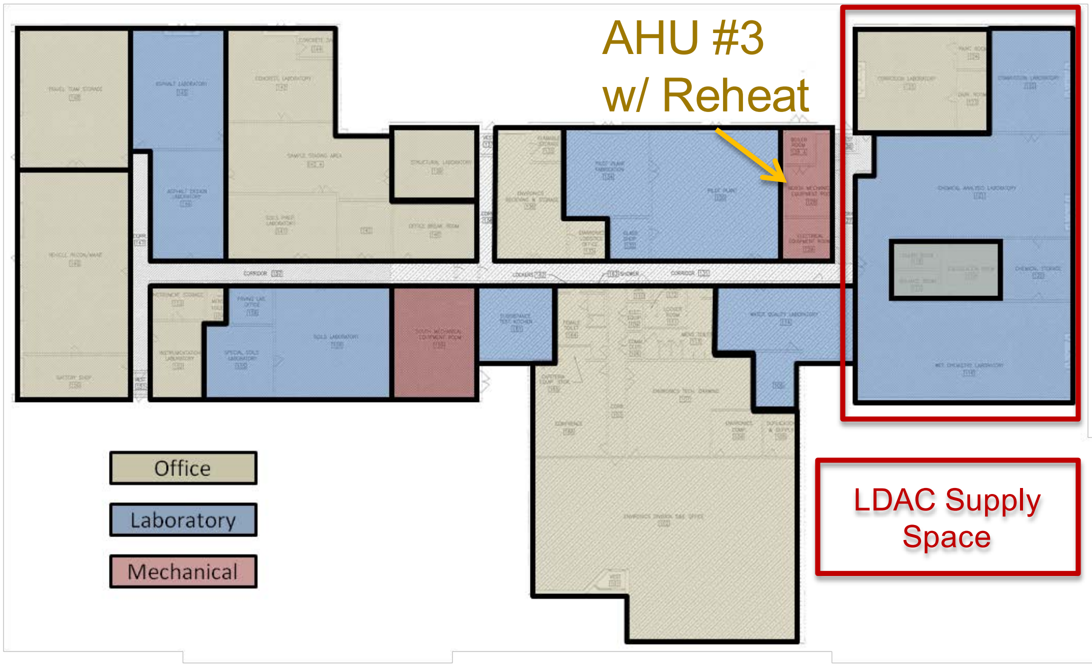


Figure 12. Tyndall AFB building layout.

(Illustration by Jesse Dean, NREL)

The laboratory wing served by the low-flow, liquid-desiccant unit underwent a chiller upgrade in 2008 because cooling loads were going unmet. The building was also recommissioned to balance the outdoor air to ensure positive pressure within the building to eliminate condensation due to infiltration. As a result, reheat coils were not being actuated because indoor temperature set points were not being reached. This implies that indoor humidity was not being controlled and that conditions were likely uncomfortably humid. The chiller upgrade provides sufficient capacity to properly dehumidify (overcool) the space, and therefore, required reheat coil operation. The upgrade also included condenser heat recovery to offset reheat energy use.

A DX, air-cooled chiller (Table 7) supplies chilled water to all three AHUs. The LDAC system conditions ventilation air that serves AHU #3 (Table 8).

Table 7. Air-Cooled Chiller Schedule

Air cooled chiller	
Evaporator Performance	
Total Capacity (tons)	180
Entering Water (°F)	56
Leaving Water (°F)	44
P.D. (Ft)	15
GPM	370
Compressor Performance	
Type	Rotary Screw
Refrigerant	22
Number of Compressors	3
Electrical Performance	
Compressor and Fan KW	205
EER	9.3

Table 8. AHU #3 schedule

AHU #3	
Fan Performance	
Fan Type	Forward Curve
Supply Air (cfm)	11,710
Outside Air (cfm)	7,000
Static Pressure (in. H2O)	2.5
Motor Size (HP)	20
Configuration	Blow-Thru
Volt/Phase/Cycle	460/3/60
Cooling Coil Performance	
Max Face Velocity (fpm)	520
Max Air P.D. (in. H2O)	1
Max Water P.D. (Ft. H2O)	15
Entering DB/WB (°F)	86.5/71.1
Leaving DB/WB	54.6/53.5
Entering Water	44
Leaving Water	56
GPM	127
Total Heat (BTUH)	758,400

5.0 TEST DESIGN

5.1 CONCEPTUAL TEST DESIGN

NREL installed instrumentation and a data acquisition system for one SOA3000 dehumidifier, powered by a 1,300-ft^2, evacuated-tube solar thermal array. The solar array and regenerator components are oversized relative to the conditioner's average dehumidification output in order to generate and store excess desiccant during the day. An 800-gal uninsulated storage tank fully utilizes the solar arrays excess heat output and allows for a few hours of average cooling operation without solar input. The unit is designed to operate continuously at maximum airflow in order to serve fume-hood makeup air needs. The LDAC technology was characterized in NREL's Advanced HVAC Systems Laboratory in 2004 and 2006. The laboratory test results are invaluable in interpreting the field results, particularly with regard to critical airflow rates, which are notoriously difficult measurements to make in the field. The unit was monitored for two cooling seasons, and its annual and peak energy use was compared to conventional AC.

5.2 BASELINE CHARACTERIZATION

The installation had the potential to generate compelling side-by-side test results in that the recent chiller upgrade should allow operation with or without desiccant unit operation. Circumstances that complicate comparison include the facts that: 1) the chiller serves the entire building; and 2) the disparity in capacities between the chiller and the desiccant system is approximately 10:1. Chiller power and calls for reheat were measured in one wing of the facility. Because mechanical AC is a well-understood technology, baselines for individual sites were not critical to project energy savings relative to conventional equipment at various efficiency levels. Once the efficiency of the desiccant system was established, comparisons of energy use relative to mechanical AC were straightforward over the full range of building applications and climates.

5.2.1 Laboratory Testing

NREL was directly involved in the development of the LDAC technology and funded the development of the three core HMXs now used: the conditioner, regenerator, and interchange heat exchanger. These HMXs were developed by AILR and NREL determined the performance of the units through testing at the Advanced HVAC Systems Test Laboratory. The final conditioner and regenerator HMX designs were tested in 2004 and 2006 respectively. The data from these tests proved invaluable in the demonstration as baseline performance was determined and verified against laboratory data. Airflow, pressure drop, and exchange effectiveness from the laboratory proved invaluable in the calibration of airflow and pressure drop measurements in the field, which improved the uncertainty of the field data. There are no standard methods for testing LDAC systems. However, NREL is a world leader in the development of desiccant technologies and draws upon sound practices in order to determine the performance of these HMXs. To determine effectiveness, each HMX was instrumented at the NREL laboratory with the following parameters and accuracy:

- Air and liquid temperature measurements have an accuracy of ± 0.36°F with <0.18°F deviation.
- Airflow measurements have an accuracy of ± 2%.
- Humidity is calculated with dew-point hydrometers and has an accuracy of 0.27°F.

- Differential pressure measurements have an accuracy of ± 0.025 inches water column (W.C.)
- Barometric pressure has an accuracy of 0.15%.
- Water flow-rate measurements have an accuracy of 0.5%.
- Liquid desiccant flow-rate measurements have an accuracy of 1.0%.
- Desiccant concentration by weight is measured to within 0.0025 lbs salt per lb of solution.

A diagram of the HVAC laboratory in the Thermal Test Facility at NREL is provided in Figure 13.

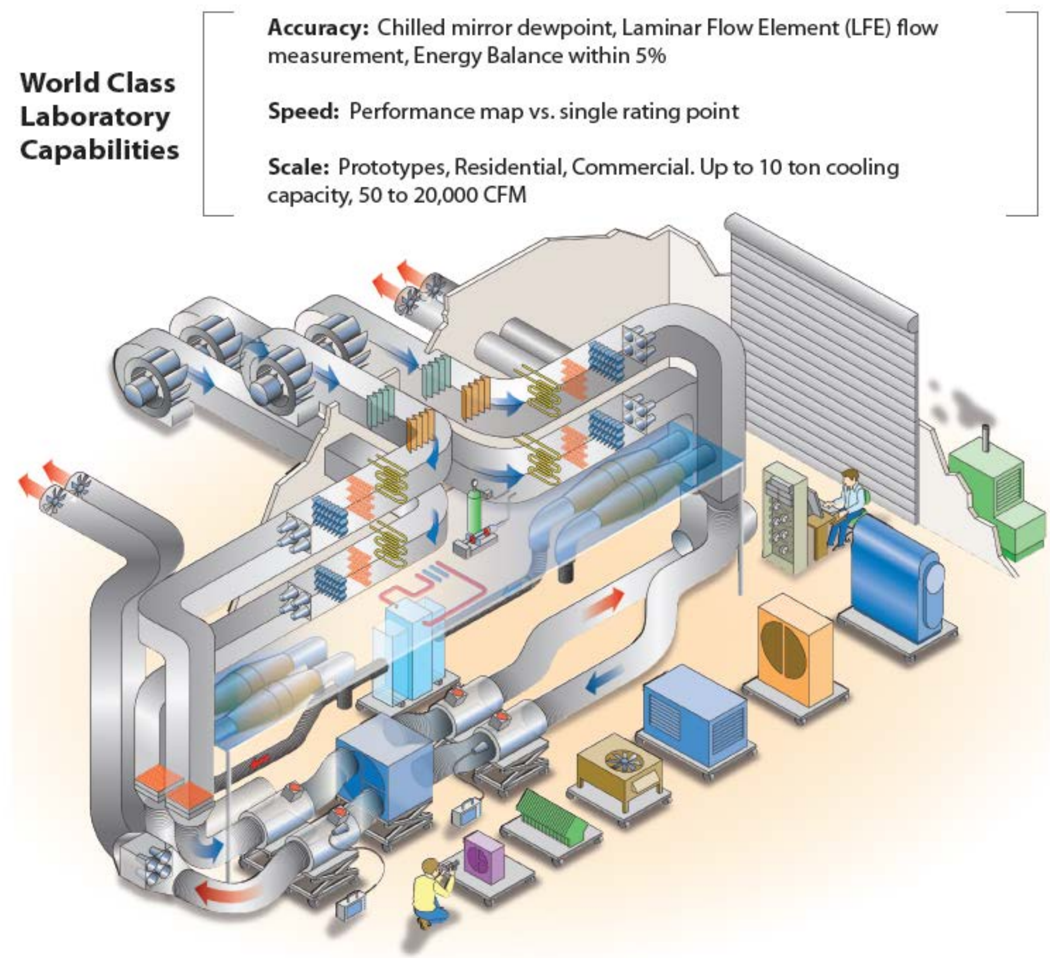


Figure 13. NREL Thermal Test Facility.

(Illustration by NREL)

The laboratory testing resulted in the development of a regression of cooling effectiveness and capacity versus airflow rate; conditioner and regenerator water flow rate and temperature; and desiccant concentration. These data were also shared with AILR to calibrate its internal modeling of the LDAC system for design purposes. It is, however, impractical to measure every combination of independent variables to determine a performance map of the LDAC system.

Thus the calibration/validation of the AILR model using NREL data improves the accuracy of the model, making it a suitable tool to determine LDAC performance using building energy simulators such as EnergyPlus.

5.3 DESIGN AND LAYOUT OF TECHNOLOGY COMPONENTS

Figure 14 illustrates the design and layout of the LDAC system. Demonstration equipment was placed in two locations at the test site. The solar array and regenerator components were collocated in the open field to the north of the building. The desiccant storage, conditioner component, and cooling tower were placed within the walled HVAC equipment area on the west end of the subject wing with the chiller. Piping connecting the storage tank and conditioner supply strong desiccant for the dehumidification process, and a 100-ft-long duct run was installed across the roof to connect the conditioner to the fresh air intake of the building.

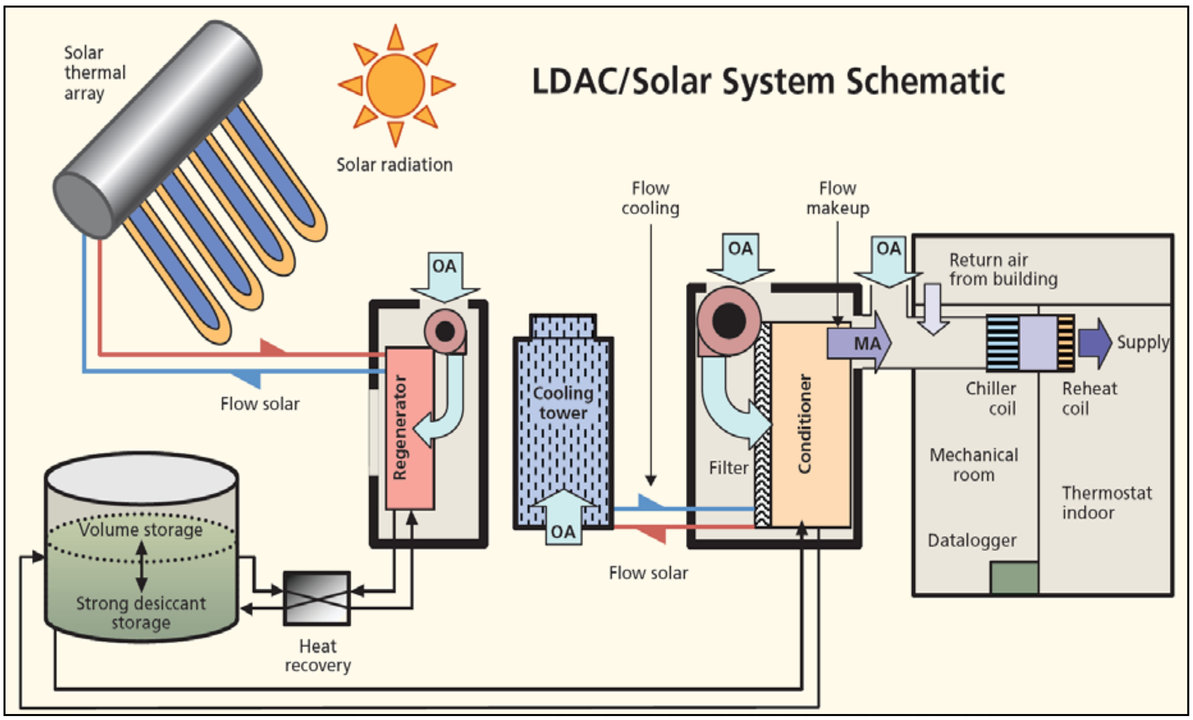
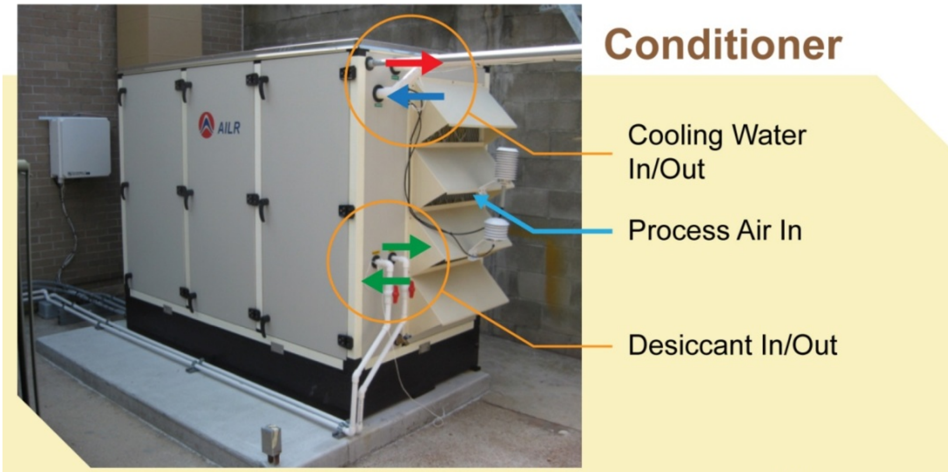


Figure 14. LDAC system and supply air layout.

(Illustration by NREL)

5.3.1 Conditioner

The conditioner component (Figure 15) is a proprietary plastic heat-exchanger design that brings liquid desiccant into direct contact with ventilation air to dehumidify the outside air, while simultaneously rejecting heat to the cooling tower water. The system was designed to operate with $CaCl_2$ as the liquid desiccant (up to 43% salt and 57% water), which is suitable for solar applications. The conditioner is designed to reduce the RH of the conditioned air to 35% - 40% RH. A 3.5-horsepower (hp) fan with a variable-frequency drive (VFD) is designed to provide 3,000 cfm of treated air to the building. Two fractional (1/4) hp pumps circulate ~ 3 gal per minute (gpm) of desiccant between the storage tank and the conditioner. A 1-hp pump supplies 37 gpm of cooling tower water to the conditioner.



	Value	Units
Heat Exchanger Material	Plastic Plates	
Inlet Desiccant Flowrate	2.89	gpm
Inlet Desiccant Concentration	0.43	lbm salt/lbm soln
Inlet Desiccant Temp	105	F
Inlet Cool Water Flowrate	36.6	gpm
Inlet Cool Water Temp	87.7	F
Inlet Air Flowrate	3000	SCFM
Inlet Air Temperature	87	F
Inlet Air RH (Ambient)	81	%
Outlet Desiccant Flowrate	3.16	gpm
Outlet Desiccant Temp	92.9	F
Outlet Desiccant Concentration	0.4	lbm salt/lbm soln
Outlet Cold Water Flowrate	18	gpm
Outlet Cold Water Temp	96.6	F
Exhaust Air Flowrate	3000	SCFM
Exhaust Air Temp	91.6	F
Exhaust Air RH	37.1	%
Process Air Pressure Drop	0.4	in w.c.
Water removal rate	18.4	gph

Figure 15. Image and design of conditioner unit.

(Photo by NREL)

The cooling tower (Figure 16) is an EVAPCO model ICT 3-63 packaged unit with a 1-hp fan with a VFD.

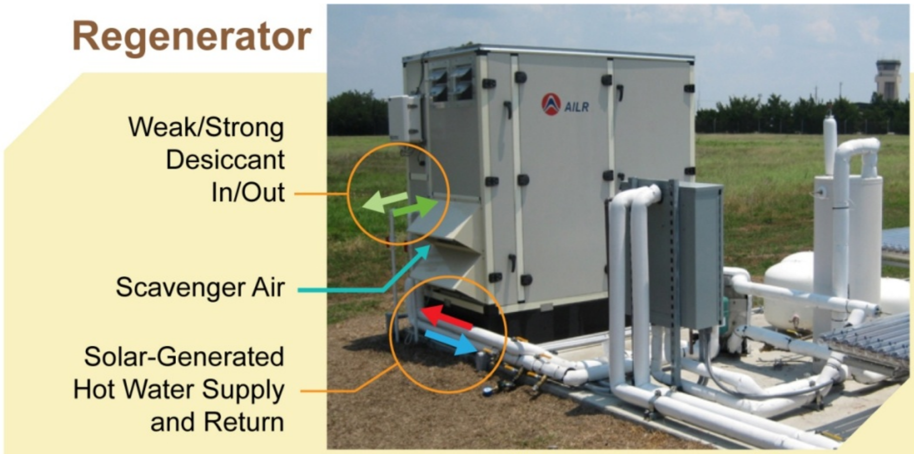
	Value	Units
Model	EVAPCO model ICT 3-63	
Type	Induced Draft, Counter-flow	
Cooling Capacity	10.4	tons
Fan	Propeller on VSD	
Footprint	9.1	ft^2
Height	6.2	ft
Inlet Water Flowrate	36.6	gpm
Inlet Water Temperature	96.6	F
Outlet Water Flowrate	36.6	gpm
Outlet Water Temperature	87.7	F
Heat Rejected	10.4	tons
Makeup water	4	gph
Max Air Flowrate	4900	CFM

Figure 16. Image and design of cooling tower.

(Photo by NREL)

5.3.2 Regenerator

The water vapor absorbed by the liquid desiccant is carried away in the humid exhaust air exiting the regenerator (Figure 17). The regenerator installed at Tyndall AFB is designed to utilize 165-190°F water to drive the dehumidification process. The water is heated by a plate and frame heat exchanger with the solar thermal heating system providing the heat. A 0.7-hp fan provides the 900 cfm of scavenging air needed to reject the moisture to the ambient air. A fractional (1/4) hp pump circulates ~4 gpm of desiccant between the storage tank and the regenerator. A 1-hp pump supplies 18 gpm of solar-heated water to the regenerator.



	Value	Units
Heat Exchanger Type	Plastic Plates	
Inlet Desiccant Flowrate	3.3	gpm
Inlet Desiccant Concentration	0.4	lbm salt/lbm soln
Inlet Desiccant Temp	90	F
Inlet Hot Water Flowrate	18	gpm
Inlet Hot Water Temp	180	F
Inlet Air Flowrate	900	SCFM
Inlet Air Temperature	87	F
Inlet Air RH (Ambient)	81	%
Outlet Desiccant Flowrate	3	gpm
Outlet Desiccant Temp	103	F
Outlet Desiccant Concentration	0.43	lbm salt/lbm soln
Outlet Hot Water Flowrate	18	gpm
Outlet Hot Water Temp	157	F
Exhaust Air Flowrate	935	SCFM
Exhaust Air Temp	121	F
Exhaust Air RH	89	%
Air-Air HX Effectiveness	0.58	
Interchange HX Effect.	0.8	
Water removal rate	19.4	gph

Figure 17. Image and design of regenerator unit.

(Photo by NREL)

5.3.3 Solar Collectors

The solar array (Figure 18) is a Viessmann 200-T model evacuated-tube collector using 29 30-tube bundles covering 1,350 ft^2 of area. The 200-T model evacuated-tube collector is a tube-in-tube flow through design, which has different performance characteristics than the standard heat pipe design. The solar collector transfers the collected solar energy to a heat transfer fluid that is then transferred through a plate and frame heat exchanger to the regenerator. Because the solar collector stagnates the entire liquid volume internal to the collector, this solar collector requires unusually high expansion tank volumes to deal with stagnation pressures as well as a heat transfer fluid that can resist 450°F stagnation temperatures. For this reason, water was used as the heat transfer fluid. Under normal conditions, propylene glycol could be used with these collectors, provided that stagnation is considered an uncommon occurrence (occurring less than about three times per year). However, the LDAC system would frequently be shut down for maintenance or lack of cooling load throughout its normal operation. The use of water then necessitates draining the unit prior to freezing conditions every year and startup must occur after the last expected freeze date. For this reason, this particular collector design was a poor choice to be paired with an LDAC system in a freezing climate. Furthermore, heat-pipe, evacuated-tube designs require far less expansion tank volume than do the 200-T. A solar system is best used when there is a year-round thermal load to maximize this more expensive thermal energy source. Currently, islands with high fuel costs are the target application for pairing LDAC with solar thermal energy.

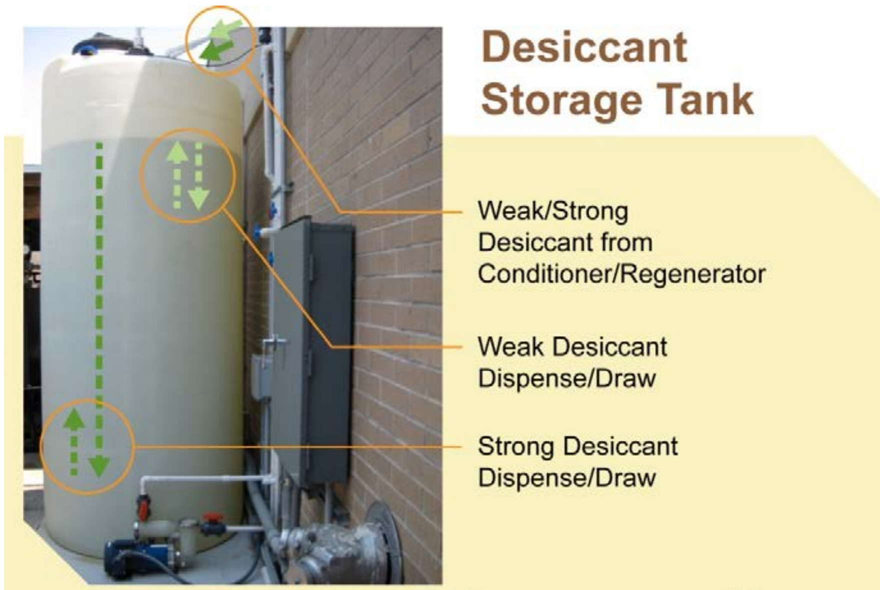
	Value	Units
Model	Viesmann 200-T	
Type	Direct-Flow Evacuated Tube	
Tube Bundles	29	
Total # Tubes	870	
Gross Area	1348.5	ft^2
Absorber Area	942.5	ft^2
Water flow	18	gpm
Inlet temperature	164.9	F
Outlet temperature	187.8	F
Heating Capacity	16.4	tons
Fluid Capacity	47.56	USG
Max Pressure	87	psi
Max Stagnation Temp	563	F
Tilt Angle	0	deg
Azimuthal Angle	90	deg
Expansion Tank?	Yes	
Pressure release?	Yes	

Figure 18. Image and design of solar collector array.

(Photo by NREL)

5.3.4 Desiccant Storage Tank

Desiccant storage is provided in an 800-gal, uninsulated, plastic tank (Figure 19). A proprietary plastic interchange heat exchanger is plumbed between the tank and the regenerator. It precools hot, strong desiccant coming from the regenerator and preheats weak desiccant going to the regenerator. This process reduces the heat input required to the regenerator and supports concentration stratification in the storage tank. This enhances performance by supplying the conditioner with the strongest desiccant possible at all times.



	Value	Units
Type	Plastic	
Tank Volume	800	gal
Desiccant Concentration Top	0.4	lbm salt/lbm soln
Desiccant Concentration Bottom	0.43	lbm salt/lbm soln
Lower Limit Volume (strong des.)	580	gal
Upper Limit Volume (weak des.)	680	gal
Insulation?	No	
Corrosion Resistant	Yes	

Figure 19. Image and design of desiccant storage tank.

(Photo by NREL)

Regenerator and conditioner operation are essentially independent. Regeneration components are activated when sensors indicate the solar array is capable of generating minimum productive temperatures (~160°F). The control sequence for the conditioner was modified throughout the 2011 cooling season in an attempt to evaluate the technology in a variety of operating modes.

The system components have pumps and fans to drive liquid and air through the system.
Table 9 lists the size and power draw of the pumps and fans that are used on the LDAC system.

Table 9. LDAC System Pump and Fan Motor Schedule

Function	Type	Motor Nameplate Data		
		Power (hp)	Power (W)	VFD
Conditioner				
Supply Pump	Centrifugal	0.25	186.5	No
Return Pump	Vertical	0.25	186.5	No
Process Fan	Motorized Impeller	3.45	2,570	Yes
Cooling Tower				
Pump	Jet Pump	1	746	Yes
Fan	Axial Propeller	1	746	Yes
Regenerator				
Supply Pump	Centrifugal	0.25	186.5	No
Return Pump	Centrifugal	0.25	186.5	No
Loop Pump	Vertical	0.24	179	No
Fan	Motorized Impeller	0.71	530	Yes
Solar Array				
Loop Pump	Circulating	0.24	180	No
Loop Pump	Circulating	0.33	245	No

5.4 OPERATIONAL TESTING

Field testing was conducted in two phases: startup and monitoring. During startup, NREL and Mountain Energy Partnership (MEP) installed sensors and confirmed HVAC/data system operation on-site. Startup commenced as the equipment installation proceeded in winter 2009 and concluded 2 weeks later. The performance of the system was monitored over the 2010 and 2011 cooling season, and the unit was shut down and winterized each winter.

5.5 SAMPLING PROTOCOL

An initial site visit to Tyndall AFB was made in December 2009 to install the monitoring system for the LDAC system. All of the sensors and data loggers were installed at that time; however, the solar collector and LDAC were not functioning properly due to the improperly designed stagnation strategy with the Viessmann solar collectors. Modifications to the original solar collector design were required to accommodate normal stagnation conditions of the system. The monitoring system could not be fully commissioned until normal operation of the LDAC was achieved in July 2010. A site visit was made in July 2010 to complete the monitoring system installation.

Normal operation of the LDAC and a complete monitoring system facilitates an initial estimate of uncertainty in the measured LDAC heat flows. The sensible and latent capacities of the LDAC are of primary importance in evaluating system performance. The monitoring system directly measured the latent and sensible capacities on the airside of the conditioner by measuring the air velocity with a pitot tube, the change in dry-bulb temperature using two resistance-temperature device sensors, and the change in humidity ratio using two capacitive RH sensors. Flow rates through the conditioner were inferred using the pressure drop through the conditioner and the fan-speed indication, measured by the conditioner's programmable logic controller (PLC) and transferred to the data logger via Modbus communication. The heat removed by the cooling tower was intended to be a direct measurement of the total capacity of the conditioner. Water-side measurement using a turbine flow meter and two thermistors were used because it was expected to have lower uncertainty than the air-side measurement. A third measurement of latent capacity was estimated by determining the change in liquid volume of the desiccant storage tank during periods when only the conditioner was in operation.

A list of monitoring points and sensor accuracy is provided in Table 10. In addition to sensors installed by MEP, outputs from the LDAC controller were transferred via Modbus communication and recorded by a Campbell Scientific CR1000 data logger.

Table 10. Sensor Accuracy Summary

Sensor	Location	Vendor	Model	Accuracy Specification
Immersed thermistor	Collector loop	Omega Engineering	TJ36-44004	±0.18 ºF
Immersed thermistor	Cooling tower	Omega Engineering	ON-910-44006	±0.18 ºF
Temperature and RH	Duct mount	Vaisala	HMD40Y	±0.36 ºF, ±2 % RH
Temperature and RH	Wall mount	Vaisala	HMW40Y	±0.36 ºF, ±2 % RH
Temperature	Supply register	Cantherm	MF52	±0.36 ºF
Turbine flow meter	Collector	Omega Engineering	FTB1431	1 % of reading
Turbine flow meter	Cooling tower water	Omega Engineering	FTB8015B-PT	1.5 % of reading
Turbine flow meter	Cooling makeup water	Omega Engineering	FTB602B-T	1 % of reading
Turbine flow meter	Desiccant	Omega Engineering	FTB6207-PS	1.5 % of reading
Differential pressure	Conditioned makeup	Setra	264	1 % of full scale
Differential pressure	Total makeup	Setra	264	1 % of full scale
Differential pressure	LDAC unit	Setra	264	1 % of full scale
Ambient pressure	Outdoor	Setra	276	1 % of full scale
Electrical energy	Regen and conditioner	Continental Controls	WNB-3D-240-P	0.5 % of reading
Current transformer	Regen and conditioner	Continental Controls	CTS-0750-30	1 % of reading
Pyranometer	Horizontal	Campbell Scientific	CS300	5 % of daily total
Level transmitter	Desiccant tank	Omega Engineering	LVU 109	+/- 0.6 cm

5.6 EQUIPMENT CALIBRATION AND DATA QUALITY ISSUES

5.6.1 Calibration of Equipment

The primary measurement instrument was the Campbell CR1000 data logger. It measured temperature, humidity, flow rate, pressure, and electric power; stored the data; and automatically relayed the data to NREL for analysis. These systems were maintained and calibrated in accordance with manufacturer's requirements.

5.6.2 Quality Assurance

Quality assurance was provided primarily by NREL's field and lab experience testing this type of device. All sensors were sampled every 10 seconds, and any mathematical manipulations of those primary measurements were also made on the same 10-second interval. Data were stored as averages or totals in four separate data tables identical in field description but varying in storage interval: 1-minute, 15-minute, 60-minute, and 24-hour (midnight-to-midnight).

Thermistor probes installed in pipes and immersed in water or glycol solution were used in calculating some of the primary energy flows of the collector and cooling tower of the LDAC system. The specified accuracy of 0.18 ºF for the immersed sensors is the best normally available accuracy for this type of sensor. The uncertainty in the energy measurement for the collector or the cooling tower depends on the temperature difference between two fluid streams. As that temperature difference becomes small, the accuracy of the energy measurement can become unacceptably large. In this application, the temperature difference for the collector loop was approximately 18 ºF on a sunny day. The temperature difference of the cooling-tower loop was typically greater than 5.4 ºF, which provides sufficient temperature difference to limit accuracy problems.

Duct-mounted temperature and RH sensors were used to calculate the sensible and latent energy removed from the conditioned airstream. The sensor accuracy of the device is ±0.63 ºF. The wall-mounted temperature and RH sensor, and the supply register temperature sensor were not used in calculation of primary energy flows, but were used as an indication of indoor comfort conditions.

A pyranometer was used to quantify the efficiency of the solar collector array. Two measurements of electric energy were made: one for the regenerator unit, including the collector, and one for the entire conditioner unit.

Differential pressure sensors were used to measure airflow rates of the LDAC unit and the main air handler. One pressure sensor was used with a pitot tube to measure the airflow rate exiting the conditioner. Another pressure sensor measured the pressure difference across the LDAC core as an additional indication of airflow rate and heat exchanger fowling over time. The measurement of liquid flow rate in the collector loop and the cooling tower loop were used in calculating primary energy flows for the LDAC system. Relative difference between the pitot tube and conditioner pressure drop is a good indicator to show consistent airflow performance.

5.7 SAMPLING RESULTS

A detailed summary of the sampling results is provided in the report's performance assessment section.

6.0 PERFORMANCE ASSESSMENT

Performance evaluation of the LDAC began in the summer of 2010. Three weeks of continuous operation was recorded during the 2010 cooling season, and around 5 months of operation were recorded for 2011. Because the majority of the LDAC system operation occurred during the summer months of 2011, the performance assessment is based on summer 2011 data. The 2010 performance data are presented to illustrate performance variability. Representative performance assessment metrics for each objective are summarized in Table 10.

Table 11 describes the performance objectives, metrics, and data requirements to determine the performance objective results, and the criteria for achieving the objectives.

Table 11. Performance Objectives

Performance Objective	Metric	Data Requirements	Success Criteria	Results
	Quantitative Performance Objectives			
Improve humidity control and comfort (energy efficiency)	• Hours outside psychrometric comfort zone • Chiller power • Reheat run-time	• Indoor temp./humidity • Chiller power • Reheat coils on	• <1% of hours outside ASHRAE summer comfort zone • Reduce chiller/reheat runtime	• Achieved but inconclusive cause • Achieved but inconclusive cause
Provide high-efficiency dehumidification (energy efficiency)	• EER • COP	• Supply-air temp./humidity • Supply-air flow rate • Ambient temp./humidity • Power consumption • Heat consumption	• EER >40 (Btu/hr)/W • >0.7 Thermal COP	• Not achieved • Achieved

Sustain high-dehumidification performance (energy efficiency and maintenance)	• Conditioner heat exchange effectiveness • Desiccant charge • Supply air pressure drop • Conditioner cooling water pressure drop • Projected service life	• Supply-air temp./humidity • Ambient temp./humidity • Desiccant chemistry and concentration • Conditioner core-air and water-pressure drop	• <5% degradation of HX eff. over 3 years • <Once-per-year desiccant/buffer adjustment • Negligible increase in air/water pressure drop • Above criteria should support >10yr service life projection	• Achieved; no degradation of desiccant during operation • Duration of performance evaluation too small to determine
	Qualitative Performance Objectives			
Maintainability (ease of use)	Ability of an HVAC technician to operate and maintain the technology	Standard form feedback from the HVAC technician on usability of the technology and time required to maintain	A single facility technician able to effectively operate and maintain equipment with minimal training	• Many unforeseen maintenance issues occurred during initial demonstration • Many lessons learned for design and ease of operation

6.1 IMPROVE HUMIDITY CONTROL AND COMFORT

The LDAC provides primarily latent cooling to the air, so the total cooling is representative of dehumidification. To show the effect of latent cooling, Figure 20 and Figure 21 illustrate the decrease in humidity of the air.

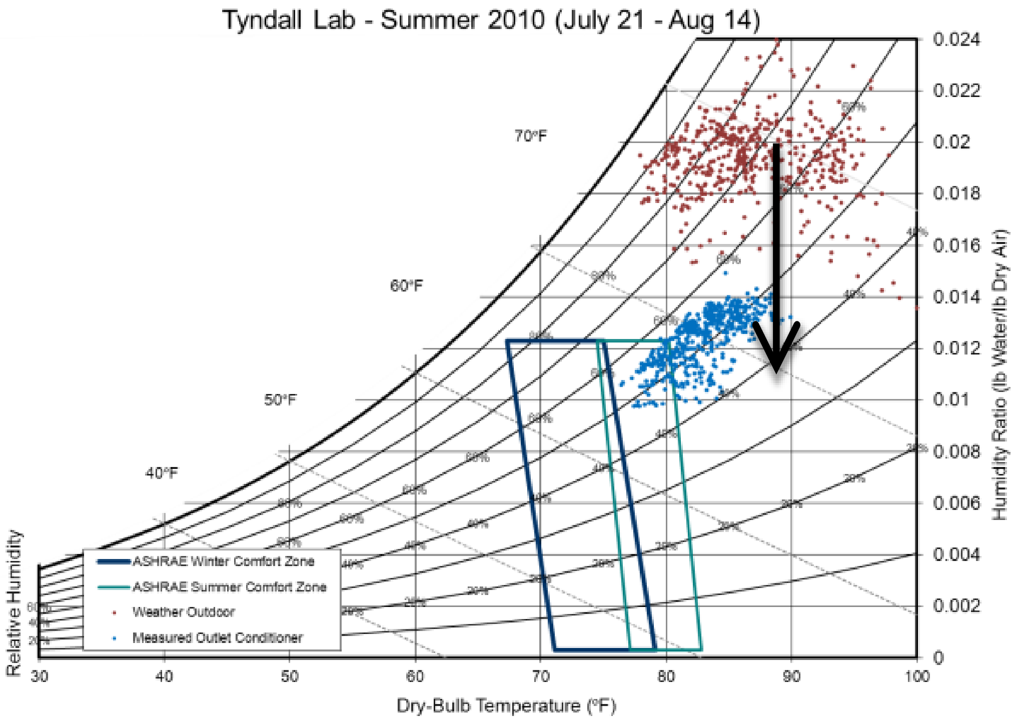


Figure 20. Outdoor air conditions and conditioner-outlet air conditions for summer 2010 operation

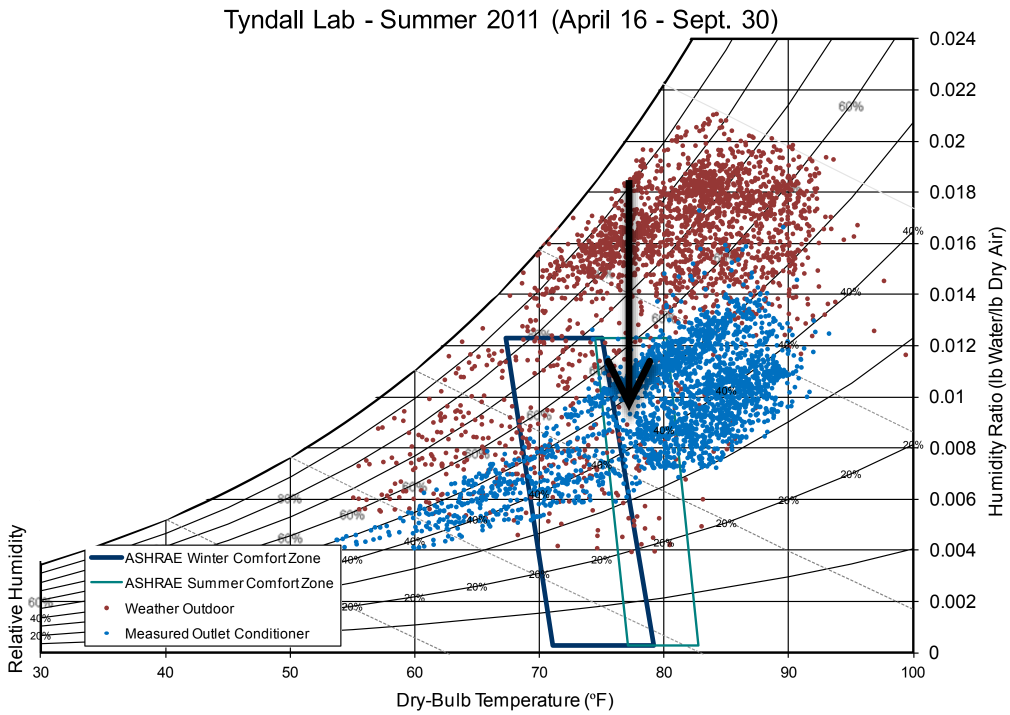


Figure 21. Outdoor air conditions and conditioner-outlet air conditions for summer 2011 operation

During the 2010 cooling season, the unit dehumidified the air into the 40% to 55% RH range and was not as close to the RH equilibrium point of ~35% RH for $CaCl_2$ desiccant solution because the system was operating on a weaker desiccant.

The LDAC unit noticeably reduced the enthalpy of the outside air during the 2011 cooling season, and the RH of the outside air was reduced to 35% to 60% RH throughout the summer. As noted above, the drying potential of the system is based on the concentration of the desiccant entering the conditioner, and the system is operating with a weaker desiccant when the conditioned air RH is in the 40% to 60% range.

Figure 22 illustrates the daily cooling supplied by the LDAC during the 3 weeks of summer 2010 operation. The daily average cooling was 79.3 ton-hr.

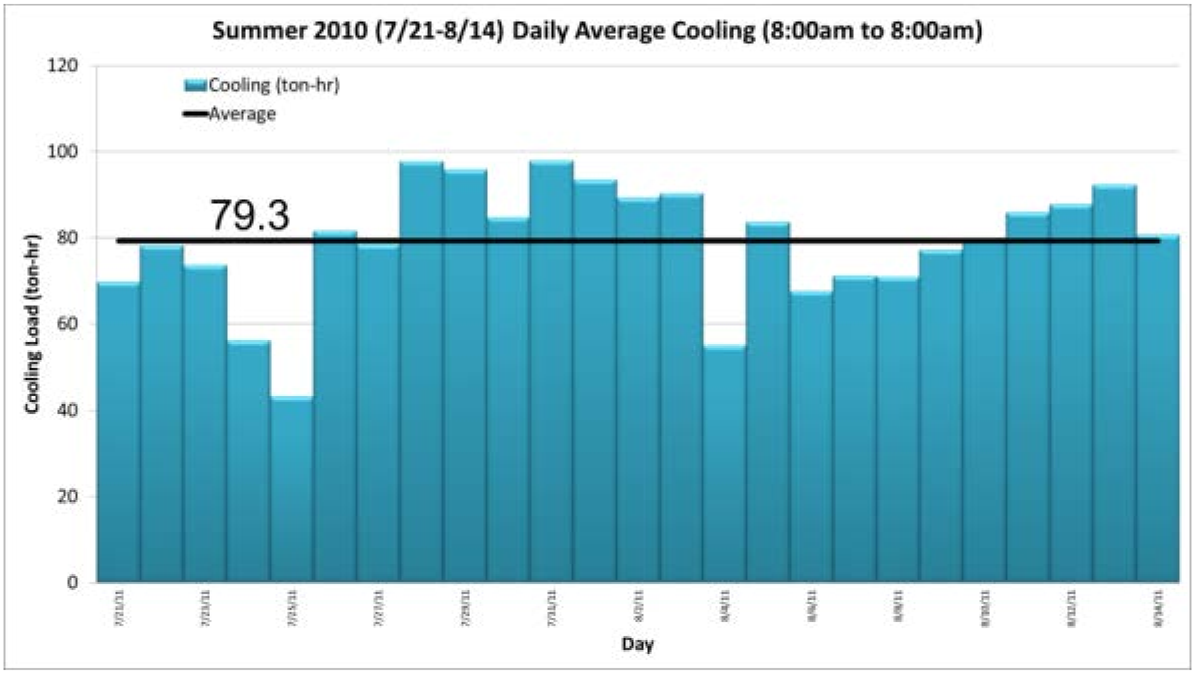


Figure 22. Daily cooling load supplied by the LDAC during the 3 weeks of summer 2010 operation

Figure 23 shows the total cooling load provided by the LDAC during the summer of 2011. Several days were omitted from Figure 23 because of low or no operation on those days. The average daily cooling is 56 ton-hr, which is representative of a 10-ton cooling system that operates for 5.6 hr during the day. This is below the expected cooling load achievable from this system, which is greater than 100 ton-hr/day. The reason the cooling load was lower than anticipated was primarily based on the fact that the supply fan for the conditioner was operated at less than 75% of the rated flow rate (3,000 cfm) for the 2011 cooling season, and the unit was not providing the full capacity that it was designed to provide. In addition, the control sequence for the desiccant storage tank was set up to allow for more total hours of cooling per day at a weaker desiccant concentration, which also had the effect of reducing the cooling potential of the unit.

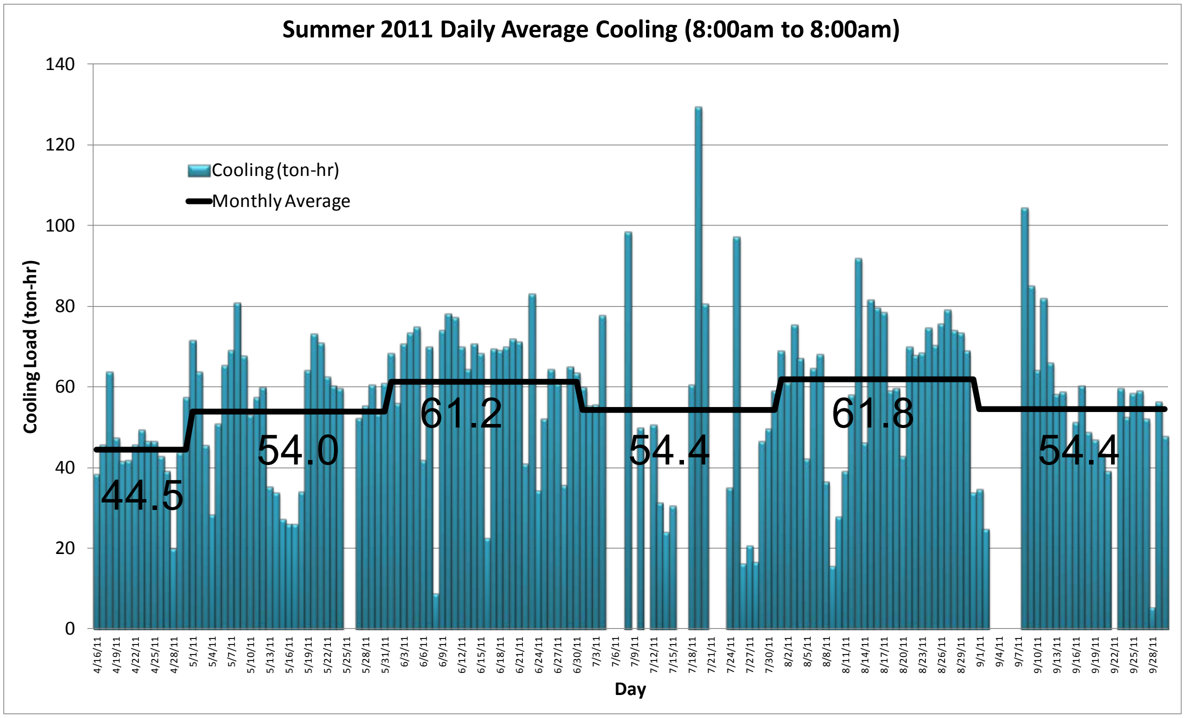


Figure 23. Daily cooling load, with monthly averages, supplied by the LDAC during the summer of 2011

6.1.1 ASHRAE Comfort Zone

The criterion for improved comfort is the number of hours indoor air conditions (temperature and RH) fall outside of the "comfort zone," which has been established by ASHRAE, as shown in Figure 24. Because the Tyndall AFB also uses a chiller for conditioning the air with a capacity much larger than the LDAC (about 10 times larger), the resulting supply air conditions are not fully dictated by the LDAC.

The success criterion for comfort is achieving less than 1% of hours outside of the comfort zone. The comfort zone for the summer is shifted to warmer air temperatures because less clothing is worn during the summer; for a laboratory and AFB, more clothing is worn than at civilian facilities, so the winter comfort zone is more applicable for the Tyndall AFB for certain occupants. This success criterion was not met for 2010 cooling season but was met for the 2011 cooling season.

To illustrate the performance of the chiller alone, Figure 24 shows the outdoor air conditions and the laboratory air conditions from summer 2010.

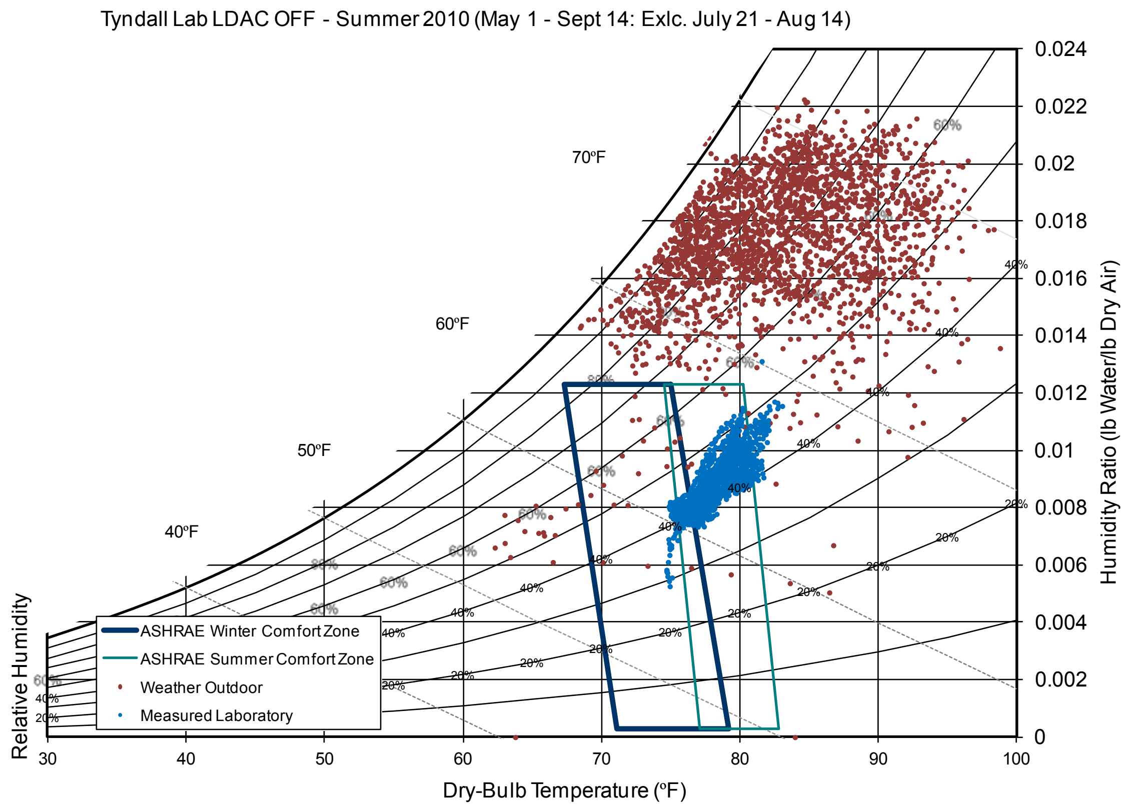


Figure 24. Tyndall Lab outdoor and indoor conditions *without* the LDAC operating in Summer 2010 (4.3% of the data is outside the comfort zone)

The existing HVAC system and air-cooled chiller have the ability to maintain the space within the comfort zone for the majority of the summer; the lab is outside of the comfort zone 4.3% of the time (when the LDAC is not operating). During the 3 weeks of LDAC operation in 2010, the system met comfort conditions 10.6% of the time. It is not apparent from this data that the LDAC system has a significant effect on the indoor conditions for the 2010 cooling season. There are two reasons that this could be the case: the first is related to the fact that the system was operating on weak desiccant and wasn't drying the air to its full potential; and secondly, the LDAC unit is treating less than one-third of the outside air for this AHU. If it was treating 100% of the outside air, the total moisture removal rate would be significantly higher.

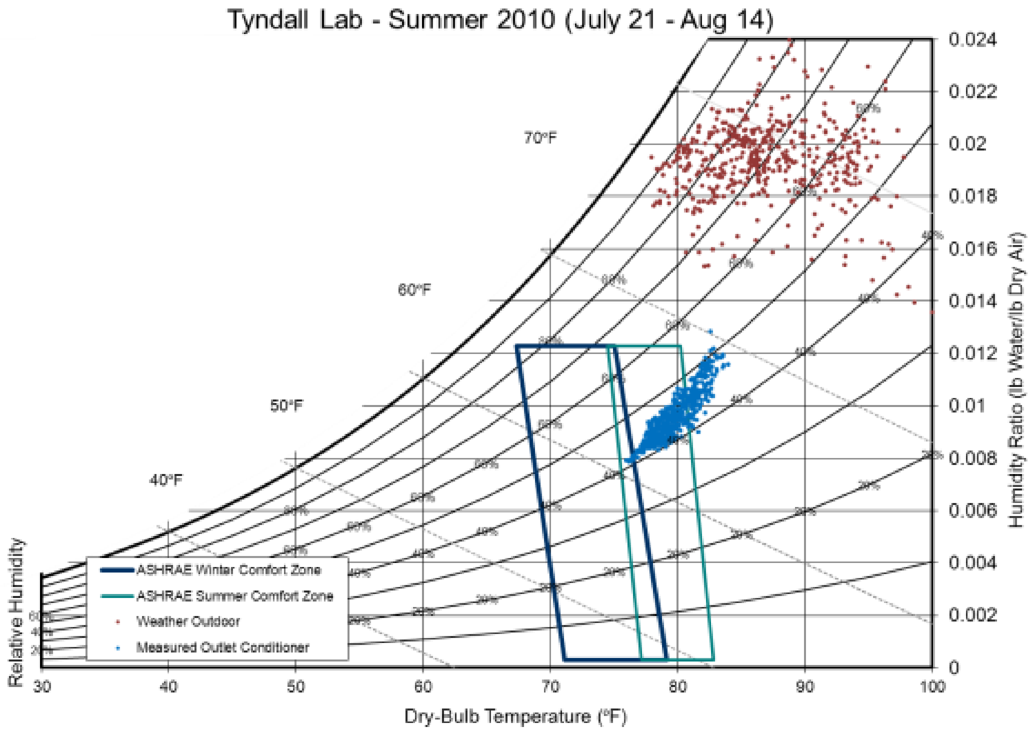


Figure 25. Tyndall Lab outdoor and indoor air conditions during summer 2010 with the LDAC in operation (10.6% of the data is outside the comfort zone)

In the summer of 2011, the LDAC operated for most of the summer; the outdoor air and laboratory conditions are shown in Figure 26. Although the outdoor air conditions are slightly different for the 2010 and 2011 summers, it is clear that the indoor air is drier and cooler. The lab is outside of the comfort zone only 0.2% of the time, which is below the success criterion of 1%. This includes both the summer and winter ASHRAE comfort zone as it is difficult to differentiate which one should be applied on a military base because a number of occupants are wearing full military gear during the summer months, which would indicate that the winter comfort zone is more appropriate than the summer comfort zone for those individuals. As mentioned, the chiller is used to cool the air in each zone's AHU, and the overcool/reheat cycle is also used to control humidity. Therefore, it cannot be conclusively stated that the LDAC system credibly achieved the comfort criterion.

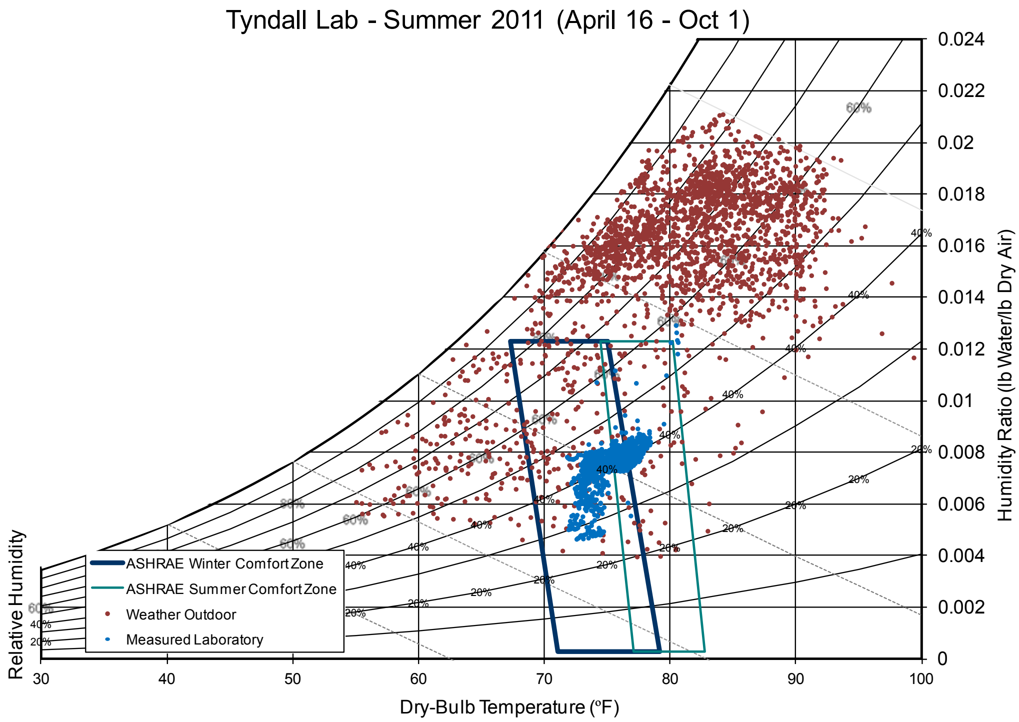


Figure 26. Tyndall Lab outdoor and indoor air conditions during summer 2011 with the LDAC in operation (0.2% of the data is outside the comfort zone)

6.1.2 Reduce Reheat Run-Time

While there is evidence that the LDAC system achieves a better range of indoor air conditions, the LDAC system was undersized for this particular application, and the chiller also controls humidity by overcooling the air to condense out the water and reheating it to a suitable supply temperature. The overcooling and reheating process adds significant cooling load to the chiller when there is a high ventilation requirement. The LDAC mitigates this process by drying the humid air before being sensibly cooled with the chiller.

The overcool/reheat process only occurs when the humidity is high. The amount of time this occurs can be determined by comparing the discharge air temperature from the main AHU to the supply-air temperature into the space (after the terminal-unit reheat coil), which activates the terminal-unit reheat coils when the temperature is too low for comfort. The discharge-air temperature was only measured after one terminal unit; and if the supply-air temperature after the terminal unit was greater than 3.5 °F above the discharge air/supply air temperature from the AHU, the reheat coils were assumed to be activated, meaning the overcool/reheat process occurred. The 3.5°F temperature differential represents about 10 times the normal differential from ductwork heat gain that was monitored in this particular facility. This method was used to approximate the reheat run-time rather than measuring the hot water reheat system directly.

In the summer of 2010, while the LDAC was off, the reheat coils activated 8.9% of the time. During the 3 weeks when the LDAC was running, the reheat coils activated 2.4% of the time. The LDAC operated for the majority of the 2011 summer, and the reheat coils activated just

1.4% of the time. Although these data are limited, they indicate that the LDAC has mitigated the necessity of the overcool/reheat cycle, and this performance metric was successfully met.

6.2 PROVIDE HIGH-EFFICIENCY DEHUMIDIFICATION

While it is apparent that the LDAC system can dehumidify air to improve comfort and mitigate the overcool/reheat cycle, it must do so while improving the efficiency of the total AC system. Several common metrics are commonly used to characterize the performance of AC systems, such as EER, kW/ton, and thermal COP. The LDAC system aims to increase the overall efficiency of the existing cooling process.

6.2.1 Electrical Efficiency: EER and kW/ton

EER is defined as the ratio of cooling supplied (Btu/hr) to the electricity [watt (W)] required by the system:

$$EER = \frac{Cooling\ Supplied\ \left(\frac{Btu}{hr}\right)}{Electriciy\ Required\ (W)}$$

Figure 27 illustrates the daily electrical performance of the LDAC for the 3 weeks of summer 2010 operation. The average EER for the 3 weeks was 14.7 (Btu/hr)/W, a 63% improvement over the rated EER of 9.0 (Btu/hr)/W of the existing chiller.

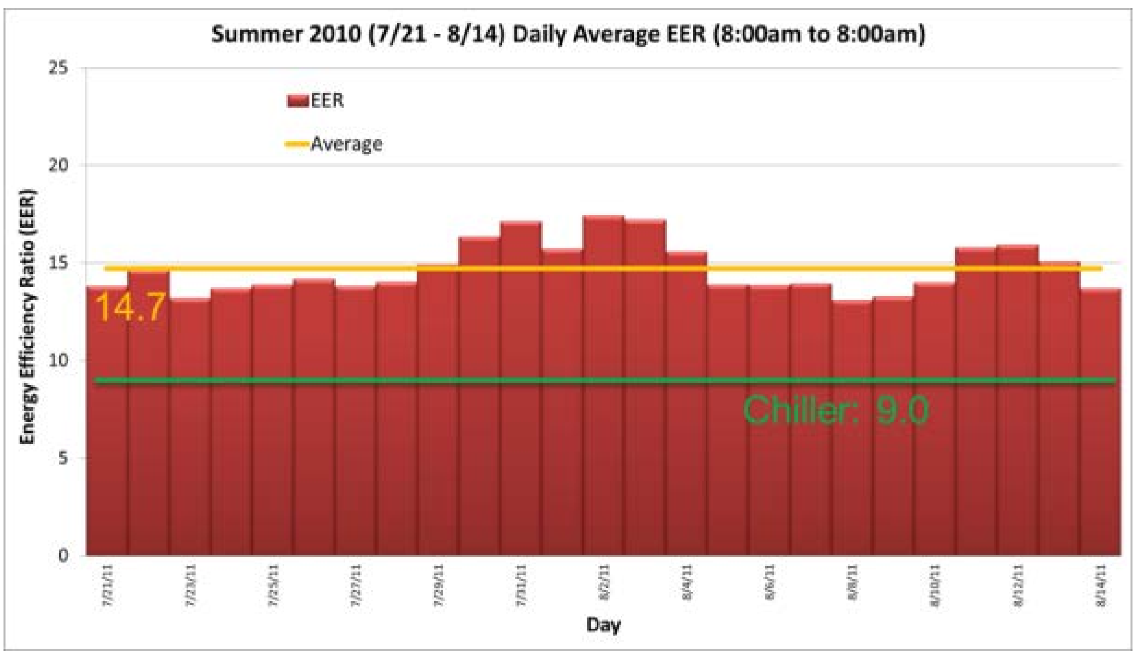


Figure 27. Daily average of LDAC EER for 3 weeks of summer 2010 operation

Figure 28 shows the daily EER performance of the LDAC system for summer 2011. It is apparent that the performance improved steadily throughout the summer. This is because various adjustments were made to the system to operate at near maximum capacity while limiting auxiliary power consumption. As indicated by the red star in Figure 28, a VFD was installed on the cooling tower pump to lower power consumption during lower capacity operation; this modification was critical to lowering the power consumption from the cooling tower and raising the EER.

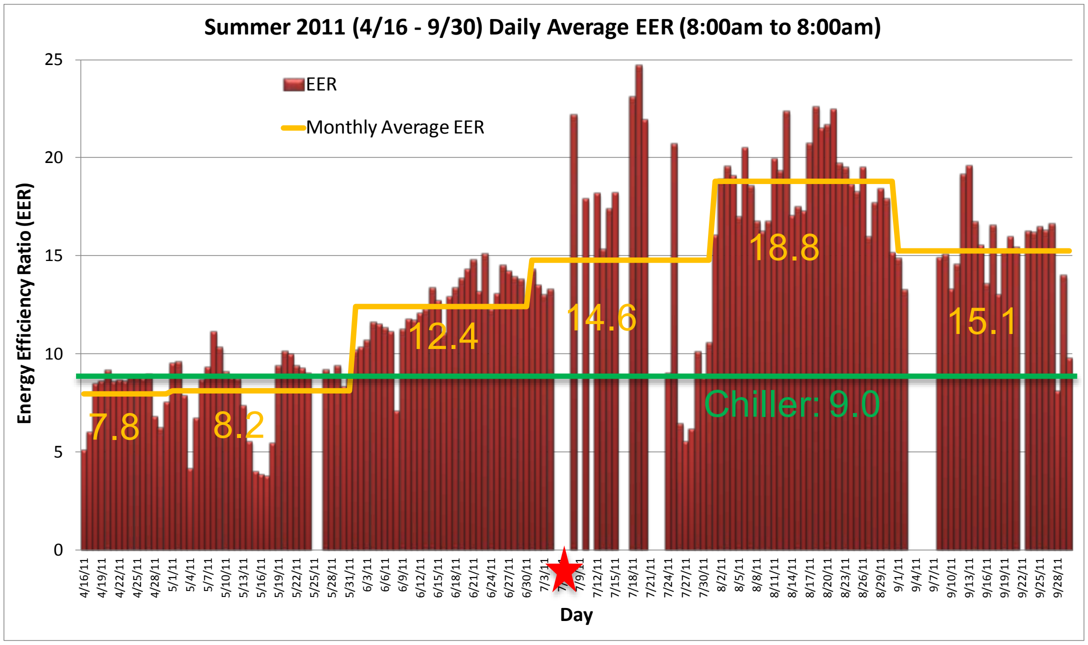


Figure 28. Daily and monthly average of LDAC EER for summer 2011

The red star indicates when the higher efficiency cooling tower pump motor and VFD were installed. As shown in Figure 28, the EER of the LDAC is between 15 and 20 for August with an average of 18.8. The best single-day performance was achieved in July at 25 (Btu/hr)/W. Although the LDAC outperformed the existing chiller in terms of EER, the success criterion for electrical efficiency is an EER of 40, *so the system did not meet this objective*.

Figure 29 shows the dependence of EER on cooling load.

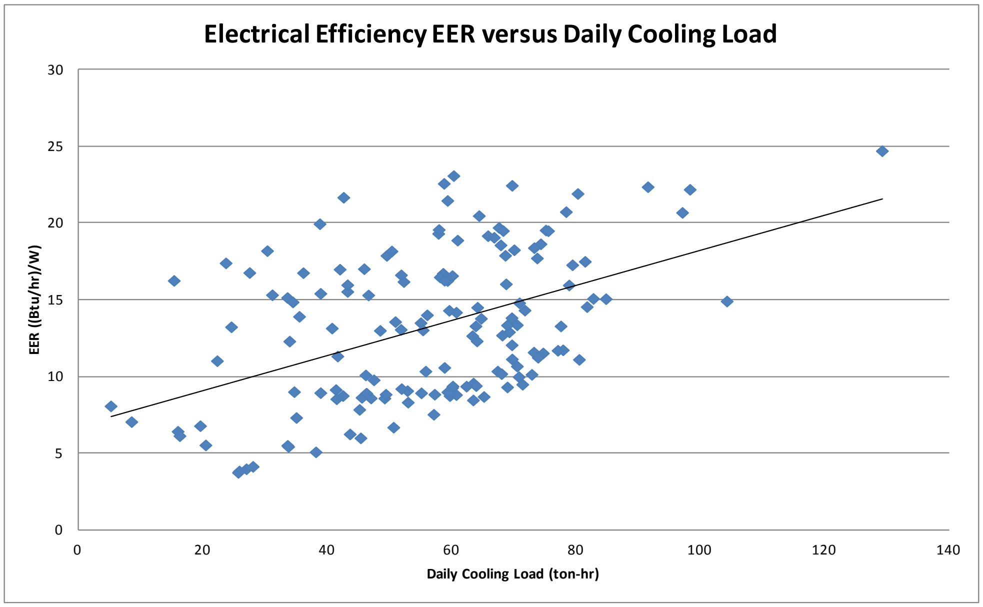


Figure 29. EER dependence on cooling load

While there is much scatter in the data due to variability in outdoor air conditions, solar availability, and operational adjustments, there is a positive linear correlation between EER and cooling load. Above a daily cooling load of 70 ton-hr, the system operates at an average EER of about 15 (Btu/hr)/W. The following list characterizes the reasons why the unit was not able to operate at an EER of 40:

- *Cooling tower fan/pump* - Several issues caused the system to operate at lower than expected efficiency during April and May of 2011. The pump and fan of the cooling tower were set at a fixed flow and speed, which led to excessive electrical energy use to meet a given cooling load. The cooling tower pump required 1.2 kW of electric power, and the motor had an efficiency of 62%. The temperature difference across the cooling tower was also small (~2°F), which indicated that excessive flow rates were present. A new cooling tower pump was installed, in conjunction with a VFD in July 2011. A new control sequence was written that maintained the cooling tower fan at 75% fan speed and modulated the cooling tower pump flow rate to maintain a 10°F temperature difference between the cooling tower supply and return water temperatures. This resulted in over a 65% reduction in cooling tower energy use.

- *Conditioner supply fan settings* – The conditioner supply fan was operated at partial fan speeds for the 2011 cooling season and was not providing the full 3,000 cfm of conditioned air. This limited the cooling capacity of the unit and the overall efficiency. Although this was fixed towards the end of the 2011 cooling season, the cooling season was coming to an end and the amount of cooling that could be accomplished was reduced.

- *Conditioner fan power* – The long 100-ft duct run from the conditioner to the outside air intake resulted in a significant pressure drop and the need for a large supply fan. The supply fan was rated at 3.45 hp and 2,570 W. If the conditioner was located directly at the outside air intake, it could reduce the supply fan horsepower to below 1 hp and the fan power by 70%.

- *Desiccant concentration* – The system was operated with a weaker desiccant during the majority of the demonstration as a way to ensure the system would work because of problems with the solar field. With this particular system, the rate of water absorption from the conditioner and removal from the regenerator is significantly different because of the variability in total heat production from the solar field; and the storage tank has to operate as a buffer between the two components. The settings were modified towards the end of the 2011 cooling season to operate on a stronger desiccant, but the conditioner unit did not operate on a fully strong desiccant throughout the day.

- *Pump and fan efficiency and part-load performance* – Because this was a solar demonstration, additional fans and pumps were needed to move the desiccant around, and many of them had really low electrical efficiencies (less than 60%). In addition, a number of pumps and fans were constant volume units and used the same amount of power regardless of the total cooling capacity. Because the unit was only providing about one-half of its rated cooling capacity and using the same amount of electrical power as it would if it was providing 100% of the cooling capacity, the total electrical efficiency is reduced.

In previous non-solar LDAC demonstrations, the conditioner and regenerator are located in the same enclosure, and the solar pumps are eliminated from the system. These systems, when located directly next to the outside-air intake, have successfully demonstrated electrical efficiencies above EER = 60. These issues and recommendations for future installations are provided in the lessons learned section of the report.

6.2.2 Thermal Efficiency – Coefficient of Performance

The daily thermal performance of the LDAC system during the summer of 2011 is shown in Figure 30. This performance metric accounts for the thermal input and output of the LDAC unit, which represents the solar heat utilization of the system.

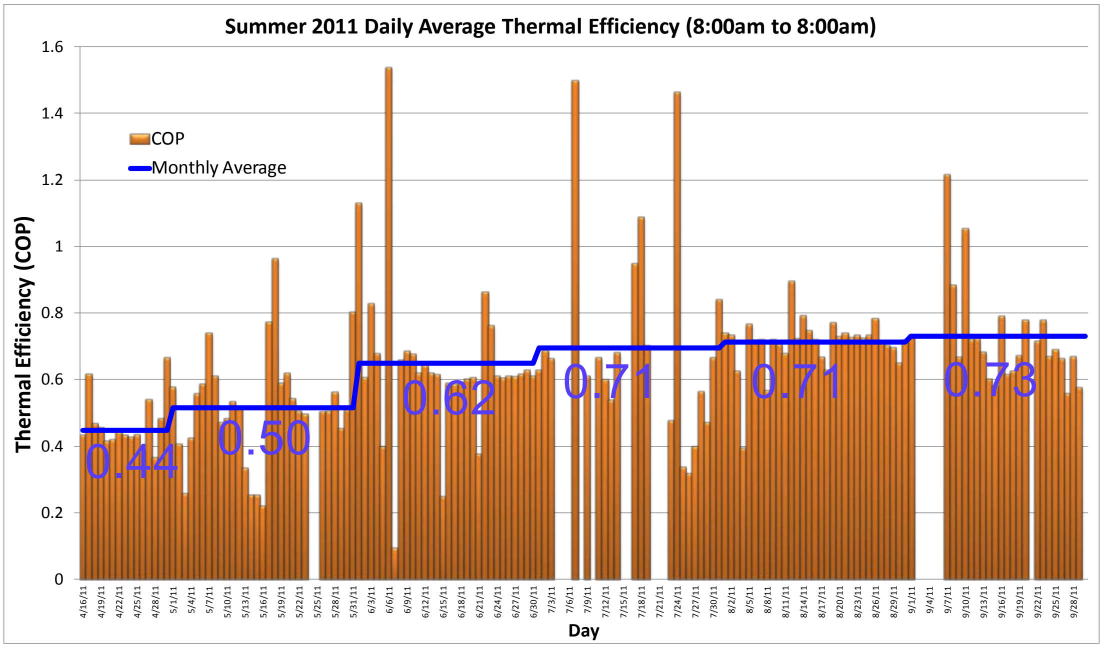


Figure 30. Thermal performance (COP) of LDAC for summer 2011

While there is no fuel cost from this heat source, it is still beneficial to utilize the maximum amount of the captured heat. The regenerator heat measurement for the summer 2010 operation was limited to 3 days, so a graphic is not shown here; the average COP of these 3 days was 0.85, where COP is defined as:

$$COP_{thermal} = \frac{\Delta h_{air}(Cooling\ Supplied)}{\Delta h_{Hot\ Water}(Solar\ Thermal\ Energy)}$$

It should be noted that the system achieves a COP of greater than 1 on a few days due to excess desiccant storage from the previous day. The COP increased throughout the summer due to an increase in solar availability and the amount of cooling the system achieves. *The success criterion for thermal COP of 0.7 was achieved for the system.*

Another metric to compare the efficiency of the heat source usage is the solar efficiency. While the solar heat source is "free" after the capital costs and operational costs are accounted for, it is useful to know how efficiently the system is using the available solar energy to maximize its value. The solar efficiency is defined as the ratio of the heat transferred to the regenerator from the solar loop to the global horizontal irradiance (GHI) in W/meter^2 times the solar array absorber area:

$$Solar\ Efficiency = \frac{Heat\ to\ Regenerator}{GHI * A_{absorber}}$$

The solar efficiency was relatively constant at an average value of 68% throughout the summer, as shown in Figure 31.

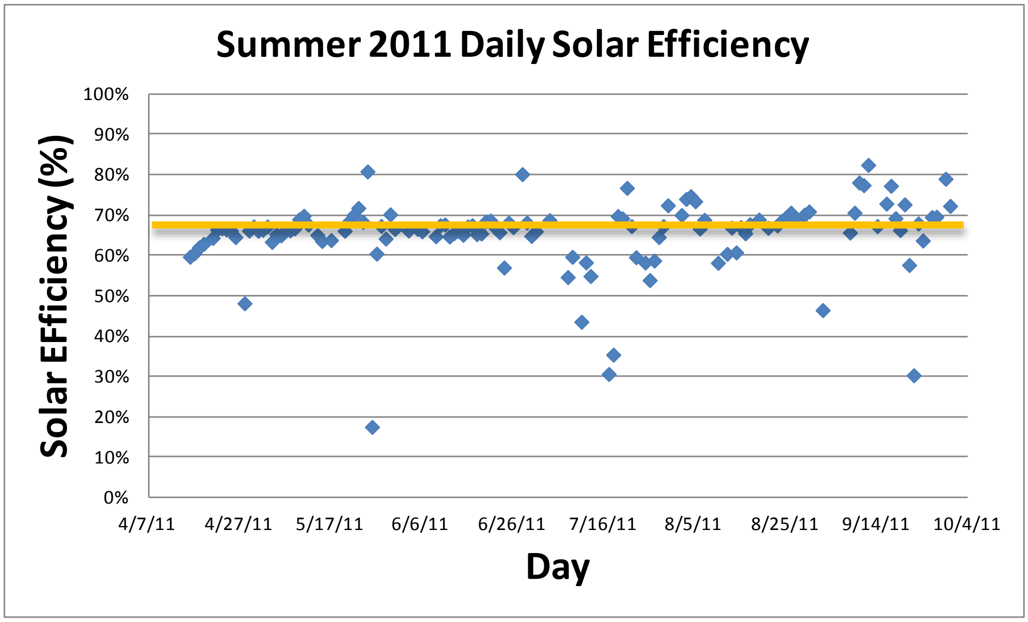


Figure 31. Solar efficiency for summer 2011

6.2.3 Water Consumption

Figure 32 illustrates the monthly water usage (gal) for each ton-hr of cooling. Several issues in the first 2 months, such as blow-down pipe leakage and over-circulation of cooling tower water (evaporation losses), caused the system to use a large amount of water. The issues were corrected in June and July 2011, and the average water usage for the rest of the summer was 1.3 gal/ton-hr. This is in line with existing arguments, but also suggests that the air temperature out of the cooling tower was higher than ambient because water consumption was less than 1.55 gal/ton-hr.

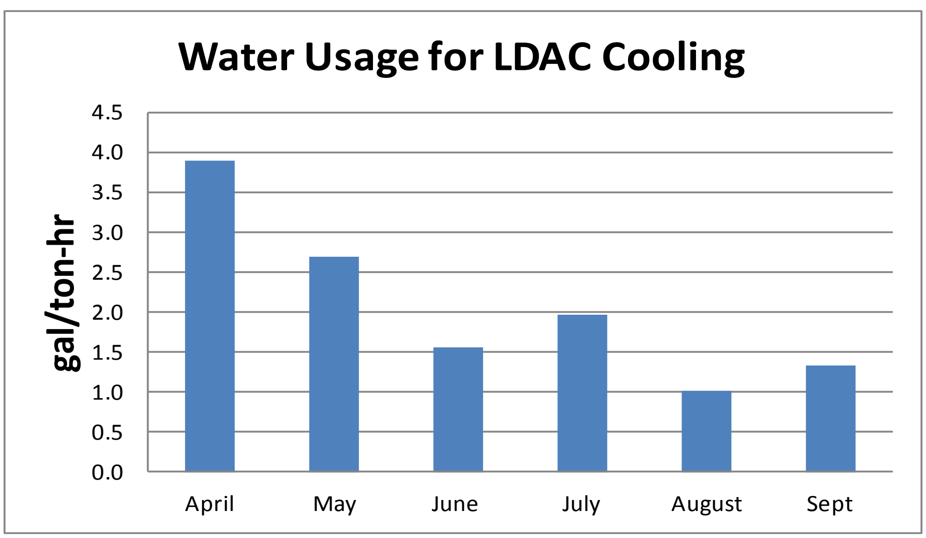


Figure 32. Water usage for LDAC cooling operation

6.3 PERFORMANCE SUMMARY

The 3 weeks of performance data for 2010 are summarized in Table 12.

Table 12. Summer 2010 (3 Weeks) Performance Summary

Date	Cooling (ton-hr)	EER [(Btu/hr)/W]	kW/ton	Solar heat (MBtu)*	COP*
7/21/10 - 8/14/10	1982	14.7	0.82	3.1	0.85

*Solar thermal generation only recorded for 3 days (7/21-7/23)

Table 13 provides a listing of electrical EER, electrical kW/ton, and thermal COP for each month in 2011. It is clear that the performance improved throughout the summer, both in electrical and thermal efficiency.

Table 13. Monthly (Averaged) Performance for Summer 2011

Month	Cooling (ton-hr)	EER [(Btu/hr)/W]	kW/ton	Solar heat (MBtu)	COP
April	667	7.8	1.54	18.1	0.44
May	1565	8.2	1.47	39.9	0.5
June	1837	12.4	0.97	35.4	0.62
July	1142	14.6	0.82	19.4	0.71
August	1916	18.8	0.64	32.2	0.71
September	1300	15.1	0.79	26.7	0.73

Figure 33 annotates the timeline of the LDAC performance to synchronize the events and changes that occurred to the LDAC system throughout the summer of 2011 with the performance metrics described in the previous sections.

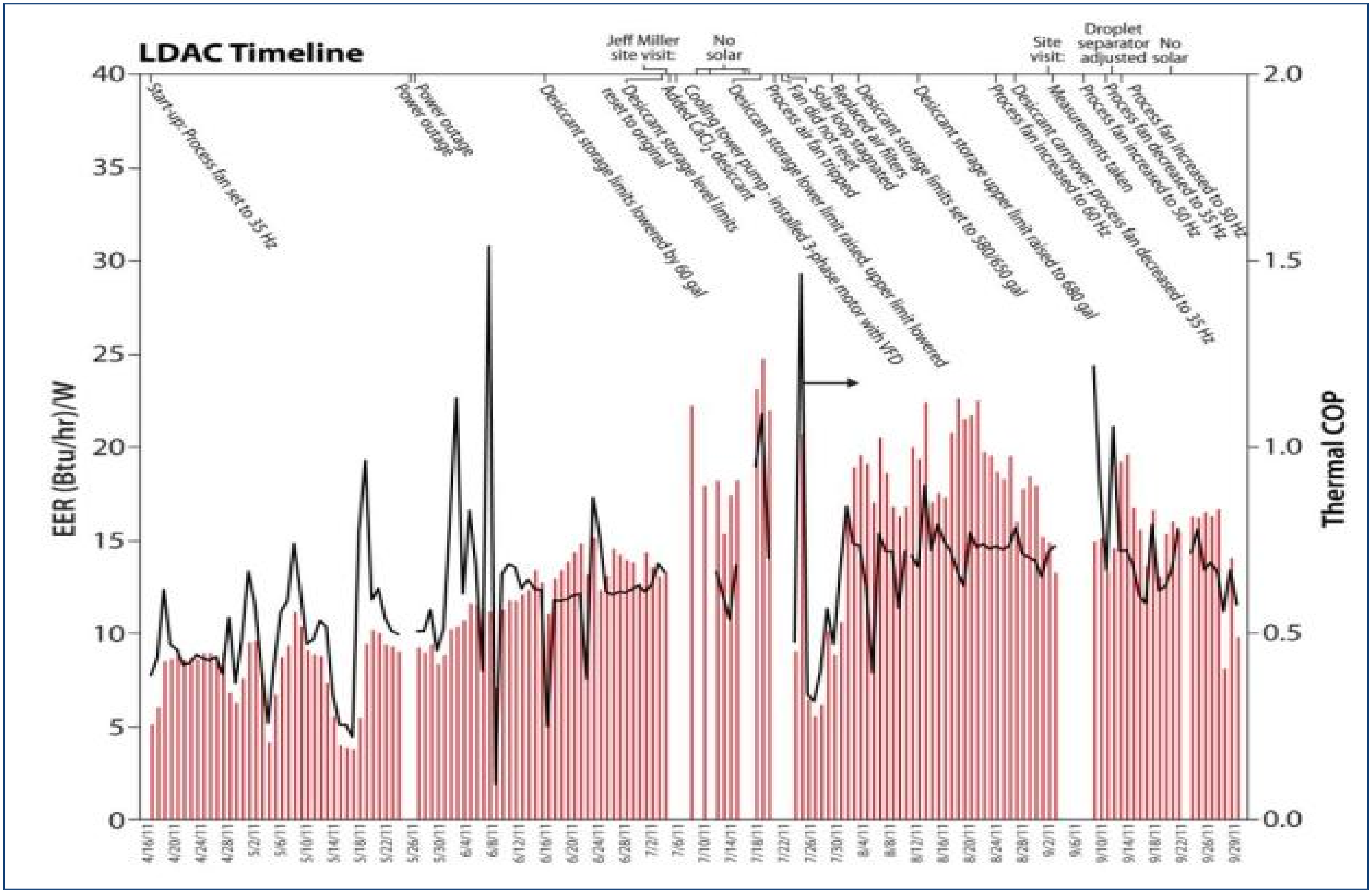


Figure 33. Timeline with electrical (EER) and thermal (COP) efficiency performance labeled with LDAC system events and changes

6.4 SUSTAIN HIGH-DEHUMIDIFICATION PERFORMANCE

Several metrics were defined for the quantification of sustained performance: conditioner heat exchange effectiveness; desiccant charge; supply air and cooling water pressure drop; and projected service life. Because the system only operated for about 6 months at various operating points, these metrics are difficult to quantify.

6.4.1 Conditioner Heat Exchange Effectiveness

The conditioner heat exchange effectiveness is defined as the change in enthalpy between the outside air and discharge air from the conditioner divided by the outside air enthalpy minus the desiccant enthalpy:

$$\varepsilon = \frac{\Delta h_{air}}{h_{air,in} - h_{des,Tcold}}$$

Because the unit only operated for a couple of weeks during the 2010 cooling season, it is not possible to calculate a change in heat exchanger effectiveness.

6.4.2 Desiccant Charge

This metric quantifies the addition or removal of desiccant that must occur during the course of a year. Because the system only operated for one complete cooling season, there is not enough statistical data to quantify this metric. Additional salt (in solution form) was added to the system in June 2011 because the original concentration calculations overestimated the desiccant concentration. This addition was not associated with a loss of desiccant concentration over time. Kathabar liquid-desiccant systems have run for many years without requiring replacement or adjustment of the desiccant charge; however, it is appropriate to inspect the desiccant at least once a year.

There are also times when the desiccant solution picks up contaminates from the air or from cooling water or boiler (solar) water leaks. For example, the 3,000-cfm Kathabar unit that ran for a number of years at the University of Maryland CHP test facility was located too close to the exhaust of a natural gas engine. The desiccant in that unit absorbed combustion products and became very acidic over time. These issues can be properly addressed with adequate planning and design.

6.4.3 Pressure Drop Increases

It is necessary to sustain low-pressure drops in the system to maintain high-efficiency cooling. The quantification and statistical certainty of this metric also suffers from the relatively low time span in which the system was operational. Several maintenance issues can be addressed to ensure low-pressure drop for both the process air and the water flows.

Air filters for the regenerator and the conditioner must be changed regularly, depending largely on the quantity of outdoor air particulates. The filters at Tyndall AFB have lasted between 2 and 3 months, while filters on a system in Los Angeles, California, must be changed every month.

Desiccant and water filters can also cause an increase in pressure drop if not inspected. One major issue, which increased the pressure drop across the cooling tower unit, was due to not maintaining biological growth control. This issue caused the cooling tower water filter to clog and required replacement every 4-6 weeks. This issue was resolved by adding biocide to the water. Additionally, a new strainer replaced the filter, which decreased the pressure drop. With proper design and filter selection, pressure drop can be reduced and maintained without intensive man-hours.

6.4.4 Service Life Projection

In order to achieve attractive life cycle cost and energy savings, the system service life must be greater than 10 years. While demonstrating a new technology over the course of 2 years, this projection does not come without uncertainty. It is expected that the service life for the LDAC system is greater than 10 years, and proper design will support this projection.

In particular, it is expected that the non-desiccant components in a well-installed and maintained system—including the solar field, the cooling tower, the storage tank, and the balance of the balance systems (enclosures, fans, pumps, controls, etc.)—would have a service life greater than 10 years. It is expected that some component failures would occur over time, such as loss of a pump, an instrument failure, etc. This is not uncommon for many HVAC systems.

Of the three key desiccant components (i.e., conditioner, regenerator, and interchange heat exchanger), the system that should be closely monitored is the regenerator (the actual plastic parallel-plate HMX inside the regenerator unit). Some degradation in the flock on the plates running in the field has been observed on the Los Angeles Whole Foods LDAC units. This problem has not been seen on two earlier prototypes, and so far there is no sign of the same problem with the Tyndall regenerator. The conditioners and interchange heat exchanger have held up well in the field on all previously demonstrated units up to this point.

6.5 MAINTAINABILITY

Ideally, regular maintenance would be limited to tasks such as inspecting and changing filters; in practice, this system has required much more attention. The largest share of those issues stem from the poor initial design and installation of the solar field. The most significant issue is that the solar system has and continues to experience stagnation problems. With no secondary hot-water load and no heat dump built into the system, stagnation happens occasionally. A system change was made to run the collectors on water and not glycol, which means that the system is no longer freeze protected; and therefore, must be shut down and drained during the winter. The cooling tower is also not freeze protected. Protecting the system against both stagnation and freezing is essential to run the system over the course of 10 years.

There were also a number of issues with the control of the system that required attention. For example, there have been zero offset shifts on the analog-to-digital cards that read the 4-20 milliamp signal from the desiccant storage tank level sensor. This seemed to occur at times after an electrical storm and was resolved by adding a lightning surge arrestor.

There are other issues with the VFD/fan motor/PLC proportional-integral-derivative loop for the process fan. The VFD/fan motor has faulted and failed to automatically reset on several occasions. The fault occurs infrequently and is difficult to reproduce for determining the source of the problem. As a result, the process fan was set at a fixed speed.

Many of the performance and maintenance issues, were identified and resolved throughout the 2011 summer of operation. This demonstration has undoubtedly been a “lessons learned” experience, and many issues will be improved upon to reduce or eliminate unnecessary maintenance and/or repair. A single facility technician should be able to maintain the system with a minimal amount of training for future projects, but that was not the case for this particular project.

7.0 MARKET ANALYSIS

7.1 COST MODEL

Table 14 summarizes the displaced load on the chiller and the approximate energy and cost savings from the LDAC. It should be noted that these savings may slightly underestimate the actual savings because excess cooling due to the overcool/reheat cycle, which is mitigated by the LDAC, is not accounted for in the analysis.

Improved performance in August 2011 led to the largest energy and cost savings, which is indicative of the performance potential of the LDAC system. Unforeseen maintenance and operation issues arose during the summer months, and this hindered the sustained high performance of the system.

Table 14. Energy and Cost Savings from the LDAC in 2011

Month	Cooling (ton-hr)	Chiller Elec. (kWh)	LDAC Elec. (kWh)	Elec. Savings (kWh)	Elec. Cost Savings ($)
April	667	890	1,026	-137	-14
May	1,582	2,110	2,325	-215	-21
June	1,837	2,449	1,774	676	68
July	1,239	1,652	1,131	521	52
Aug	1,916	2,554	1,223	1,331	133
Sept	1,333	1,778	1,099	678	68

The total cost savings for the 2011 cooling season was $321. The installed costs for the solar thermal system were $170,000, and the installed costs for the LDAC components were $40,000, for a total installed cost of $210,000, and a simple payback of 654 years. Because this was a pre-commercial system, the simple payback is not indicative of the paybacks of a commercial system. If the system would have operated per design intent, the cost savings would be substantially higher. In addition, in building types with electric reheat, the zone-level reheat savings dwarf the energy savings from the mechanical chiller. Reheat energy use in hospitals for example has been documented to account for over 30% of the total energy use. Finally, when the system is coupled with solar thermal, the solar thermal component becomes the most expensive part of the system and solar incentives or high utility rates are required to offset the increased costs of the solar thermal system.

One of the first commercial LDAC systems is being installed at the Coral Reef Fitness and Sports Center on Andersen AFB in Guam. A 6,000-cfm conditioner was designed for this system. The power requirements per ton of cooling for the existing building level chiller and LADC are 1.05 kW/ton and 0.3 kW/ton, respectively. Note that the power requirement of the chiller does not account for the chiller water pumps, so the power requirement may be slightly greater in reality. The system is designed with an evacuated-tube solar thermal field supplying 80% of the thermal power and a backup diesel-powered boiler providing 20% of the thermal power. The system is expected to reduce HVAC energy use by 34% and save $145,395 per year with an estimated simple payback of 11.6 years.

7.2 RELEVANT MARKETS

The LDAC system typically used for outdoor air *dehumidification*, and an electric chiller is typically required to sensibly cool the air to the desired temperature. The energy consumption from the LDAC includes heat for regeneration and electricity for the pumps and fans in the system. The LDAC is most suitable where:

- The existing HVAC system is not able to meet latent loads on a facility
- Humidity control is required
- Overcool/ reheat strategies are used in traditional HVAC systems
- Large quantities of ventilation air are needed

The LDAC should be applied to hot/humid climates that require year-round cooling and de-humidification. Future installations should focus on facilities in ASHRAE climate zones 1A and 2A. A full market analysis with a listing of appropriate building types, locations, design characteristics, and life cycle costs will be provided in the latest DoD LDAC demonstration in Guam.

8.0 IMPLEMENTATION ISSUES

The project's focus was necessarily changed to focus on discovery of technical issues with this new emerging technology. Many of the issues arose because the installation had many unique features including the following:

- The demonstration was the first combination of solar heat with this type of LDAC system.
    - Due to initial budgetary constraints, the LDAC relied solely on solar heat with no natural gas backup to ensure that the unit operated throughout the cooling season. A properly designed system that uses solar heat will have backup. Due to this, the system did not achieve peak-cooling capacity for significant hours of operation. Because the system largely has static power draw, this resulted in a low average EER.
    - The solar field designer and LDAC system design were not tightly coordinated by the prime installation contractor (Regenesys). This resulted in a design that did not consider the frequency and duration of stagnation periods for the solar field. The collector design was not designed to withstand more than about two stagnations per year. Furthermore, the collector system was not initially designed to withstand the massive volume of steam from these collectors when stagnation occurred. The solar field required significant redesign. The end result was workable for the demonstration despite being problematic and suboptimal in operation.

- The demonstration was the first to create a split system where the conditioner and regenerator were contained in separate packages and separated by around a 100-ft distance. This technical challenge resulted in a suboptimal pumping design configuration because of the necessary pump size to transfer desiccant this distance. Future designs should reduce the distance from the regenerator and conditioner.

- This demonstration was the first to have 10 hours of desiccant storage using $CaCl_2$. Tuning the storage to achieve optimal efficiency was required. The desiccant charge and the tank's low and high levels have significant impact on efficiency, capacity, and solar utilization. These variables were tuned as the demonstration progressed.

- This demonstration required the placement of the conditioner unit about 100 feet from the outdoor intake to the building. This required significant fan power to move the air from the mechanical yard to the building. Future designs and applications should consider the duct length reduce the duct run from the conditioner to the outdoor air intake as much as possible.

- The demonstration did not treat 100% of the outdoor air, thus limiting the benefit to energy savings from offset cooling. In order to offset the reheat for such an installation, a system should be designed to ensure that the LDAC meets a significant portion of the latent load. Typically, the LDAC can meet 100% of a building's latent load if designed to treat 100% of the outdoor air.

8.1 SUMMER 2010

Many issues came up in the summer of 2010, which led to limited operation during the 2010 cooling season. The Viessmann 200-T solar collector array caused the internal working fluid (Tyfocor HTL glycol) to heat up to greater than the designed limitation (339°F) during stagnation, which caused two main issues for the system: 1) the burning of the glycol renders its antifreeze properties less effective, and makes the fluid acidic; and 2) high-pressure buildup triggers the pressure release valve. These issues could be avoided by using a heat-pipe (Viessmann 300-T) design instead of a direct-flow design because the heat-pipe design has a stagnation temperature of 302°F, which is below the upper limit for the glycol. Also, the glycol has a vapor pressure of 4 bar at 302°F, which is below the release pressure of the relief valve. An alternative solution was used to replace the glycol solution with water, which eliminated the risk of corroding the piping from burned glycol.

It was found that the dead-bands used to limit short cycling of the system for the upper and lower limits of the storage tank were larger than necessary; the dead-bands were decreased accordingly. It was also determined that decoupling the conditioner and regenerator operation allowed for optimal control on the system by allowing each unit to operate independently based on the availability of concentrated desiccant and solar radiation, respectively.

8.2 SUMMER 2011

The LDAC system successfully operated for the majority of the 2011 cooling season (May through September), but several issues arose regarding the operation and performance monitoring. During the start-up period, several maintenance procedures were required; there were pressure leaks in the solar collector array piping and a collector tube required replacement. It was observed that the system was running on lower desiccant concentrations than the design condition, so several drums of concentrated salt solution were ordered to be added to the system. The storage tank volume limits were also lowered in effort to run the system with higher concentrated desiccant, which increased the cooling capacity and efficiency of the system.

Several issues caused the system to operate at lower than expected efficiency during April and May 2011. The pump and fan of the cooling tower were set at a fixed flow and speed, which led to excessive electrical energy use to meet a given cooling load. The cooling tower pump required 1.2 kW of electric power, and the motor had an efficiency of 62%. The water temperature change across the cooling tower was also small (~2°F), which indicated that excessive flow rates were present. A new cooling tower pump was installed, in conjunction with a VFD. A new control sequence was written that maintained the cooling tower fan at 75% fan speed and modulated the cooling tower pump flow rate to maintain a 10°F temperature difference between the cooling tower supply and return water temperatures. This resulted in over a 65% reduction in cooling tower energy use.

The concentration of the liquid desiccant was initially below the desired value. Ideally, the desiccant would be 43% concentrated, which is the upper limit of concentration before crystallization occurs. The liquid level in the storage tank indicates the desiccant concentration; the lower level means the desiccant concentration is at a maximum. The concentration associated with the lower level was initially under the desired maximum concentration, so the level was decreased accordingly (allowing the tank to fall to a lower level, which corresponds to a more concentrated solution). Increasing the concentration of the desiccant solution increased the drying rate of the air and increased the cooling capacity and overall efficiency.

The solar array used for desiccant regeneration was oversized for the system and was designed to allow for excess desiccant to be stored during the day. This regenerator flow rate exceeded the rate at which the conditioner weakened the desiccant (by absorbing water from the air) unless the conditioner was running at near maximum capacity.

Under normal operation, the desiccant in the tank is sufficiently weak such that a full day of regeneration plus operation of the conditioner system should maximize solar utilization. However, in the event the desiccant tank is fully strong and the latent load is small, the solar regeneration system may have to shut off. This occurrence creates stagnation in the solar field. The solar field stagnates in the following manner:

1. The solar water pump shuts off.
2. The fluid internal to the evacuate tubes raises in temperature and turns to steam.
3. The steam displaces the fluid in the system, which is forced into expansion tanks.
4. Continued steam generation is dissipated by a passive heat-dump radiator installed in-line between the collectors and the expansion tank.
5. The fluid pumps are locked out until the system temperature drops well below the boiling point of the fluid in the system to avoid flashing the fluid and creating a pressure spike. The pump is usually locked out until after sunset.

The solar loop was initially directly connected to the regenerator. There was far too little expansion capacity in the system to allow the collectors to stagnate properly. This resulted in an overheating/over-pressurizing event that caused steam and burnt propylene glycol to leak into the regenerator slump the first time the collectors were charged with glycol. As a result, some initial charged desiccant was lost. The final design included a heat exchanger between the solar field and regenerator to fix this issue.

8.3 LESSONS LEARNED

There are a number of lessons that should be learned from this project and applied to future projects in order to ensure successful design, installation, and operation of a solar-powered LDAC system.

8.3.1 Climate

The climate in Tyndall only required dehumidification for a few months per year (3-4 months) and experienced freezing temperatures during the winter. This limited the need for an outside air dehumidification system. Future installations should focus on climates that require year-round cooling and dehumidification, or on facilities that have problems with maintaining interior humidity levels.

8.3.2 Building Type

The LDAC system should be applied to 100% outside air AHUs that are set up to overcool and reheat the supply air. The LDAC should also be sized to meet 100% of the outside air ventilation requirements. Because this system was only sized to meet 33% of the ventilation air, it limited the overall impact the system has on removing the appropriate amount of moisture from the outside air.

8.3.3 Solar Field

In the future all solar field installations should include the use of heat-pipe based, evacuated-tube collectors. The cheaper draw through evacuated-tube collectors used on this project caused the majority of the problems. In addition, the system should include an appropriately sized heat-dump radiators or be connected to a secondary heating system so that the heat can be used when the dehumidification loads are reduced. This will mitigate the stagnation problems and provide a means for using the waste heat.

Solar should be used where electricity and thermal energy is expensive, as with island nations. Otherwise, natural gas should be used.

8.3.4 LDAC Fans and Pumps

The efficiency of the fans and pumps should be carefully considered when designing a system. Thought should also be given to the part-load efficiency of the pumps and fans. In the future all motors should have a minimum 80% electrical efficiency, and all of the pumps and fans should either utilize an electronically commutated motor or VFD to allow for part-load operation and much better part-load electrical efficiencies. This is the new "normal" in the LDAC design. All pumps and fans are on VFDs.

8.3.5 Cooling Tower

A cooling tower water treatment program should be implemented when the cooling tower is installed. In addition the cooling tower should come equipped with high-efficiency motors and VFDs on both the cooling tower fans and pumps.

8.3.6 Sequence of Operation

The sequence of operation should modulate the flow of all of the fans and pumps during part-load operation to increase the part-load electrical efficiency. In addition, the system should be set up to shut off the system when there is not a sufficient need for dehumidification.

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APPENDIX –POINTS OF CONTACT

List all the important points of contact (POC) involved in the demonstration, such as co-investigators, sponsors, industry partners, and regulators. The list should include the following information: (1) full name; (2) complete mailing and FedEx addresses (if different); (3) telephone number, fax number, and e-mail address; and (4) the role of the individual in the project.

Use the tabular format below:

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