Low-Flow Liquid Desiccant Air-Conditioning: Demonstrated Performance and Cost Implications

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Technical Report
NREL/TP-5500-60695
September 2014

Contract No. DE-AC36-08GO28308

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Acknowledgments

This work was made possible with funding from the American Recovery and Reinvestment Act of 2009. The authors would like to thank the U.S. Department of Energy (DOE) Building Technologies Office for its support of the project. This report was prepared by the National Renewable Energy Laboratory (NREL) Center for Buildings and Thermal Systems under Task Number ARCB.1201.

The authors would like to recognize and thank Stevens Institute of Technology and Whole Foods for providing demonstration sites for this project. The authors also recognize the major contributions to the project and to this report from Andy Lowenstein and Jeff Miller of AIL Research, Jeff Halley, and John Ed Masopust of J&J Mechanical, and Joe Ryan (independent). The authors also thank Ron Judkoff, Feitau Kung, and William Livingood of NREL for reviewing this document; and Joe Ryan for processing performance data.

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Executive Summary

Cooling loads must be dramatically reduced when designing net zero energy buildings or other highly efficient facilities. Advances in this area have focused primarily on reducing a building's sensible cooling loads by improving the envelope, integrating properly sized daylighting systems, reducing unwanted solar heat gains, reducing internal heat gains, and specifying cooling equipment with high nominal efficiencies. As sensible loads decrease, however, latent loads remain relatively constant, and thus become a greater fraction of the overall cooling requirement in highly efficient building designs, particularly in humid climates. This shift toward low sensible heat ratio (SHR) systems is a challenge for conventional heating, ventilating, and air-conditioning (HVAC) systems.

Electrically driven vapor compression systems typically dehumidify by first overcooling air below the dew-point temperature and then reheating it to an appropriate supply temperature, which requires additional energy. Another dehumidification strategy incorporates solid desiccant rotors that remove water from air more efficiently than vapor compression; however, these systems are large and increase fan energy consumption due to the increased airside pressure drop of solid desiccant rotors. A third dehumidification strategy involves high-flow liquid desiccant systems. These systems require high-maintenance mist eliminators to protect the air distribution system from corrosive desiccant droplet carryover. These are commonly used in industrial applications but rarely in commercial buildings because of the high maintenance cost.

Low-flow liquid desiccant air-conditioning (LDAC) technology provides an alternative solution with several potential advantages over previous dehumidification systems:

- Eliminates the need for overcooling and reheating associated with vapor compression systems.
- Avoids the increased fan energy associated with solid desiccant systems.
- Allows for more efficient ways to remove the heat of sorption than is possible in solid desiccant systems.
- Reduces the amount of liquid desiccant needed compared to high-flow LDAC systems.
- Is smaller and allows more flexible configurations than solid desiccant systems.
- Reduces the desiccant droplet carryover problem, thereby reducing maintenance requirements compared to high-flow LDAC systems.

Liquid desiccant systems have also been suggested as a way to shift latent loads to times when energy is cheaper and/or renewable or waste energy is abundant. Latent load shifting can be accomplished with liquid desiccant, which is a relatively inexpensive form of storage. Liquid desiccants can also be used for removing biological and chemical pollutants from process airstreams thereby improving indoor air quality.

In an effort to better understand the potential of this promising new technology in the marketplace, DOE enlisted NREL to assess the performance of low-flow LDAC technology in several real buildings. To accomplish this, NREL worked with the LDAC manufacturer and facility managers to field test LDAC in classes of buildings with good potential for energy savings. Four LDAC systems were installed and monitored on three building types (two grocery

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stores, a pool facility, and a multipurpose campus building) in three U.S. climate zones to observe mechanical performance and analyze the strengths and weaknesses of the systems (see Table ES–1). In addition, one system used waste heat and one system used solar thermal energy for desiccant regeneration.

Table ES–1 Demonstration Facility Summary

Facility	Building Type	Location	Climate Zone
Whole Foods Market	Supermarket	Encinitas, California	3B
Whole Foods Market	Supermarket	Kailua, Hawaii	1A
Schaeffer Pool	Indoor pool facility	Hoboken, New Jersey	4A
Babbio Center	Multipurpose campus building	Hoboken, New Jersey	4A

The first three systems in Table ES-1 were monitored for several months. Installation of the LDAC in the Babbio Center was delayed several times and no performance monitoring was completed. The measured performance in the first three installations was near expectations. The LDAC technology proved capable of providing a large latent capacity and low dew-point air , which is required to provide comfortable and desirable space conditions for supermarkets and the challenging environment of a natatorium. Indoor conditions at Whole Foods, Encinitas were consistently maintained within acceptable humidity levels (between 35% and 55% RH) without using overcool-and-reheat strategies. Humidity levels at Whole Foods, Kailua were kept between 50% and 75% RH, owing mainly to the effects of large unanticipated infiltration rates due to entry and loading dock doors being kept open during store operation. Keeping the doors open is a cultural response to a warm humid environment where air velocity across the human body can be an effective passive comfort strategy; however, this is counterproductive from an energy and comfort perspective in a grocery store. The natatorium humidity was maintained within 10% of the ideal condition of 60% RH for 90% of the time. Humidity control was accomplished with a regeneration-specific heat input (RSHI) near the expected value of 1.5 kBtu per pound of water removed for Whole Foods, Kailua and the Schaeffer Pool. The RSHI was higher than expected at Whole Foods, Encinitas because of a suspected water leak in the system. The average electrical consumption at all demonstrations was very low, at 0.32–0.45 kW/ton.

There were periods when some of the LDAC systems were not fully functional because of mechanical issues. Observations during these periods were quite useful in showing how LDAC systems and installations can be further improved, and prepared for mass market readiness. Key lessons from the demonstrations were:

(1) The Encinitas grocery store and the Schaeffer natatorium both experienced precipitate formation in the desiccant resulting from air contaminants. For Encinitas the scavenging air intake for the regenerator was located at a loading dock with heavy diesel exhaust. A carbon filter added to this airstream, which solved the problem. For the natatorium, the reason is less clear and needs to be understood before widespread use of LDAC in this application. The precipitate problem is also an indicator of the potential for liquid desiccant to operate as an air cleaning agent for both biological and chemical contaminants, thus potentially adding to the value proposition for LDAC.

(2) The natatorium LDAC system design integrated a CHP system to utilize waste heat for desiccant regeneration. The CHP system did not always deliver high enough water temperatures for optimal performance indicating the need for more careful system design.

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(3) The Whole Foods grocery in Kailua was operated in a manner atypical for grocery stores on the U.S. mainland. The main entrance doors and the doors to the loading dock were kept open creating a strong cross ventilation airflow that created a large latent load. The LDAC system was not sized to accommodate such a large unanticipated load and indoor relative humidity drifted up to as high as 75%. It is important to understand any special operational conditions that will increase latent load when sizing an LDAC. This is especially critical in supermarkets where sufficiently dry air enables refrigeration energy savings.

Operation of the LDAC system at the Babbio Center was delayed, beyond the time frame for this report due to installation problems, showing the need for proper training of installers.

We performed energy modeling to provide estimates of the savings available with LDAC in supermarket applications across the United States; the source energy savings are shown in Figure ES-1 and Figure ES-2 for a single-stage and two-stage regenerator, respectively. The baseline models included four reheat options: 1) natural gas reheat coils, 2) RTU condenser hot-gas reheat with auxiliary natural gas reheat, 3) electric reheat coils, and 4) RTU condenser hot-gas reheat with auxiliary electric reheat. For a supermarket in a climate with high latent loads requiring 4,000 cfm of ventilation, we calculated energy cost savings ranging between $3,000 and $30,000 in the hot humid climate zones of 1A and 2A, with corresponding source energy savings between 1% and 6% of the supermarket's whole building energy expenditure. For climate zones 1A–2A, the estimated space conditioning source energy savings in grocery stores are 12%–40% for the four reheat strategies modeled.

Space conditioning savings are realized because the large expenditure for overcooling and reheat in a DX system was eliminated. Supermarkets in mixed-humid climates (3A and 4A) are projected to show savings of around 1% to 4% of building source energy, and utility cost savings between $200 and $13,000. Cold-humid climates and marine climates are expected to show minimal differences in energy use, although some cost savings may be possible due to the shifting of energy consumption from electricity to gas where RTU condenser hot-gas reheat and/or electric reheat coils were used.

Additional savings can be achieved with the use of a two-stage regenerator, which is estimated to save 40% of the thermal energy required for regeneration. With a two-stage regenerator, total building source energy savings are estimated to be between 4% and 8% in hot humid climate zones, with corresponding annual energy cost savings between $10,000 and $36,000. HVAC savings in hot humid climates range from 34%-57%. We chose to model the LDAC conservatively by not accounting for savings from improved control strategies and waste heat integration. Figures ES-1 and ES-2 show that the largest source energy end use is natural gas for regeneration. Significant energy savings can be achieved with the LDAC system when there is a waste or free heat source at the appropriate temperature. Additional savings may also be achieved if high efficiency lighting is used, which reduces the SHR and increases the need for efficient dehumidification. These saving will be greatest in climate zone 1A, which is dominated by cooling loads.

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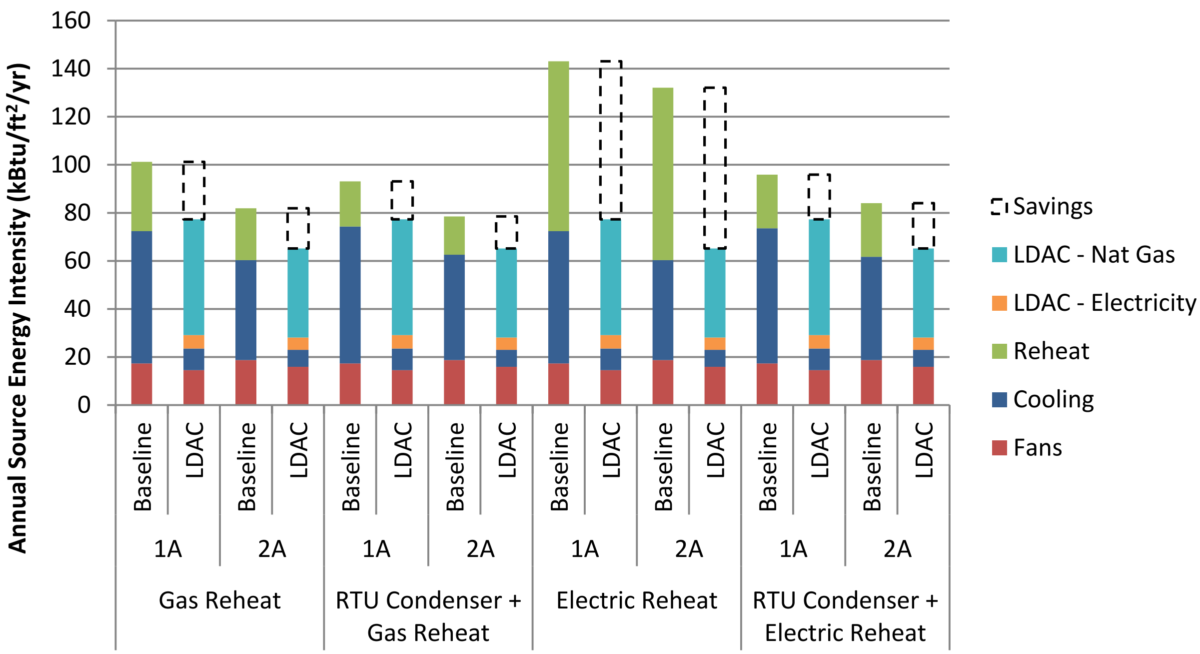

Figure ES–1 Annual source end use energy consumption and savings – single-stage regenerator (kBtu/ft^2/yr)
(Credit: Lesley Herrmann/NREL)

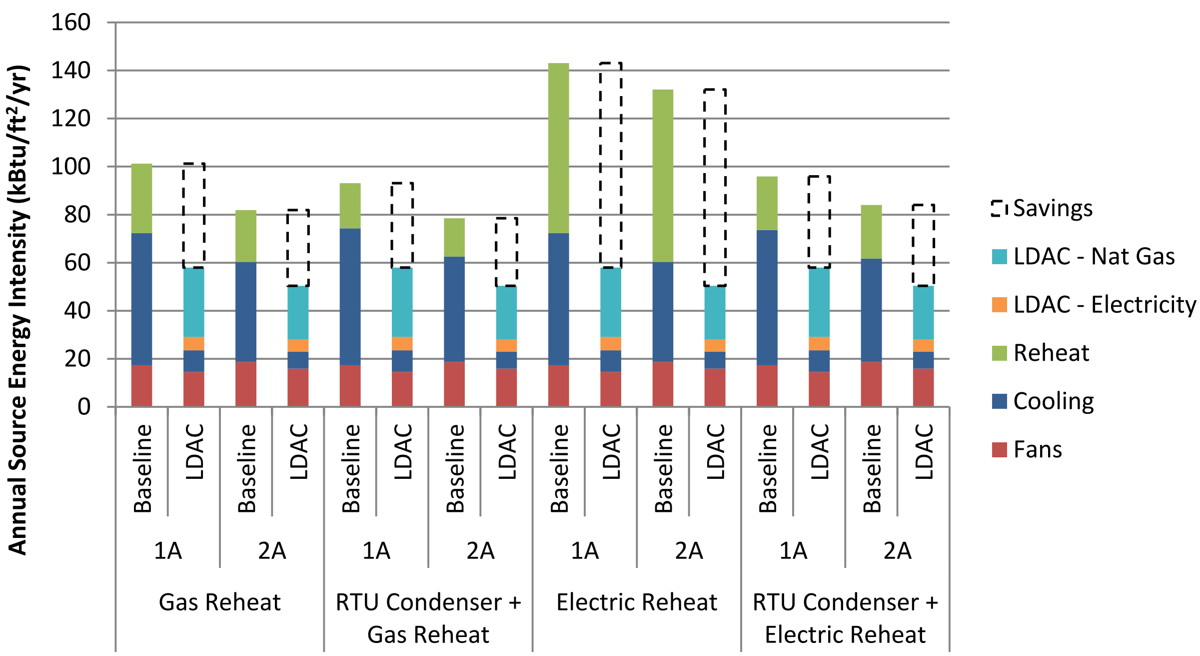

Figure ES–2 Annual source end use energy consumption and savings – two-stage regenerator (kBtu/ft^2/yr)
(Credit: Lesley Herrmann/NREL)

Low-flow LDAC is a rapidly evolving emerging technology and is not yet mature enough to allow a detailed economic analysis. However, the incremental cost of the LDAC was determined

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based on a 3-year, 5-year, and 10-year simple payback period based on the current performance levels. Table ES–2 and Table ES–3 list the incremental costs of the LDAC with a single-stage regenerator for the baseline case using RTU condenser hot-gas and auxiliary natural gas reheat. This case results in the lowest annual energy cost savings so the incremental costs listed are the most conservative of the four cases. Appendix B of this report lists the incremental costs for each of the four baseline reheat strategies.

Table ES–2 LDAC Incremental Cost – RTU Condenser Hot-Gas and Natural Gas Reheat Coils – Single-Stage Regenerator ($)

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	8,522	9,652	657	(10,515)	11,215	4,569	1,995
5-year	14,204	16,086	1,095	(17,524)	18,691	7,615	3,325
10-year	28,408	32,172	2,191	(35,049)	37,382	15,229	6,649

Table ES–3 LDAC Incremental Cost – RTU Condenser Hot-Gas and Natural Gas Reheat Coils – Two-Stage Regenerator ($)

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	29,146	25,587	11,562	702	19,523	11,453	8,968
5-year	48,577	42,644	19,271	1,170	32,539	19,088	14,947
10-year	97,153	85,289	38,541	2,341	65,078	38,177	29,894

LDAC improvements are currently under development by industry to improve energy efficiency and reliability. These include: (1) two-stage regeneration, which improves LDAC regenerator efficiency by about 40%; (2) wicking fin design, which improves efficiency, simplifies the design, and solves leak problems with the current LDAC element design; (3) membrane based LDAC unit, which completely eliminates carryover; (4) improved LDAC control strategies; (5) better integration of alternative heat sources, which saves regenerator energy; and (6) integration of heat pumps with LDAC systems, enabling all-electric systems. These hardware improvements could benefit from research in the labs to better characterize the thermodynamics and model and optimize the system designs and building interactions. LDAC technology has promise as an effective means to save energy in applications where humidity control is essential and energy intensive; however, further development is needed for increased energy savings and improved reliability.

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Acronyms and Abbreviations

AHU	air handling unit
ASH	anti-sweat heater
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
$CaCl_2$	calcium chloride
cfm	cubic feet per minute
CHP	combined heat and power
COP	coefficient of performance
DB	dry bulb temperature
DOE	U.S. Department of Energy
DP	dew point temperature
DX	direct expansion
hp	horsepower
HR	humidity ratio
HVAC	heating, ventilation, and air conditioning
LDAC	liquid desiccant air-conditioning
LHR	latent heat ratio
LiCl	lithium chloride
MCBD	mean coincident dry bulb temperature
MRC	moisture removal capacity
NREL	National Renewable Energy Laboratory
OA	outdoor air
RH	relative humidity
RSHI	regeneration specific heat input
SHR	sensible heat ratio
TMY3	Typical Meteorological Year 3

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Nomenclature

Product or process airstream	Air that leaves the conditioner and will eventually be introduced into the building as supply air. The product air may go through other equipment or processes before it is introduced into the conditioned space.
Regeneration specific heat input	The amount of thermal energy consumed by a desiccant regenerator to remove one pound of moisture from the air, in kBtu/lb (ASHRAE 2007a). The typical range of RSHI values for single stage liquid desiccant regenerators are between 1.25 and 2.1 kBtu/lb. Two stage regenerators can achieve RSHI values as low as 0.9 kBtu/lb. The RSHI does not include the energy from the regenerator's pump(s) and fan.
Gas RSHI	The amount of fuel energy consumed by the gas boiler to remove 1 lb of moisture from the air, in kBtu/lb. The difference between the gas RSHI and the thermal RSHI is a result of the gas boiler efficiency.
Moisture removal capacity	The amount of moisture the LDAC removes from the air per hour in lb/hr; a function of system size, design supply conditions, and outside air conditions.
MRC cost	The utility cost (gas and/or electricity) to remove 1 lb of moisture from the air, in $/lb.
Latent heat ratio	The fraction of the total space air conditioning load associated with dehumidifying the air (i.e. latent cooling). LRH + SHR = 1.
Suction group	A set of two or more compressor racks in a product refrigeration loop.
LDAC latent COP	The ratio of latent cooling (or dehumidification) provided by the LDAC to the thermal energy consumed during the regeneration process.

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1 Introduction

1.1 Liquid Desiccant Air-Conditioning Technology

Cooling loads must be dramatically reduced when designing net zero energy buildings or other highly efficient facilities. Advances in this area have focused primarily on reducing a building's sensible cooling loads by improving the envelope, integrating properly sized daylighting systems, reducing unwanted solar heat gains, reducing internal heat gains, and specifying cooling equipment with high nominal efficiencies. As sensible loads decrease, however, latent loads remain relatively constant, and thus become a greater fraction of the overall cooling requirement in highly efficient building designs, particularly in humid climates. This shift toward high latent heat ratio (LHR) is a challenge for conventional heating, ventilation, and air-conditioning (HVAC) systems. Conventional electrically driven vapor compression systems typically dehumidify by first overcooling air below the dew-point temperature and then reheating it to an appropriate supply temperature, which requires additional energy. Another dehumidification strategy incorporates actively or passively regenerated solid desiccant rotors into a system to enhance the removal of water from the air; however, these systems can be large and increase fan energy consumption due to the increased airside pressure drop of the rotors. A third dehumidification strategy involves high-flow liquid desiccant systems. These systems require high-maintenance mist eliminators to protect the air distribution system from corrosive desiccant droplet carryover. These are commonly used in industrial applications but rarely in commercial buildings because of high maintenance cost.

Low-flow liquid desiccant air-conditioning (LDAC) technology avoids these carry over problems while still providing the dehumidification potential of the high-flow liquid desiccant systems. LDAC is a relatively new technology with several approaches being developed, but the essential feature of all designs is that it can deeply dry air before it enters the main cooling system without the need for overcooling and reheating. LDAC technology provides an alternative solution for dehumidification with several advantages over previous systems, as it:

- Eliminates the need for overcooling and reheating associated with vapor compression systems
- Avoids the increased fan energy associated with solid desiccant systems
- Allows for more efficient ways to remove the heat of sorption than is possible in solid desiccant systems
- Can be smaller than solid desiccant systems, and is more flexible in the configuration of ducts and system components because supply and exhaust ducts must be adjacent to each other at the point where a desiccant wheel is installed; for example, the LDAC conditioner and regenerator can be configured as a split system, whereas the solid desiccant system cannot
- Reduces the amount of liquid desiccant needed compared to high-flow LDAC systems
- Reduces or eliminates the desiccant carryover problem, thereby reducing maintenance requirements compared to high-flow LDAC systems

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- Consumes less energy per unit of water removed from the ventilation airstream compared to other systems in low-sensible heat ratio (SHR) situations where low interior humidity is required
- Can reduce peak electricity demand compared to vapor compression systems if thermal energy sources such as natural gas, solar thermal energy, and waste heat are used for regenerating the desiccant
- Can shift loads by using relatively inexpensive desiccant storage to delay regeneration until times when thermal energy is readily available and cheaper
- Reduces other energy loads through integrated design; for example:
  - In grocery stores, lowering humidity levels with LDAC can also reduce loads on: (1) refrigeration system compressors; (2) defrost heaters; and (3) anti-sweat heaters (ASHs) on display case doors
  - In swimming pools, using the heat of absorption to warm the pool water while using the pool water to remove the heat of absorption.

LDAC systems may also lead to additional benefits, including:

- The ability to optimize temperature and humidity to increase worker comfort and productivity (Abdou et al. n.d.; LBNL 2013)
- A competitive marketing and sales advantage for stores with comfortable indoor conditions
- Avoided refurbishment and maintenance costs related to problems created by high indoor humidity, such as mold and mildew
- Improved product shelf life from improved humidity control
- Improved sales through the reduction of frost or condensation, which obscures views of products through doors on refrigerated display cases
- Greater likelihood that outdoor air requirements will be met during operation; by contrast, operators of traditional systems may override ventilation controls to address humidity issues or reduce energy costs
- Removal of air pollutants, thus improving indoor air quality
- Secondary benefits from reduced peak demand, such as improved energy security and reduced air pollution and water consumption from grid-supplied power.

LDAC technology is designed to deeply dry air before it enters the main cooling system. In addition to eliminating the need for overcooling and reheating, LDAC technology may provide additional energy savings by allowing the main cooling system's evaporator temperature to be set higher, which lowers the cooling load and associated energy consumption. It should be possible to downsize the sensible cooling system for new buildings and major HVAC retrofits. Additional energy savings are possible in grocery stores, where maintaining dry indoor conditions can save HVAC energy and reduce the load on the refrigeration system evaporator coils, display case and open freezer defrost systems, and ASHs on display case doors. As an

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example, reducing the indoor relative humidity (RH) from 55% to 35% was shown in one study to reduce the latent load and compressor power demand of open vertical dairy cases by 74% and 19.6%, respectively; the RH reduction also reduced defrost duration by 40% (Faramarzi et al. 2000). Another benefit is that store managers may be more willing to install refrigerated case doors because product view will remain unobscured by fog or frost. Case doors reduce the load on the food refrigeration systems.

Figure 1–1 shows three of the main components of the LDAC system: the conditioner, the regenerator, and the interchange heat exchanger are shown in the top portion of the figure. A cross-section of the flocked plates and air gap is shown in the bottom portion of the figure. Figure 1–2 shows the inner workings of the LDAC's conditioner (the regenerator has a similar configuration).

The system cools the air via the following steps:

1. Hot-humid outdoor air (OA) (process air) enters the conditioner and flows past the film of liquid desiccant flowing down the flocked external surfaces of each plate. The plates of the system are configured as a water-cooled (internal to each plate) parallel-plate heat exchanger.
2. The air is dried as water vapor from the air is absorbed into the desiccant. The diluted desiccant is then pumped to the regenerator. The now dry air is further cooled by a standard vapor-compression evaporator coil or chilled water coil if needed and then supplied to the space.
3. Scavenging air (usually OA) enters the regenerator and contacts the diluted desiccant flowing down the plates (much like the conditioner working in reverse). The plates and the desiccant are heated by hot fluid (water or glycol) flowing in the plates to help the water desorb from the desiccant.
4. The scavenging air picks up the desorbed moisture and is exhausted to ambient.

The thermal energy required for regeneration can be provided by fossil fuel boilers; solar thermal collectors; or heat recovered from reciprocating engine generators, microturbines, turbines, fuel cells, or other processes with recoverable heat at 150°–210°F (Lowenstein et al. 2006). The desiccant used in LDAC systems is most often lithium chloride (LiCl). In some cases calcium chloride ($CaCl_2$) is used because it is significantly cheaper, which is especially advantageous in applications where more than about 60 minutes of desiccant storage is needed; however, $CaCl_2$ cannot dry air as deeply as LiCl.

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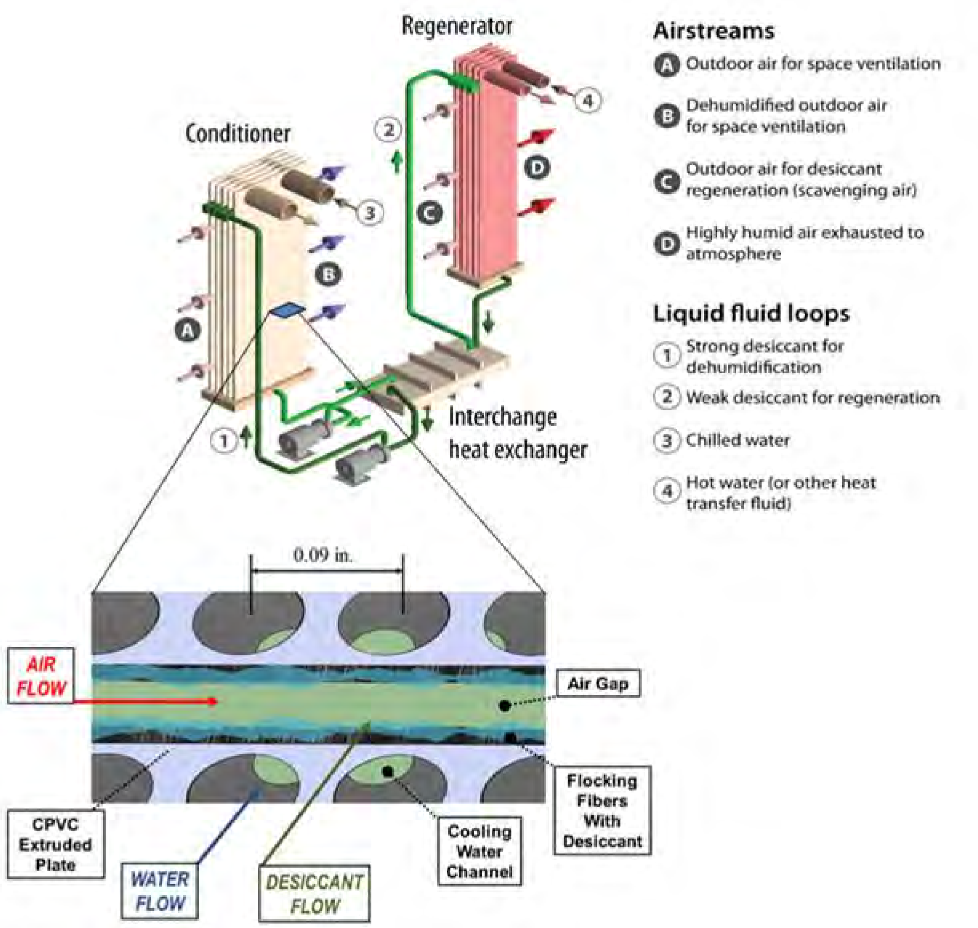

Figure 1–1 Core components of a low-flow LDAC
(Credit: Adapted from Lowenstein et al. 2006, adapted and used with permission)

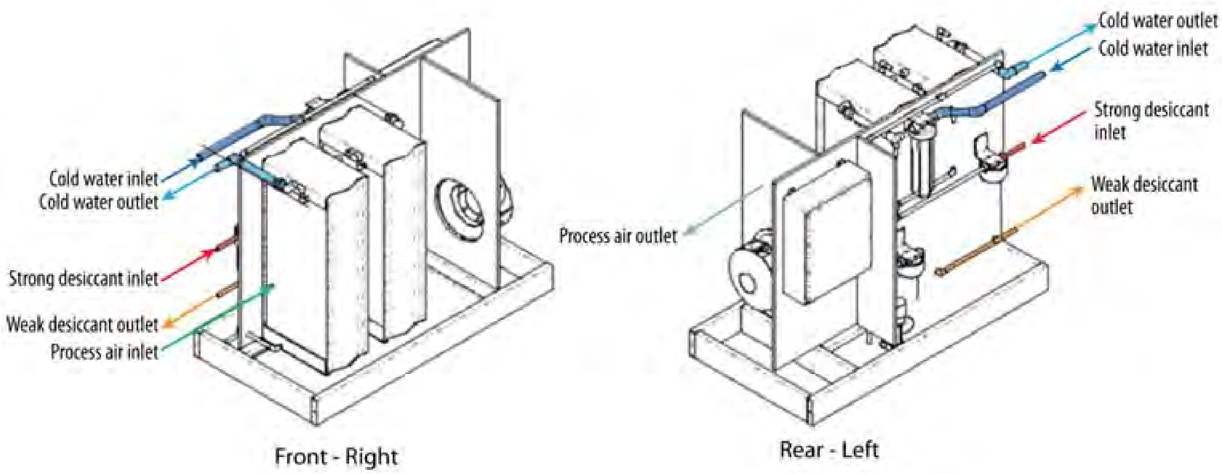

Figure 1–2 Cutaway view of LDAC conditioner (regenerator not shown)
(Credit: Andy Lowenstein, AIL Research, with permission)

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1.2 Purpose

In an effort to better understand the market potential of this promising new technology, the U.S. Department of Energy (DOE) enlisted the National Renewable Energy Laboratory (NREL) to assess the performance of low-flow LDAC technology in several real buildings. To accomplish this, NREL worked with the LDAC manufacturer and facility managers to field test the uses of LDAC in building types with good potential for energy savings. Some of the integrated system designs included waste heat and solar thermal energy for desiccant regeneration. Four LDAC systems were installed and monitored on various building types (two grocery stores, a pool facility, and a multipurpose campus building) in three U.S. climate zones to observe mechanical performance and analyze the strengths and weaknesses of the systems (see Table 1–1).

Table 1–1 Demonstration Facility Summary

Facility	Building Type	Location	Climate Zone
Whole Foods Market	Supermarket	Encinitas, California	3B
Whole Foods Market	Supermarket	Kailua, Hawaii	1A
Schaeffer Pool	Indoor pool facility	Hoboken, New Jersey	4A
Babbio Center	Multipurpose campus building	Hoboken, New Jersey	4A

The effects of the LDAC on several variables were analyzed to understand the performance of each LDAC system, including space conditions (temperature and RH), LDAC energy consumption (electricity and natural gas), and refrigeration energy consumption. Commonly used dehumidification metrics used to quantify LDAC performance include:

- **Regeneration specific heat input (RSHI):** A metric used to quantify the amount of thermal energy consumed by a desiccant regenerator to remove one pound of moisture from the air, in kBtu/lb (ASHRAE 2007a). The RSHI does not include the energy from the regenerator's pump(s) and fan. For liquid desiccants, the theoretical minimum RSHI value is the specific heat of vaporization of water (at the regeneration temperature) plus the specific heat of dilution of the desiccant. The specific heat of dilution is the energy required to disassociate the water from the desiccant solution due to intermolecular attractions (this does not include the energy to change the water's phase from liquid to vapor). The specific heat of dilution increases with higher desiccant concentration, thus increasing the amount of heat required to regenerate it. For regenerating lithium chloride solution with final concentration in the range of 25% to 43% while using a 185°F heat source in a single stage regenerator, the theoretical minimum RSHI values are between 1.00 and 1.12 kBtu/lb. Approaching the theoretical minimum requires regenerators with high effectiveness heat and mass exchangers. The typical range of RSHI values for single stage liquid desiccant regenerators are between 1.25 and 2.1 kBtu/lb. Two stage regenerators can achieve RSHI values as low as 0.9 kBtu/lb.
- **Gas RSHI:** The amount of fuel energy consumed by the gas boiler to remove 1 lb of moisture from the air, in kBtu/lb. The difference between the gas RSHI and the thermal RSHI is a result of the gas boiler efficiency.
- **Moisture removal capacity (MRC):** The amount of moisture the LDAC removes from the air per hour in lb/h. This metric is a function of system size, design supply conditions, and outside air conditions.

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- **MRC cost:** The utility cost (gas and/or electricity) to remove 1 lb of moisture from the air, in $/lb.

Sections 2.1 through 2.4 provide details on each demonstration building including a description of their LDAC systems, and a discussion of the results of each demonstration. Because these demonstrations occurred over an extended period (September 2012 through August 2013), the performance analysis for each system focuses on particular time periods that highlight optimal performance of the LDAC system. As with any field project demonstrating the use of a new technology, these systems periodically experienced issues that prevented them from operating at full design capacity. Some issues, such as faulty valves or pumps, were minor; others were significant. Observations during these periods were quite useful in showing how LDAC systems and installations can be further improved. These issues and the associated lessons-learned will be discussed in Sections 2.1.4 and 2.3.4.

Energy modeling was used to assess the energy and utility cost savings implications of LDAC technology in a typical supermarket application in seven locations across the United States. The supermarket building type was chosen for analysis because of its broad applicability nationwide and the potential for relatively large energy savings. We used the DOE reference supermarket building model (DOE-2.1c) as the starting point for the simulations. Modeling results are useful in that they provide insight into, which climates are most appropriate for this technology. The energy modeling analysis is presented in Section 4.

In addition to presenting the result of the field demonstrations and energy modeling exercise, this report also serves as a technical resource document for a second report, *Design Guidance and Site Considerations for Low-Flow Liquid Desiccant Air-Conditioning Technology* (NREL 2014).

This report is available at no cost from the National Renewable Energy Laboratory (NREL) at www.nrel.gov/publications.

2 Site Descriptions and Demonstration Results

2.1 Whole Foods – Encinitas, California

2.1.1 Building and Climate Description

The Whole Foods Market in Encinitas, California, is a 25,000-ft^2 store and is the anchor tenant in the Pacific Station Center, a retrofitted commercial/residential complex (see Figure 2–1). It is approximately ¼ mile from the Pacific Ocean and about 20 miles north of downtown San Diego. This high-humidity microclimate is fairly atypical along the California coast. The store opened in July 2011.

Figure 2–3 Whole Foods Market, Encinitas, California
(Courtesy of Whole Foods Market. "Whole Foods Market" is a registered trademark of Whole Foods Market IP, L.P.)

The store is located on the ground level of a luxury condominium complex; three levels of condominiums are located above the sales floor. This layout created challenges for traditional packaged HVAC equipment (typically located outside) related to aesthetics, insufficient structural framing, and noise restrictions. The condominiums eliminated space for attic- or roof-mounted equipment and created low floor-to-ceiling heights throughout the store. Thus, typical packaged rooftop units (RTUs) with electric direct expansion (DX) coils were not viable. An acceptable option was to use four-pipe fan coil units for sensible heating and cooling in combination with the LDAC to meet the dehumidification requirements. The fan coils are controlled by the building management system and operate on chilled water from an air-cooled chiller and hot water from a boiler. One environmental benefit of using an HVAC system with chilled water distribution is that it generally requires less refrigerant than one with DX coils. The specifications of the installed HVAC system are outlined in Table 2–1.

The mechanical refrigeration system is composed of three suction groups, each of which corresponds to a collection of refrigerated cases that are maintained at the same temperature set point; all three suction groups are served by one air-cooled condenser (see Table 2–1 for specifications).

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Table 2–2 Whole Foods, Encinitas, California—HVAC and Refrigeration System Description

Component	Specification
Total refrigeration system power	90.0 hp
Refrigeration system condenser temperature	100°F
Refrigeration system suction group temperature; power	–25°F; 32.5 horsepower (hp) +15°F; 29.0 hp +35°F; 28.5 hp
HVAC system chiller capacity	50 tons
HVAC system gas-fired boiler capacity	600 kBtu/h
Total OA flow for space ventilation	3,000 cfm

The proximity of the store to the ocean and the warm ambient air conditions create a demand for dehumidification to manage the energy consumption of the HVAC and refrigeration systems. The selected reference point corresponds to the most likely product air conditions from the LDAC. The LDAC is designed to deliver 45°F or lower DP air to maintain higher delta-enthalpy, which maximizes system efficiency and minimizes operation cost. This analysis is an example of how an LDAC would treat OA to dehumidify the space. Treatment of indoor air requires a separate analysis, which has not been identified here. Figure 2–2 shows the hourly Typical Meteorological Year 3 (TMY3) weather data plotted on the psychrometric chart for Long Beach, California, which is in close proximity to Encinitas but was determined to be less humid. Figure 2–3 gives another visualization of the humidity load distribution in this climate: a grouping of the amount of time (in days) the outdoor conditions are fall in a certain humidity ratio (HR) bin (in $lb_{water}/lb_{dry\ air}$). The integral of this graph can give a single index (i.e., lb/lb days), which is roughly proportional to the dehumidification needs in a certain climate (used in Table 2–2).

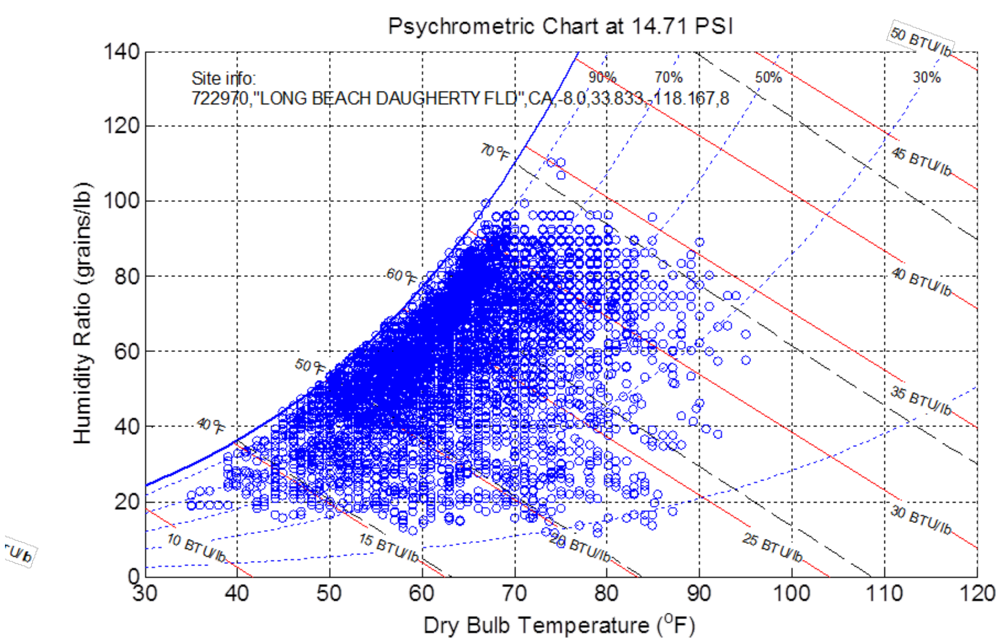

Figure 2–4 Climate analysis, Long Beach, California
(Credit: Eric Kozubal/NREL)

The climate in this area is mild and humid, with a fairly high humidity level throughout most of the year. The monthly average RH ranges from 63% to 74%; the RH is 65% or greater for 75%

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of the year and 80% or greater for 45% of the year. The average HR during the summer months (May through September) is 75 gr/lb (0.0107 lb/lb). Although most hours are below 28 gr/lb (0.004 lb/lb) (see Figure 2–3), the psychrometric chart shows that many hours are warmer and more humid than the reference conditions of 75°F dry bulb temperature (DB) and 45°F DP. This simple analysis shows that the grocery store would benefit from humidity control and that ventilation air should not be supplied without some form of dehumidification. With these reference conditions, the LDAC is estimated to operate for 7,213 hours/year (see Table 2–2).

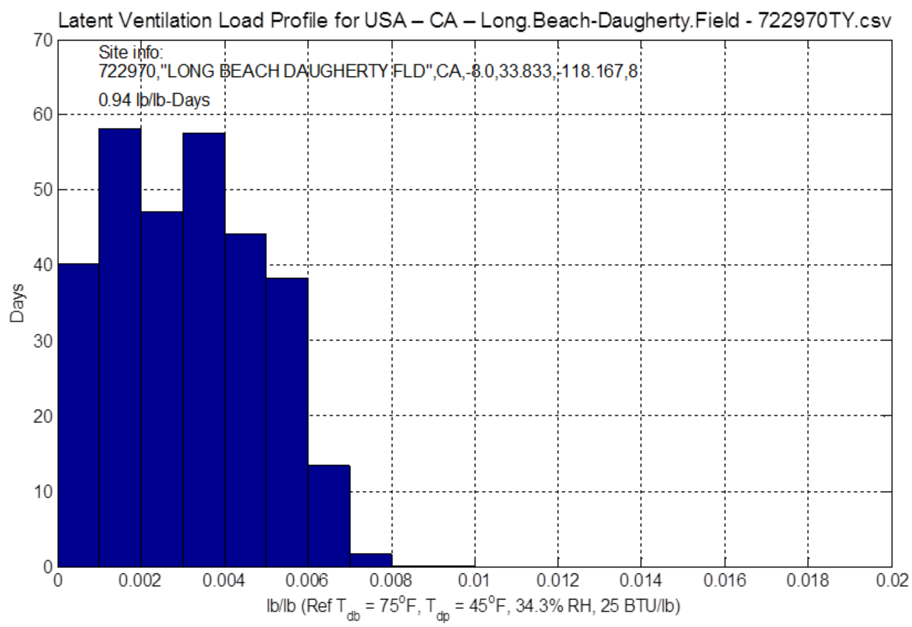

Figure 2–5 Pound-per-pound days, Long Beach, California
(Credit: Eric Kozubal/NREL)

Table 2–3 Total Annual Dehumidification Loads—Long Beach, California (75°F DB, 45°F DP reference)

Specification	Load
Total ventilation load (Btu/lb-days)	1,359
Total moisture load (lb/lb-days)	0.94
Estimated hours of operation (h)	7,213
ASHRAE 1% design conditions (DP; HR; mean coincident dry bulb temperature (MCDB)	66°F; 97.5 gr/lb; 72.6°F

2.1.2 LDAC Design

The LDAC and air-cooled chiller are located adjacent to the loading dock (see Figure 2–4). Because the store has limited space for mechanical systems, the LDAC processes 100% of the ventilation requirement (i.e., the other fan coil units condition 100% recirculation air). The dehumidified OA from the LDAC is ducted through the exterior wall and backroom and is then mixed with the return air before passing through four fan coil units above the open dairy/deli cases (see Figure 2–5). Introducing the dry air into the space in the refrigerator/freezer section provided the biggest opportunity for defrost and ASH energy savings. A bypass damper was also included to provide fresh air when the ambient air does not require dehumidification from the

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LDAC. The LDAC uses a dedicated chilled water circuit served by the air-cooled chiller as the heat sink for the LDAC conditioner. Generally, a cooling tower would have been used in place of the air-cooled chiller, as it is a more efficient system, but this option was not viable because roof space was not available. Because the chiller serves the fan coil units as well as the LDAC, its capacity could not be downsized in the way it could if the LDAC used a cooling tower. The desiccant is regenerated with a gas-fired boiler. The system is controlled based on store conditions via a thermostat and humidistat located on a pillar near the dairy/deli cases (see Figure 2–5). The design specifications of the LDAC are listed in

Table 2–3 and the installed cost of the system is provided in Table 2–4. The chiller also served the building's fan coil units, so the cost listed is a percentage of the chiller required for LDAC operation.

Figure 2–6 LDAC and chiller system in loading dock at Whole Foods, Encinitas, California
(Credit: Jeff Miller/AIL Research, with permission.)

Table 2–4 Whole Foods Market, Encinitas, California—LDAC Description

Specification	Design
Latent cooling capacity (estimated)	18 tons
OA flow rate for space ventilation	4000 cfm
Air RH delivered by LDAC (projected)	20%–25%
Liquid desiccant concentration	~40% LiCl
LDAC latent COP	0.65

Table 2–5 Whole Foods Market, Encinitas, California—LDAC System Installed Cost

Component	Cost ($)
LDAC	75,000
Chiller (including pump, valves, etc.)	18,500
Boiler (including pump, expansion tank, etc.)	9,500
Outbound freight cost	500
Installation (labor)	76,000
Total cost	161,000

The refrigeration system was not downsized with the installation of the LDAC because the LDAC manufacturer wanted to ensure that the store was not reliant on this experimental equipment that would likely have downtimes. Energy savings should be recognized automatically from shorter cycle times resulting from drier air. Currently, the defrost cycles are programmed to operate on daily timed schedules; however, the intention is to reprogram this system according to evaporator coil conditions after the LDAC has shown a significant period of continuous operation.

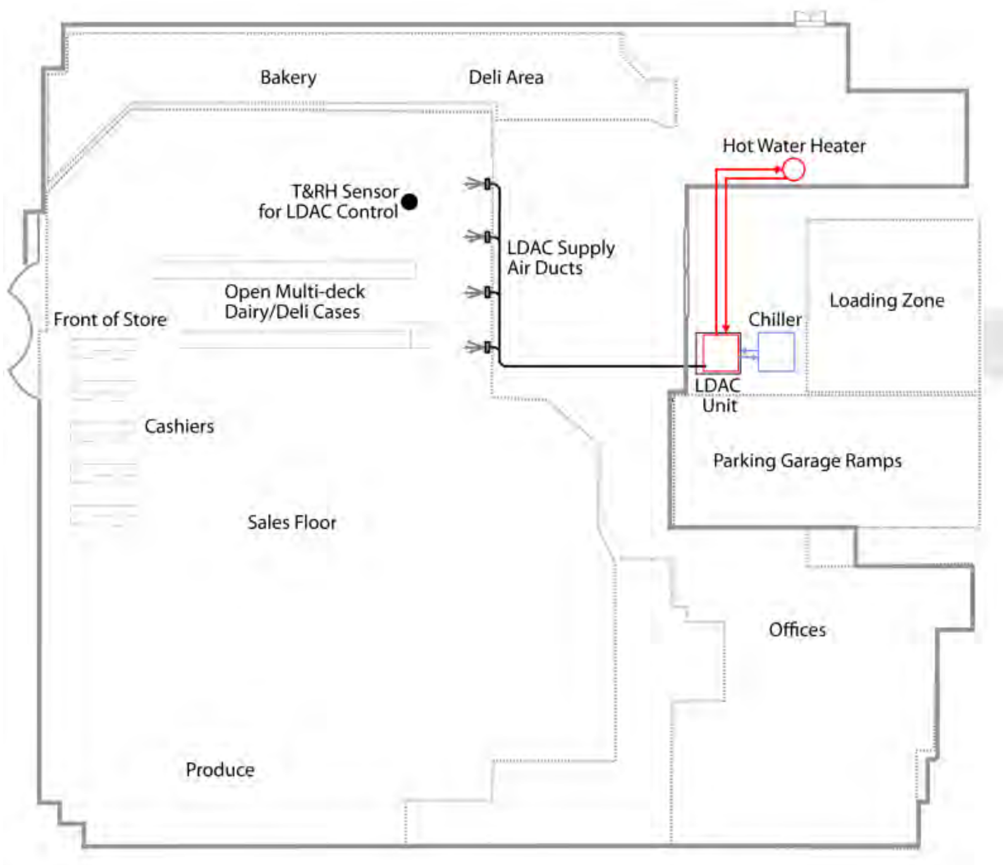

Figure 2–7 Building layout at Whole Foods, Encinitas, California
(Credit: NREL, adapted from Whole Foods Market, with permission)

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2.1.3 Performance Results

The period of performance highlighted here includes operation between August 21 and October 10, 2012 during store hours (7:00 a.m. to 9:00 p.m.). The LDAC consistently delivered air at 26–47 gr/lb when the outdoor conditions were 70–100 gr/lb (about 70% of the time) (see Figure 2–6 through Figure 2–8). The delivered DB was kept at 68.7°–74.1°F. The delivered air humidity was influenced by the OA humidity, as seen in Figure 2–8. In aggregate, the LDAC was able to maintain an indoor air humidity that was 37 gr/lb lower than outdoor conditions (see Table 2–5), which equates to an average indoor DP of 48°F (37% RH at 75°F DB). The LDAC was not always able to deliver air at 30% RH as anticipated by the manufacturer, which led to some periods of indoor air humidity that were higher than expected. However, the LDAC was always able to deliver air at less than 45 gr/lb (33% RH at 75°F DB) (Figure 2–8). Regeneration energy, as quantified by RSHI, is greater than expected because of a leak in the regenerator. This has prompted industry to develop the metallic wicking fin design that should resolve the leakage problems.

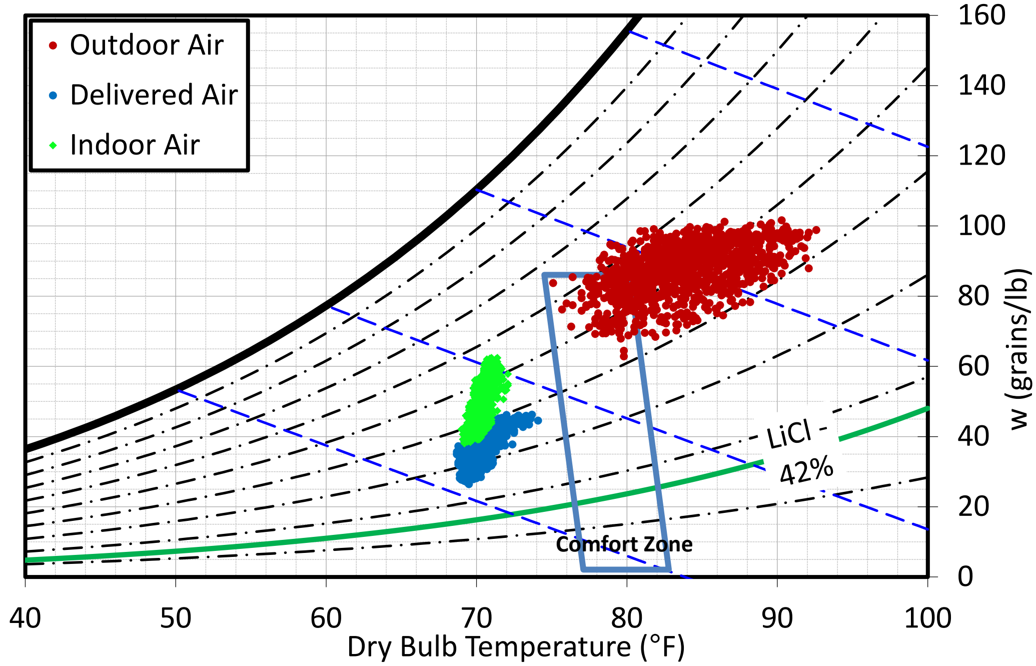

Figure 2–8 Psychrometric chart showing outdoor and delivered air from the LDAC, Encinitas, California, August–October 2012
(Credit: Eric Kozubal/NREL)

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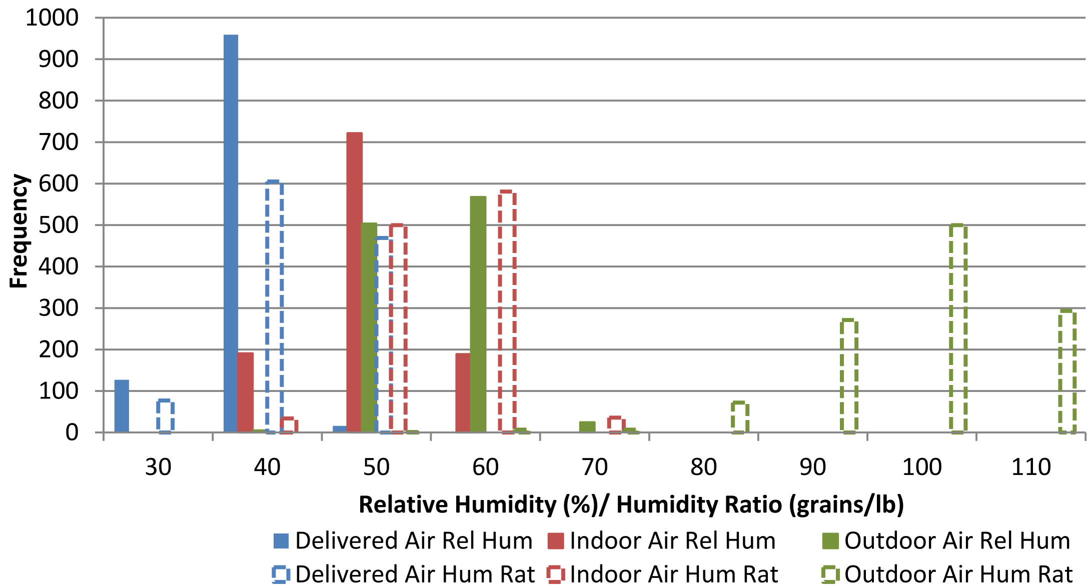

Figure 2–9 Histogram of humidity, Encinitas, California, August–October 2012
(Credit: Lesley Herrmann/NREL)

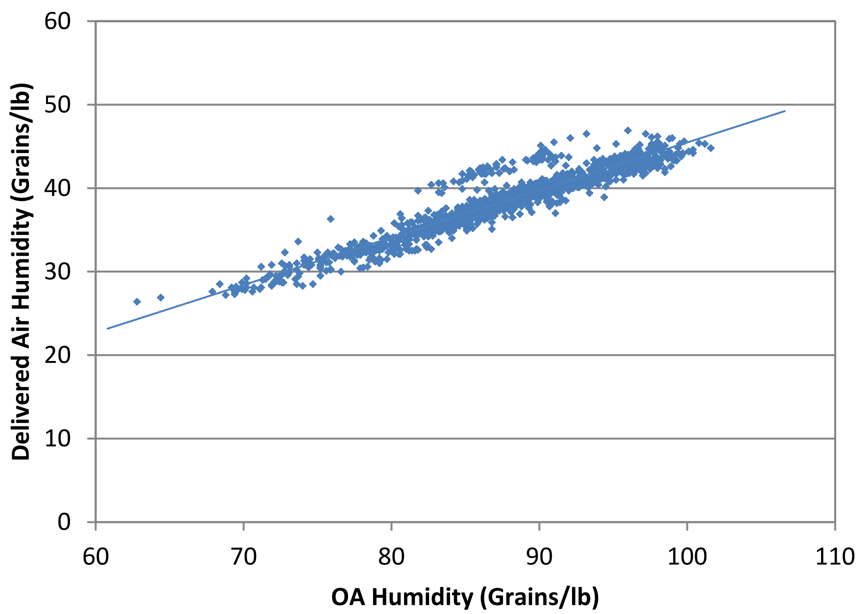

Figure 2–10 LDAC delivered humidity versus OA humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

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Table 2–6 Whole Foods Market Encinitas, California–Key Performance Metrics, August–October 2012

Performance Metric	Average	Range
Outdoor humidity (gr/lb)	87	63–102
Indoor humidity (gr/lb)	50	38–62
Delivered humidity (gr/lb)	38	34–42
Delivered temperature (°F)	70.5	68.7–74.1
MRC (lb/h)	126	118–134
RSHI (natural gas)	1.8	1.6–2.0
Electric power* (kW/ton)	0.33	0.31–0.35
MRC cost (gas + electricity) ($/lb)	0.015	0.013–0.028

*Chiller energy not included.

Figure 2–9 shows the hourly average conditions of the space, OA, and delivered air. The LDAC maintains a temperature that is almost equal to the delivered temperature. However, the humidity is influenced by the time of day, and has an element of infiltration during store open hours. This sensor is very near the supply registers of the conditioned air and thus would represent the driest location in the store. Clearly infiltration is a major component of store humidity levels, and delivery location of the LDAC air is critical to areas located nearest to the refrigerated cases, and most importantly nearest to the open cases.

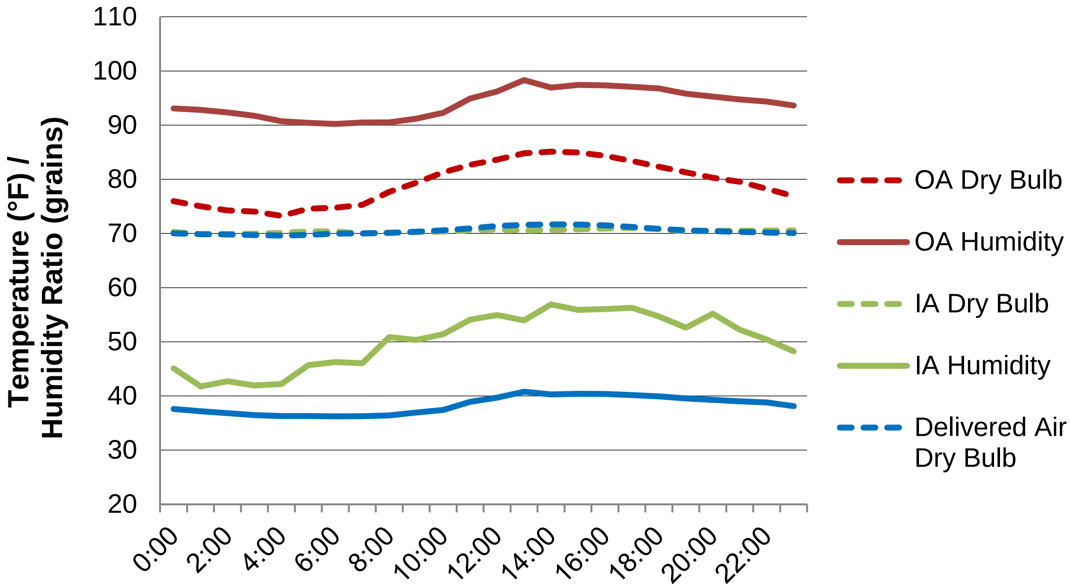

Figure 2–11 Store averaged conditions during each hour of the day, Encinitas, California, August–October 2012
(Credit: Eric Kozubal/NREL)

As expected, Figure 2–14 shows that the refrigeration power is lower when indoor air humidity levels are lower. The lower values correspond to morning and evening hours of operation, whereas higher values correspond to midday and early afternoon hours, as seen in Figure 2–9; occupant traffic almost certainly has a direct impact on both parameters.

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Figure 2–10 and Figure 2–11 show that refrigeration power is positively correlated to OA DB and indoor air humidity. To separate these two effects we recommend measuring the condensate from the refrigerated cases in future studies.

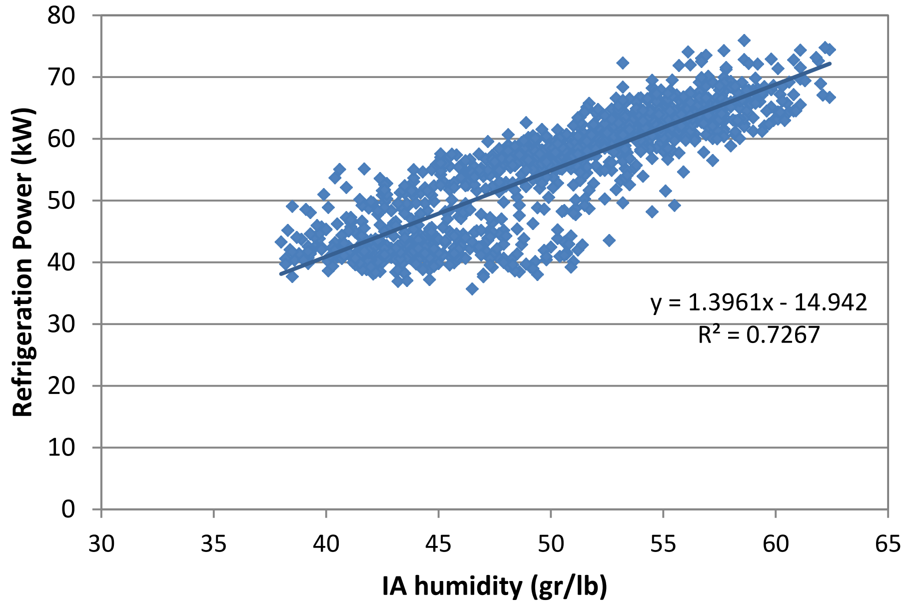

Figure 2–12 Refrigeration power versus indoor air humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

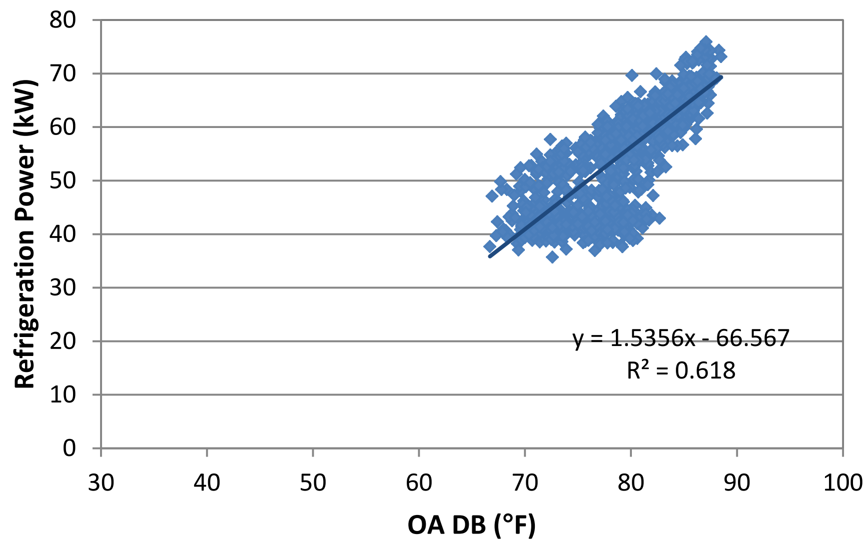

Figure 2–13 Refrigeration power versus OA DB, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

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Figure 2–12 shows that the MRC of the unit increases with OA humidity levels. This is due to the increase in system water removal capacity at higher OA humidity; as the difference in HRs (w) increases, the rate of latent heat transfer also increases according to the equation:

$$\dot{Q} = \dot{m}h_{fg}(w_1 - w_2) \qquad \text{(Equation 1)}$$

where h_{fg} is the latent heat of vaporization.

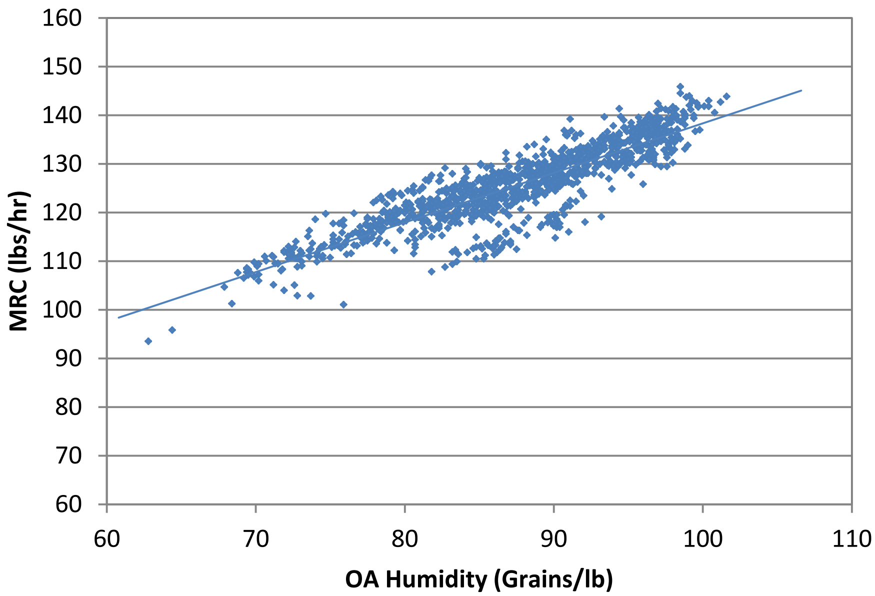

Figure 2–14 MRC versus OA humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

Because the unit can achieve a greater MRC at higher OA humidity levels, the electric efficiency of the LDAC (related to the constant-speed fans and pumps) also increases, as expected, because the electricity consumption remains constant while the unit capacity increases (see Figure 2–13). The thermal performance of the regenerator becomes slightly more efficient at higher OA humidity levels (see Figure 2–14). Although cycling losses were not measured, these were likely due to increased runtime, which resulted in fewer cycling thermal losses at higher water removal rates (higher OA humidity).

The MRC cost of the unit decreases slightly as a result of this increase in efficiency (see Figure 2–15). The total cost for dehumidification is the sum of the MRC cost components for electricity and gas. Note that the electricity cost is only a small percentage of the total. Based on energy costs from the store's utility bills ($0.12/kWh for electricity and $0.60/therm for natural gas), the dehumidification cost, or hourly operation cost for the LDAC, during this 50-day period was $1.89/h; the total cost of operation was about $1,234. These annual measured costs cannot be accurately extrapolated to other situations. See Section 3.0 for modeling results and economic metrics with broader generic application.

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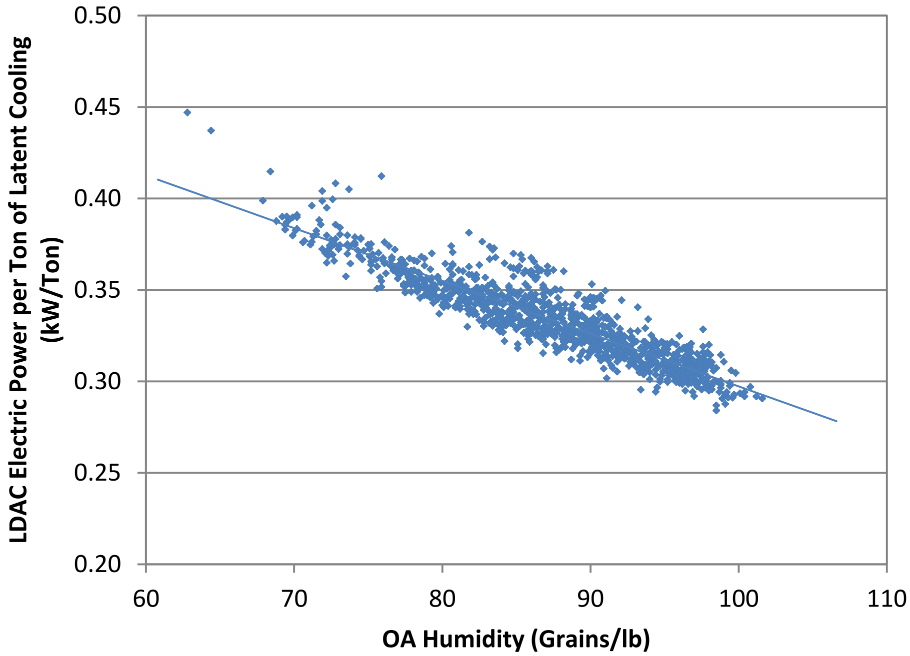

Figure 2–15 LDAC-specific electricity use versus OA humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

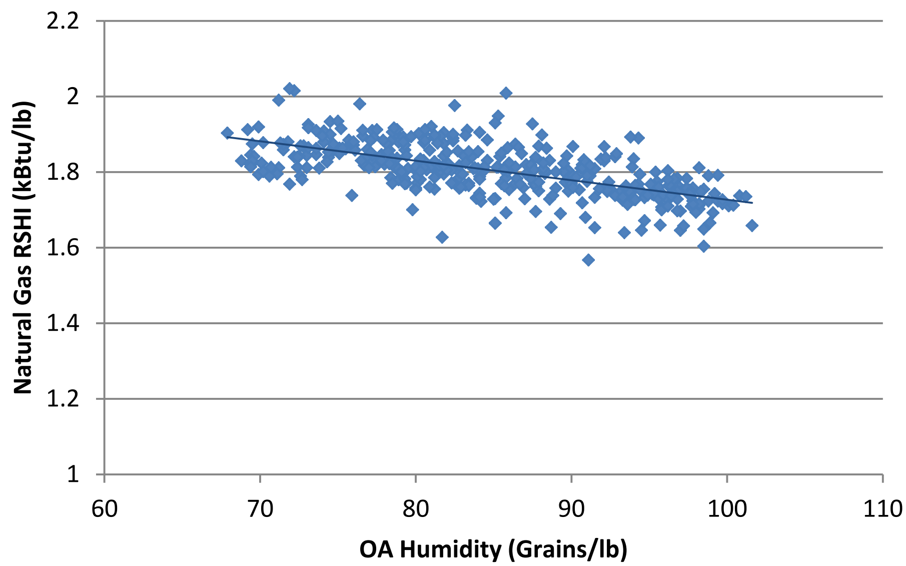

Figure 2–16 Natural gas RSHI versus OA Humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

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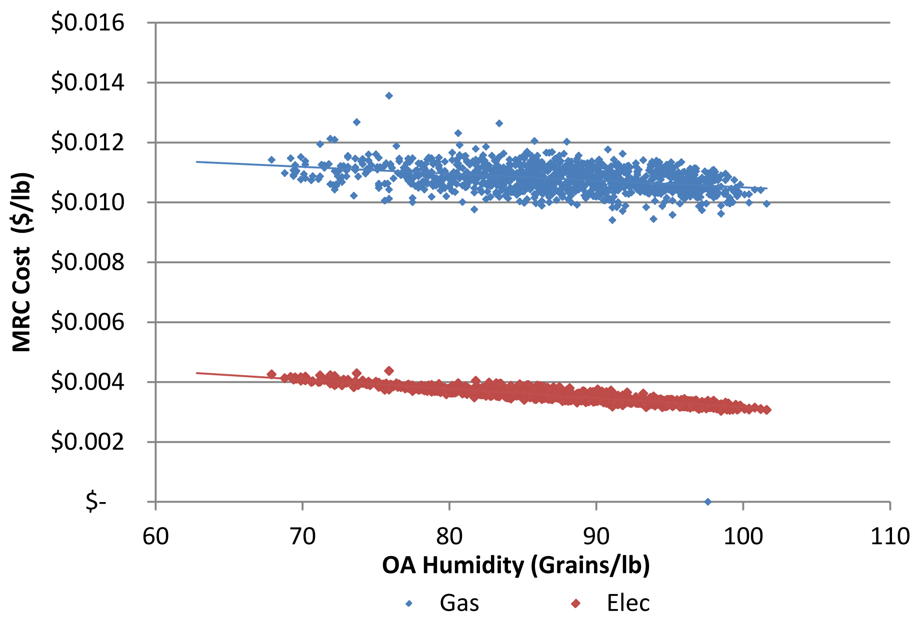

Figure 2–17 MRC cost versus OA humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

2.1.4 Operational Issues

An issue that arose at this site was precipitate formation in the desiccant of the regenerator loop. As shown in Figure 2–16 and Figure 2–17, scum formed on the top of the liquid in the desiccant sump, which clogged the filter and caused the unit to shut down. Solid precipitates were also found in the regenerator sump. A flame test confirmed the presence of sodium, which could have come from the salty air blowing off the Pacific Ocean. Another likely suspect is sulfur, because the scavenging air is ducted into the regenerator directly from the loading dock, which contains sulfurous exhaust fumes from diesel delivery trucks. In addition, lithium sulfate is not very soluble. These precipitates did not appear in the conditioner loop, which used a carbon filter on the OA supply. A carbon filter was later installed on the inlet to the scavenging air stream on the regenerator, which seemed to solve the precipitate problem.

Figure 2–18 Regenerator sump showing precipitate in the desiccant, Encinitas, California
(Credit: AIL Research, with permission)

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Figure 2–19 Regenerator filter clogged with precipitate, Encinitas, California
(Credit: AIL Research, with permission)

Another major issue was a leak in the regenerator. This was a result of adhesive disintegration, which caused the flocking surface to detach from the regenerator plates. The unit sensed the problem and shut down automatically, as designed. The unit was then disassembled and returned to the manufacturer for repair in late February 2013. A stronger adhesive was used to reattach the flocking surface to the regenerator plates and the unit was reinstalled in mid-March 2013. During the repair, weather conditions did not impose high latent loads and the building's conventional HVAC system was able to meet the air conditioning requirements.

2.2 Whole Foods – Kailua, Hawaii

2.2.1 Building and Climate Description

The Whole Foods Market in Kailua, Hawaii opened in April 2012 (see Figure 2–18). The conditioned sales floor area is approximately 30,000 ft^2 and is served by four packaged heat pump RTUs (refer to Table 2–6 for system specifications). The units are controlled with indoor temperature and RH sensors set to 72°F and 50% RH; all systems condition recirculation air. The mechanical refrigeration system is composed of four separate suction groups and is served by two air-cooled condensers (see Table 2–6). Supermarkets typically use overcool-and-reheat strategies for dehumidification; however, Hawaii's humid environment imposes a high latent load, which exacerbates the inefficiencies in HVAC and refrigeration systems when a conventional dehumidification approach is used. To address these issues, an LDAC system was included during the original design and construction of the building.

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Figure 2–20 Whole Foods Market, Kailua, Hawaii
(Courtesy of Whole Foods Market. "Whole Foods Market" is a registered trademark of Whole Foods Market IP, L.P.)

Table 2–7 Whole Foods Market, Kailua, Hawaii—HVAC and Refrigeration System Description

Component	Specification
Refrigeration system total compressor power	127.5 hp
Refrigeration system condenser temperature	110°F
Refrigeration system suction group temperature; power	–25°F; 30 hp; +35°F; 27.5 hp +15°F; 10 hp +20°F; 40 hp
HVAC rooftop heat pump capacity (4 units serving the sales floor)	32 tons each

Kailua is located approximately ½ mile from the Pacific Ocean. As with the Encinitas location, the Kailua store's proximity to the ocean and ambient temperature and humidity create a similarly high demand for dehumidification to reduce HVAC and refrigeration energy use. Figure 2–19 shows the TMY3 hourly weather data plotted on the psychrometric chart and Figure 2–20 shows the distribution of humidity conditions. The climate is warm and humid most of the year. Assuming reference process air conditions of 75°F DB and 45°F DP (as described in Section 2.1.1, the LDAC is expected to operate 8,753 hours per year (see Table 2–7).

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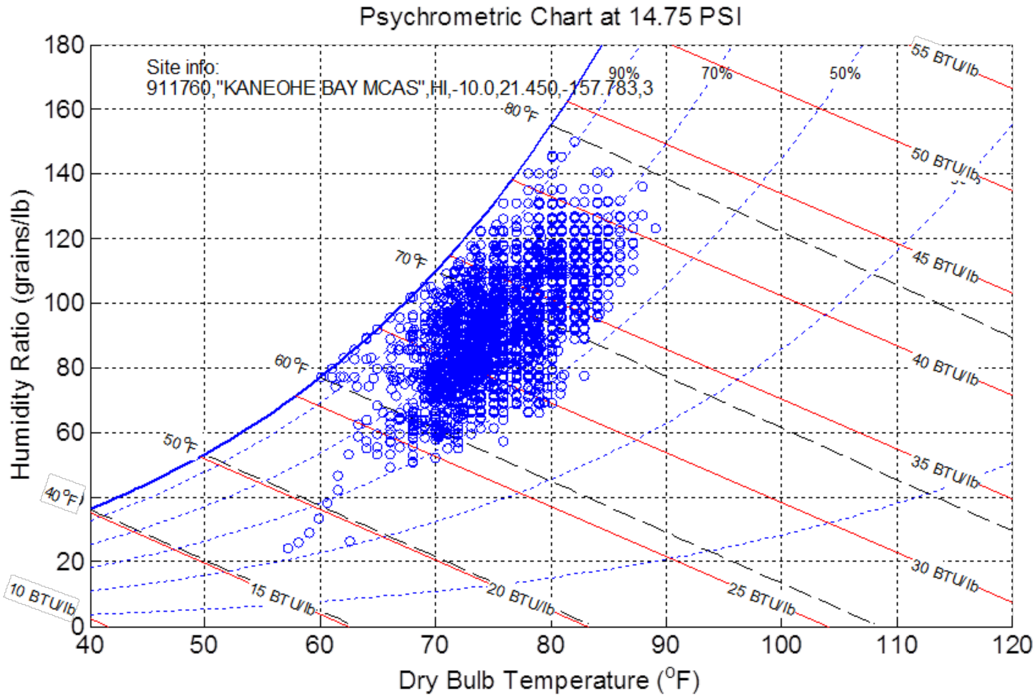

Figure 2–21 Climate analysis, Kailua, Hawaii
(Credit: Eric Kozubal/NREL)

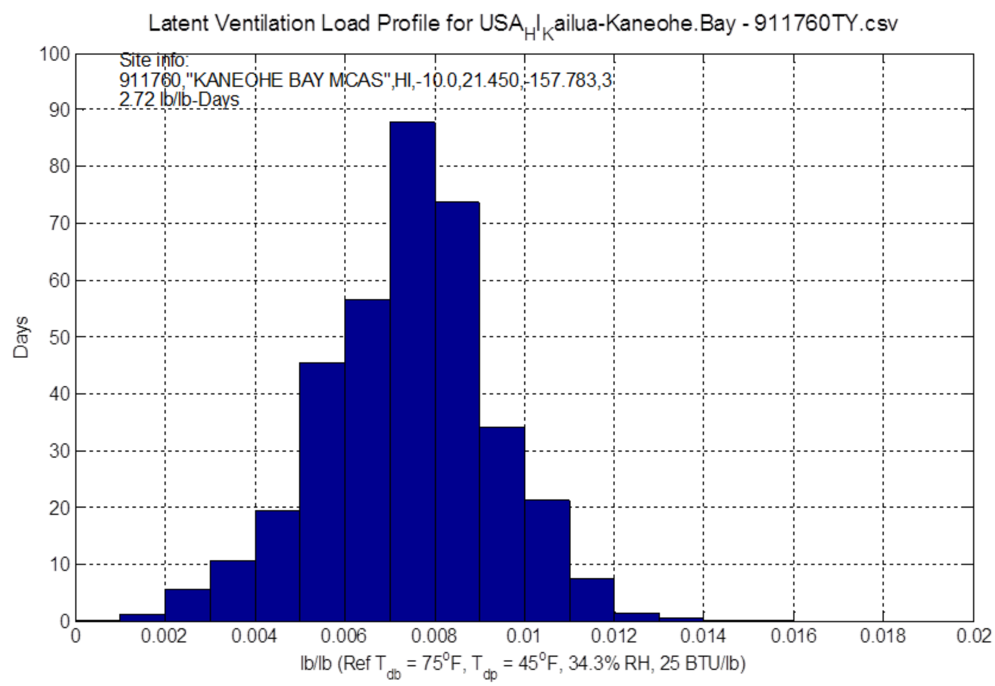

Figure 2–22 Pound-per-pound days, Kailua, Hawaii
(Credit: Eric Kozubal/NREL)

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Table 2–8 Kailua-Kaneohe Bay, Hawaii— Annual Dehumidification Loads (75°F DB, 45°F DP reference)

Specification	Load
Total ventilation load (Btu/lb-days)	7,331
Total moisture load (lb/lb-days)	2.72
Estimated hours of operation (h)	8,753
ASHRAE 1% design conditions (DP; HR; MCDB)	74.6°F; 129.7 gr/lb; 80.1°F

2.2.2 LDAC Design

This LDAC system was commissioned in April 2012. The LDAC deeply dries the OA supply before it mixes with return air and is ducted into one of the four heat pump units (RTU-4) for sensible cooling (see Figure 2–22). In an effort to reduce the latent load on the evaporator coils of the other RTUs, all the required ventilation air was supplied by the LDAC and delivered to this RTU. The cool dry air was then mixed with conditioned return air before being supplied to the space. This air was strategically introduced above the open multideck dairy deli cases and near the reach-in frozen food and ice cream cases in an effort to reduce the frequency and duration of defrost and display case ASH cycles. A temperature and RH sensor, located above the frozen food case, monitored the indoor air conditions and signaled the LDAC to shut off when the space DP was about 43°F (i.e., when RH was 35% or lower).

The desiccant is regenerated with thermal heat from two sources: (1) an evacuated tube solar thermal array; and (2) a propane-fired boiler. The solar thermal system demonstration enabled NREL to evaluate the economic feasibility of this alternative thermal energy source in a location with relatively high gas and electricity costs. The LDAC system includes a cooling tower that provides cooled water to remove the heat of absorption from the LDAC air drying element. It also includes approximately 2 h of thermal energy storage (using a hot water tank), which provides consistent heat for the desiccant regenerator element from the combined solar and propane thermal system. The design specifications for the LDAC are listed in Table 2–8.

Figure 2–23 LDAC unit and solar system at Whole Foods Market, Kailua, Hawaii
(Credit: Ian Doebber/NREL)

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Table 2–9 Whole Foods Market, Kailua, Hawaii—LDAC Description

Specification	Design
Latent cooling capacity	19 tons
Supply airflow rate for space ventilation	4,000 cfm
Air RH supplied by LDAC	18%–20%
Liquid desiccant concentration	43% LiCl
LDAC latent COP (gas basis)	0.65
LDAC latent COP (solar heat basis)	0.8
Solar thermal array	Apricus AP-30
• **Area**	3,500 ft^2 (80 panels)
• **Heat transfer fluid**	Water
• **Tilt angle/ azimuth**	34°/200°

All components except the thermal storage are on the roof; the thermal storage is in the loading dock. The roof equipment layout and solar thermal piping diagram are shown in Figure 2–22 and Figure 2–23, respectively. Figure 2–23 is the specific schematic for the Kailua project. The Kailua solar system had several problems (see Section 2.2.4). There are several resources for designing and integrating solar thermal systems into a variety of overall system designs, including: (1) the ASHRAE Active Solar Heating Systems Design Manual 1988 (ASHRAE 1990); (2) The Solar Rating & Certification Corporation Commercial Systems Index (ASHRAE 1988); and (3) the ASHRAE District/Central Solar Hot Water Systems Design Guide 2013 (ASHRAE 2013). We recommend using these resources when designing a solar system with an LDAC system. The installed costs of the LDAC and the solar thermal array are listed in Table 2–9.

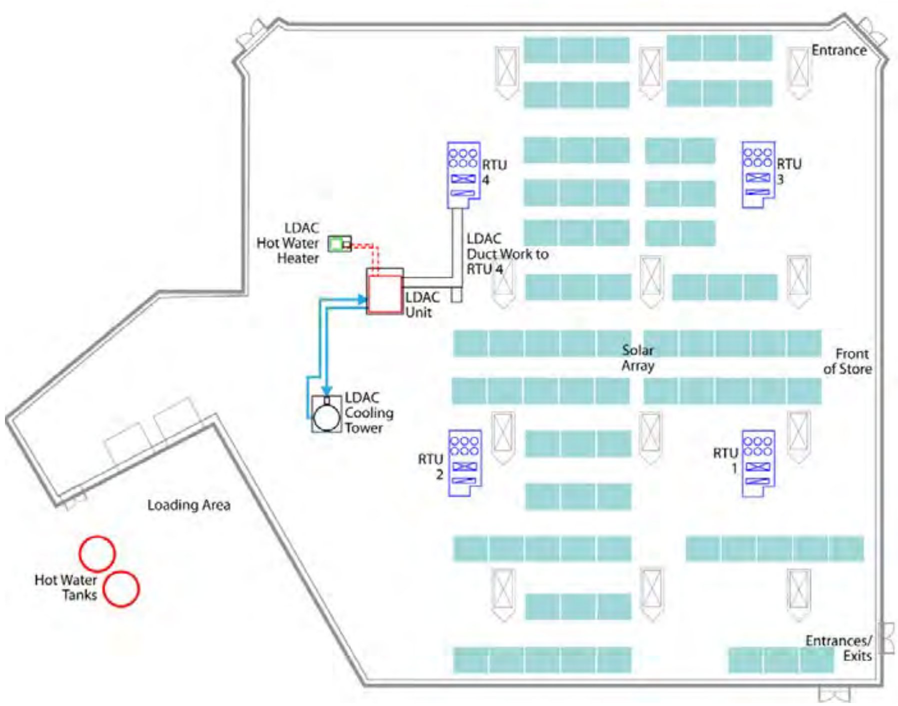

Figure 2–24 Roof equipment layout at Whole Foods Market, Kailua, Hawaii
(Credit: NREL, adapted from Whole Foods Market, with permission)

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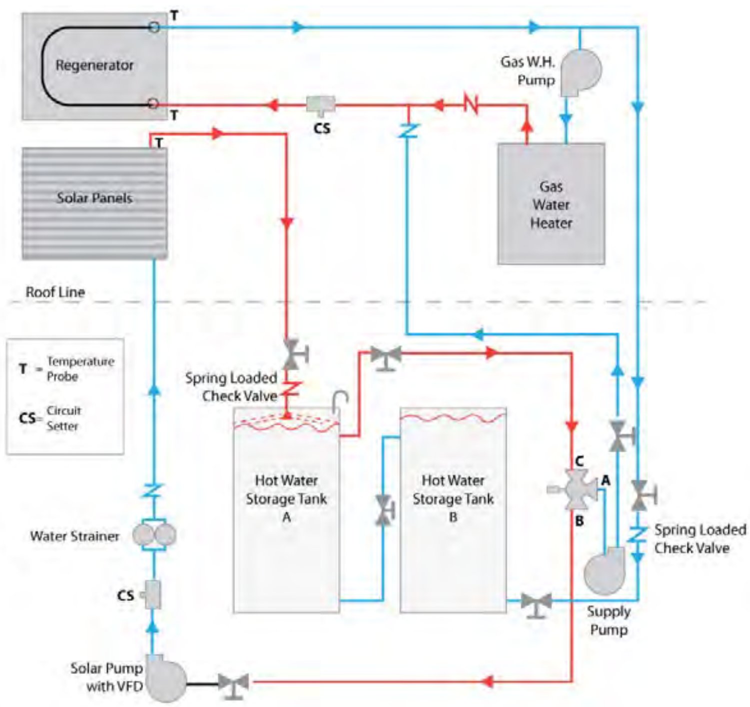

Figure 2–25 Solar thermal piping schematic at Whole Foods Market, Kailua, Hawaii
(Credit: J&J Mechanical, with permission)

Table 2–10 Whole Foods Market, Kailua, Hawaii—LDAC System Installed Costs

Component	Cost ($)
LDAC	75,000
Cooling tower (including pump, filters, valves, etc.)	3,850
Boiler (including pump, expansion tank, etc.)	6,147
Solar array (including hot water tank, evacuated tubes, fame, pump, etc.)	226,500
Outbound freight cost	18,000
Installation	118,000
Total cost	429,497

As with the Encinitas grocery store, the refrigeration system was not downsized with respect to the LDAC because the LDAC manufacturer wanted to mitigate the risk of damaging products. Refrigeration energy savings should still result from lower latent loads caused by drier indoor air. The defrost cycles are currently programmed to operate on daily timed schedules; however, the intention is to reprogram this system according to evaporator coil conditions once the LDAC has shown a significant period of continuous operation and key building operation issues are under control. These issues will be discussed in Section 3.0.

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2.2.3 Performance Results

The full period of performance for this unit spanned August 2012 through August 2013. During this time, the LDAC sometimes performed according to expectations; at other times the system experienced mechanical problems, some of which were associated with the LDAC and some of which were associated with a solar system designed to supply hot water for the LDAC regenerator (see Section 2.2.4). The analysis presented here highlights the LDAC's performance while operating either with the solar thermal system (solar mode) or with the propane boiler (boiler mode). Solar mode refers to times when the regenerator uses thermal energy from the solar system during the day and receives no heat from the boiler. This includes times when solar generation has stopped and heat is delivered from the hot water storage. Boiler mode refers to times when the regenerator operates on propane.

For the 32-day period from July 22 to August 30, 2013 the LDAC performed according to expectations (see Table 2–10). During this time, the regenerator operated in boiler mode 100% of the time. Figure 2–24 shows the OA and delivered air conditions on a psychrometric chart. The LDAC is able to dry very humid OA to an average HR of 59 gr/lb (0.0084 lb/lb) for this period of performance. The RSHI over the range of OA humidity conditions averaged 1.5 kBtu/lb when considering only the thermal performance of the regenerator and 1.8 kBtu/lb when the efficiency of the propane boiler is taken into account (see Figure 2–25). The electrical efficiency of the unit averaged about 0.32 kW/ton (for fans and pumps) (see Figure 2–26).

Table 2–11 Whole Foods Market, Kailua, Hawaii—Key Performance Metrics, July–August, 2013

Performance Metric	Average	Range
Delivered air humidity (gr/lb)	59	49–69
MRC (lb/h)	147	120–174
Electric power (kW/ton)	0.32	0.26–0.38
RSHI – regenerator (Btu/lb)	1.5	1.2–1.9
RSHI – natural gas (Btu/lb)	1.8	1.5–3.4
MRC cost – gas + electricity ($/lb)	0.09	0.06–0.16

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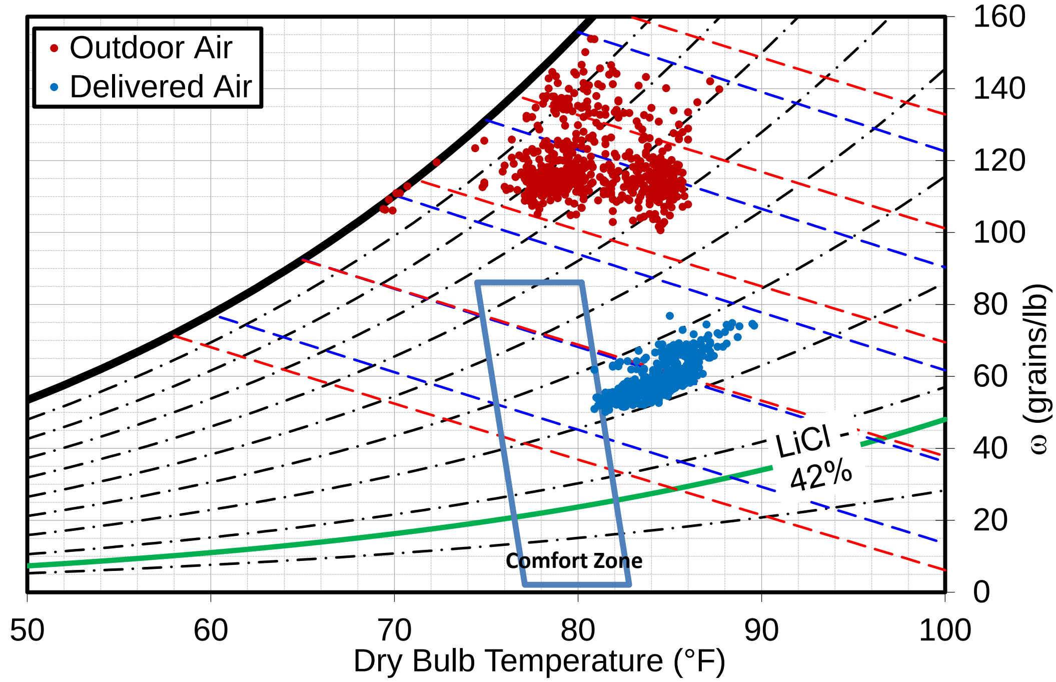

Figure 2–26 OA and delivered air plotted on a psychrometric chart, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

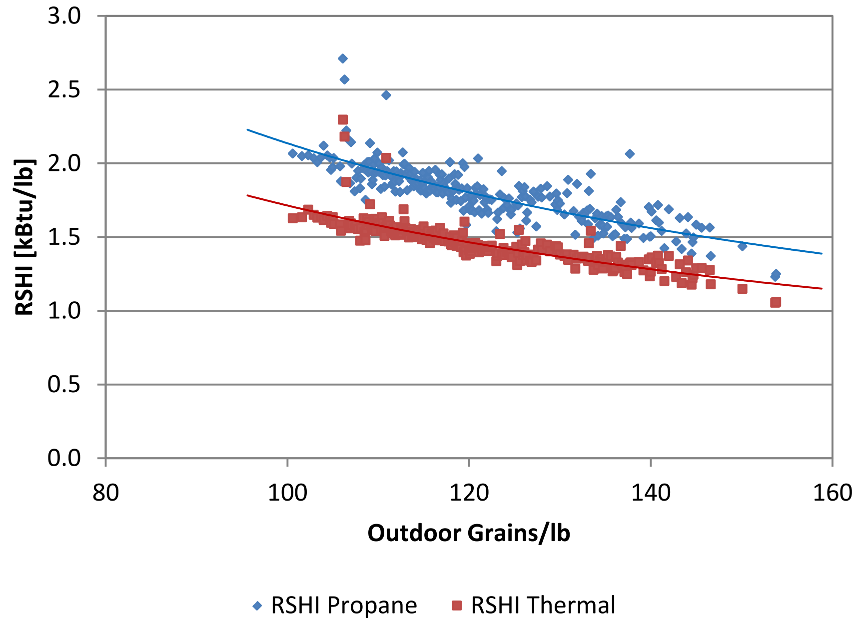

Figure 2–27 Regenerator RSHI, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

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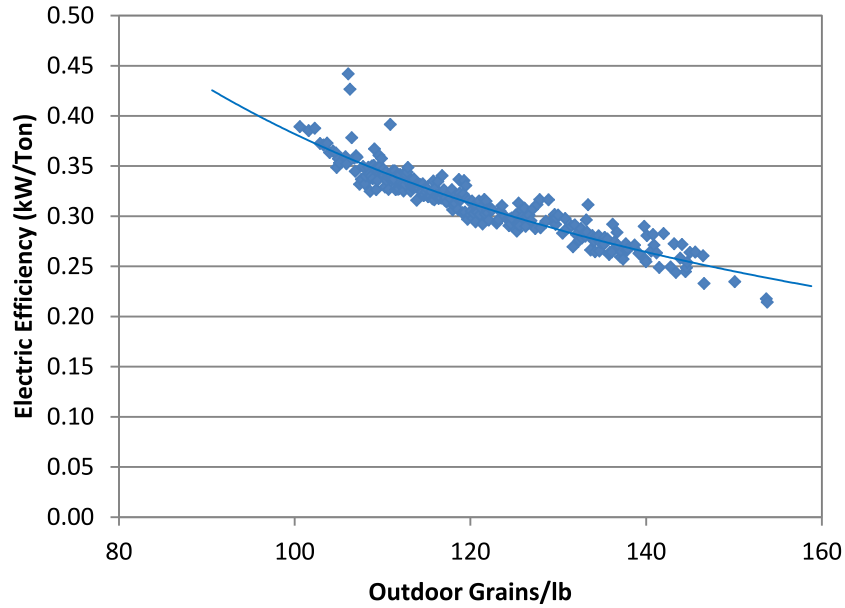

Figure 2–28 LDAC electrical efficiency, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

The MRC of this unit averaged 147 lb/h (see Figure 2–27). The unit was expected to supply air below 30% RH; however, the measurements showed that the supply air was often in the 30%–35% range (still quite dry) for this period. This is because the regenerator has a suspected water leak that reduces its capacity by about 10% than designed. The wicking fin regenerator is expected to solve this problem because the metallic construction is less prone to water leaks than the current plastic regenerator design. The MRC cost shows that the gas component cost of the system decreases with OA humidity levels, while the small electricity cost component stays relatively constant (see Figure 2–28). The decrease in MRC gas cost is due to the increase in system efficiency at higher OA humidity; as the difference in HR increases, the boiler operates more efficiently and at a higher temperature. MRC electricity cost is more or less constant because pump and fan energy is independent of the latent load imposed by outdoor conditions. The total cost for dehumidification is the sum of the MRC cost components for electricity and gas. Using the local utility costs for Kailua, Hawaii ($4.25/therm and $0.30/kWh), the dehumidification cost ranged from about $0.06 to $0.13/lb water removed when the regenerator operated on propane 100% of the time. The average cost was $0.09/lb. The average operating cost was $12.64/h and the total operation cost over this 30-day period was about $3,323.

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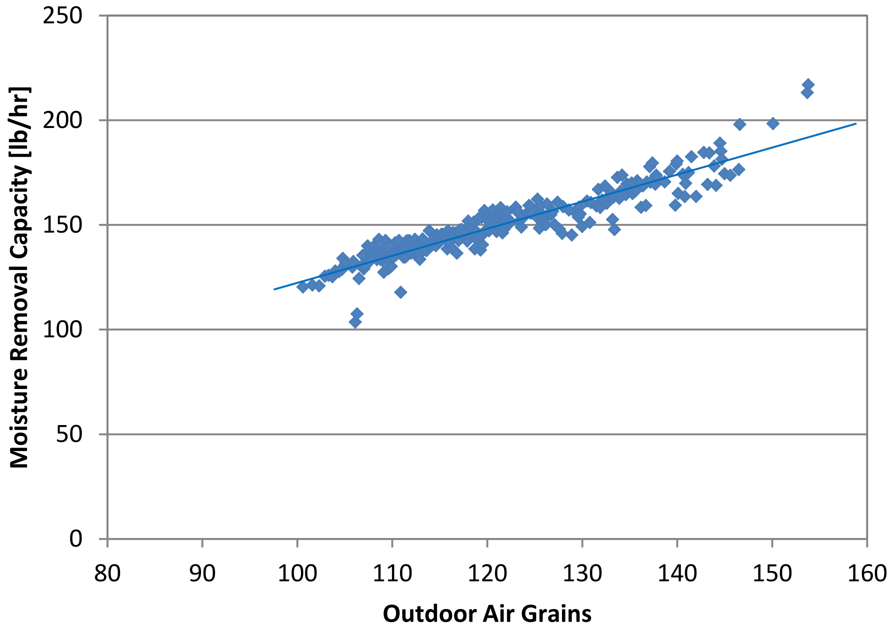

Figure 2–29 LDAC MRC, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

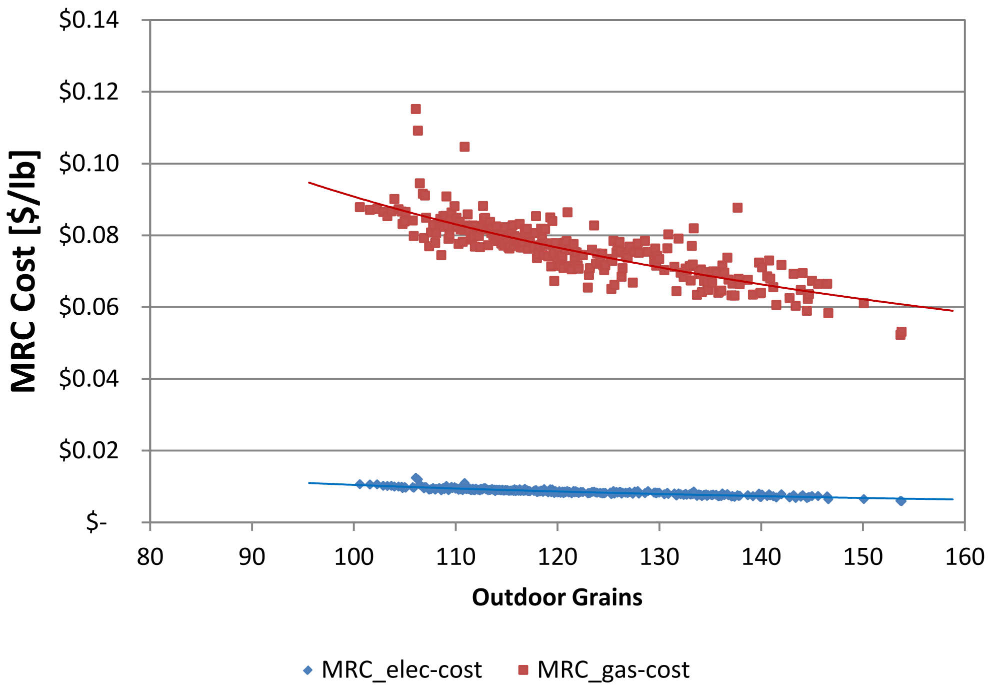

Figure 2–30 LDAC MRC cost, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

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This store had major unanticipated infiltration control issues because the two main entry ways and the loading dock doors were kept open. The main entry doors were often locked in the open position to create an "open-air market" type feel. Also, employees did not understand the effects of infiltration on store conditions and paid little attention to it. This led to a negative pressure in the space and high infiltration rates on the sales floor (estimated to be up to 20,000 cfm). This appears to be a cultural response to a warm-humid environment where air velocity across the human body has traditionally been an effective passive comfort strategy. However, in a grocery store, this is counterproductive from an energy and comfort perspective. Even when store management was informed of this issue the doors remained open. Despite this, the LDAC showed it could consistently provide dry air to the space as designed, but it did not have the capacity to keep up with the excessive latent load brought about by the infiltration. Thus, the RTUs had to be run in dehumidification mode (reheat enabled) for much of the time to help remove the unusually high latent load.

Figure 2–29 shows histograms of HR and RH for the supply air from the LDAC, the indoor conditions, and the outdoor conditions (note that "dairy aisle" data were recorded at the sensor location above the frozen food cases and near the diffuser where LDAC conditioned air enters the space). Figure 2–30 shows the hourly conditions plotted on a psychrometric chart with the comfort zone shown. The LDAC supplied air at 30%–35% RH for most hours, but store conditions were 60%–65%, and often higher than 70%. To address this issue, the store manager adjusted the RTUs to provide additional dehumidification. However, the RTUs were not equipped with sufficient reheat capacity to maintain comfortable temperatures, so the store was often cold; employees frequently complained of uncomfortable conditions (see Figure 2–31). Not only did this eliminate any possible energy saving potential from the RTUs, the additional infiltration caused frost buildup on products, sweating cases, and foggy display case doors, eliminating any expectations of refrigeration energy savings (see Figure 2–32).

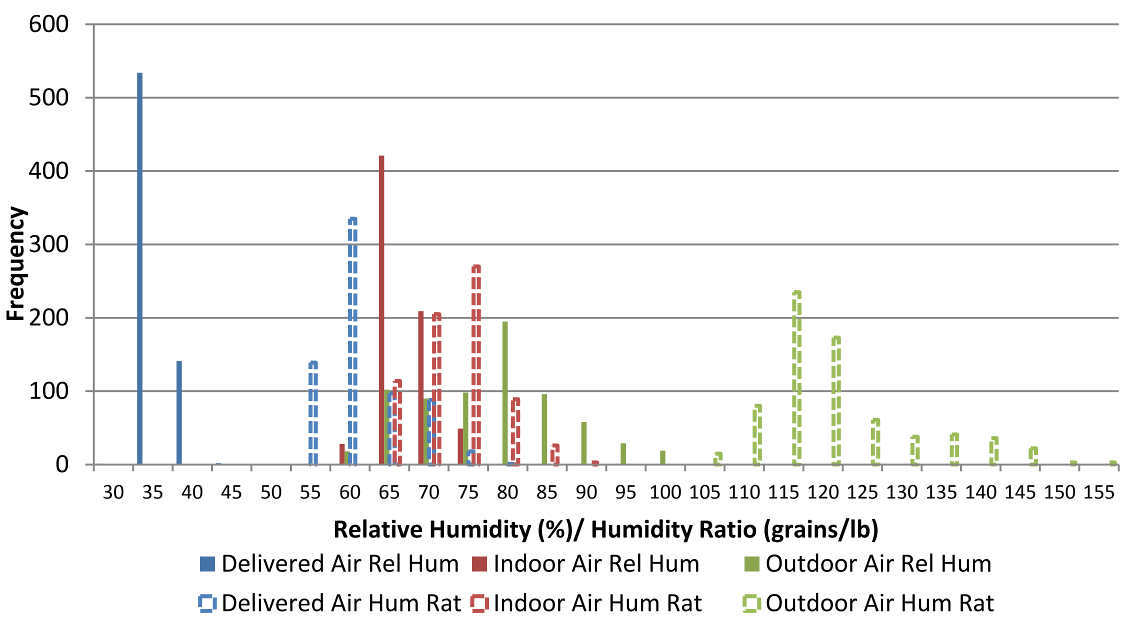

Figure 2–31 Store and site humidity levels, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Lesley Herrmann/NREL)

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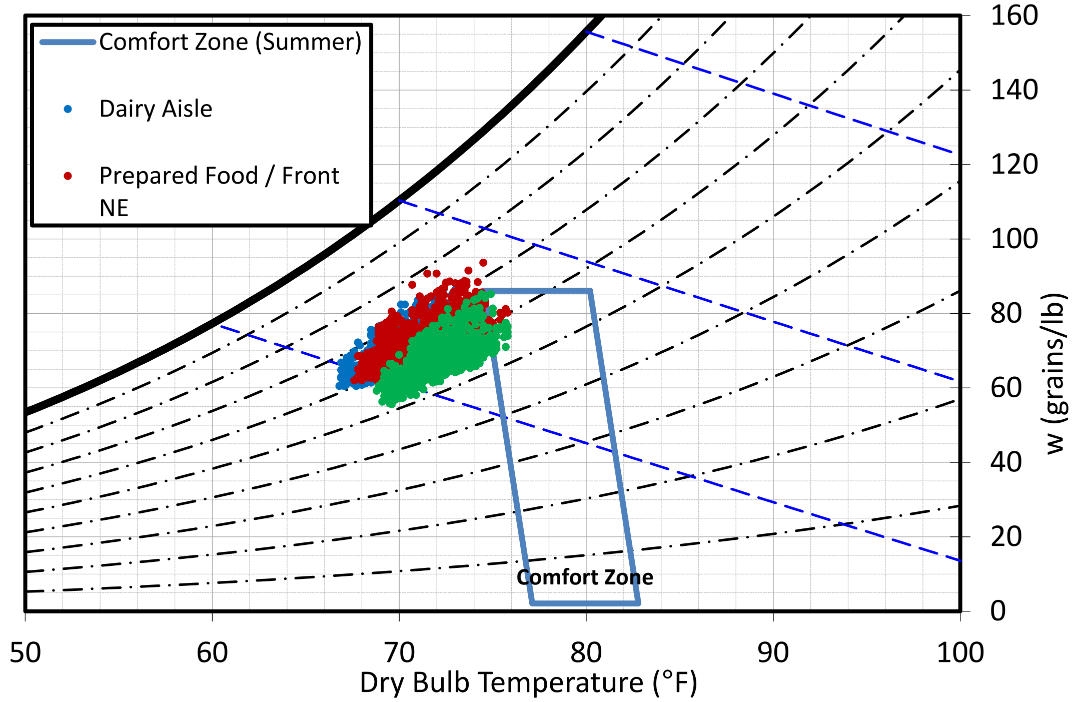

Figure 2–32 Psychrometric chart showing indoor air conditions, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Eric Kozubal/NREL)

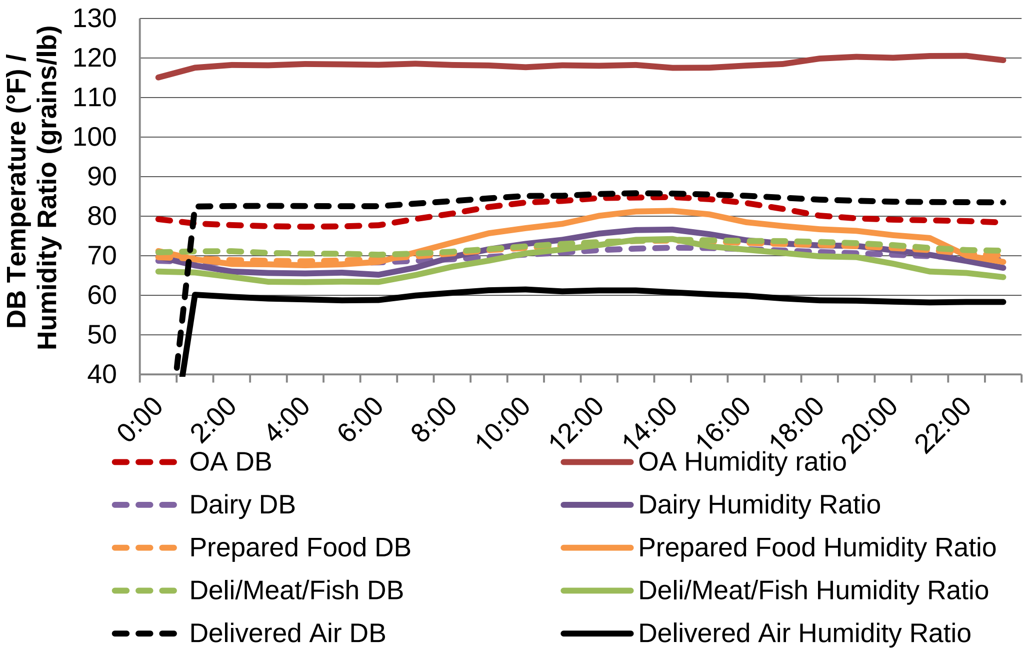

Figure 2–33 Daily average conditions, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

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Figure 2–34 Effects of high humidity on grocery store refrigeration, Whole Foods, Kailua, Hawaii, summer 2013
(Credit: Ian Doebber/NREL)

To show how the LDAC unit would perform without the additional latent load from the open doors, a series of tests with different LDAC and RTU controls were carried out from May 3 through June 3, 2013. Data from these tests were analyzed at night to observe energy consumption and store conditions while the building's doors were closed. Table 2–11 describes the three tests. Figure 2–33 shows the power draw for the LDAC electric, propane, RTUs and refrigeration systems. The dips in LDAC propane power during the tests were due to displacement by solar thermal power. However, the propane was not displaced during nighttime hours during this analysis. Table 2–12 summarizes the average power draw for each of the systems during the hours from 11pm until 5am when the store is closed and the customer entrances are shut. The site to source energy conversion factors were 4.022 for electricity and 1.23 liquid propane gas (LPG) (Deru et al. 2007). The LPG factor of 1.23 was calculated using the mainland U.S. conversion factor (1.15) adjusted for additional pre-combustion energy to transport the fuel to Hawaii using the same ratios cited for fuel burned in electric plants in the mainland United States (1.05) versus Hawaii (1.12).

$$\text{Hawaii Precombustion Adjustment Factor } = \frac{1.12}{1.05} = 1.067$$

$$\text{LPG Site to Source Factor } = 1.067 \times 1.15 = 1.23$$

Table 2–13 shows just the electric power savings for test 2 and 3 versus test 1. Figure 2–34 shows the indoor and outdoor DBs and HRs for all three tests. Note the slight increase in indoor air DB, but no increase in humidity from test 1 to test 2 despite increased OA DB and humidity. As discussed in the Encinitas section, we are unable to disaggregate indoor air humidity and OA DB effects on refrigeration power.

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Table 2–12 Whole Foods Market, Kailua, Hawaii—
Nighttime/Low Infiltration Test Matrix, May–June 2103

Test Number	Dates of Tests	Test Description
Test 1	May 3 12	• LDAC off • RTUs on 24 h/day • Ventilation through RTUs 24 h/day
Test 2	May 20–24	• LDAC on 24 h/day • RTUs off at night (11:00 p.m. to 5:00 a.m.) • Ventilation through LDAC 24 h/day
Test 3	May 26–June 3	• LDAC off at night • RTUs off at night • No Ventilation

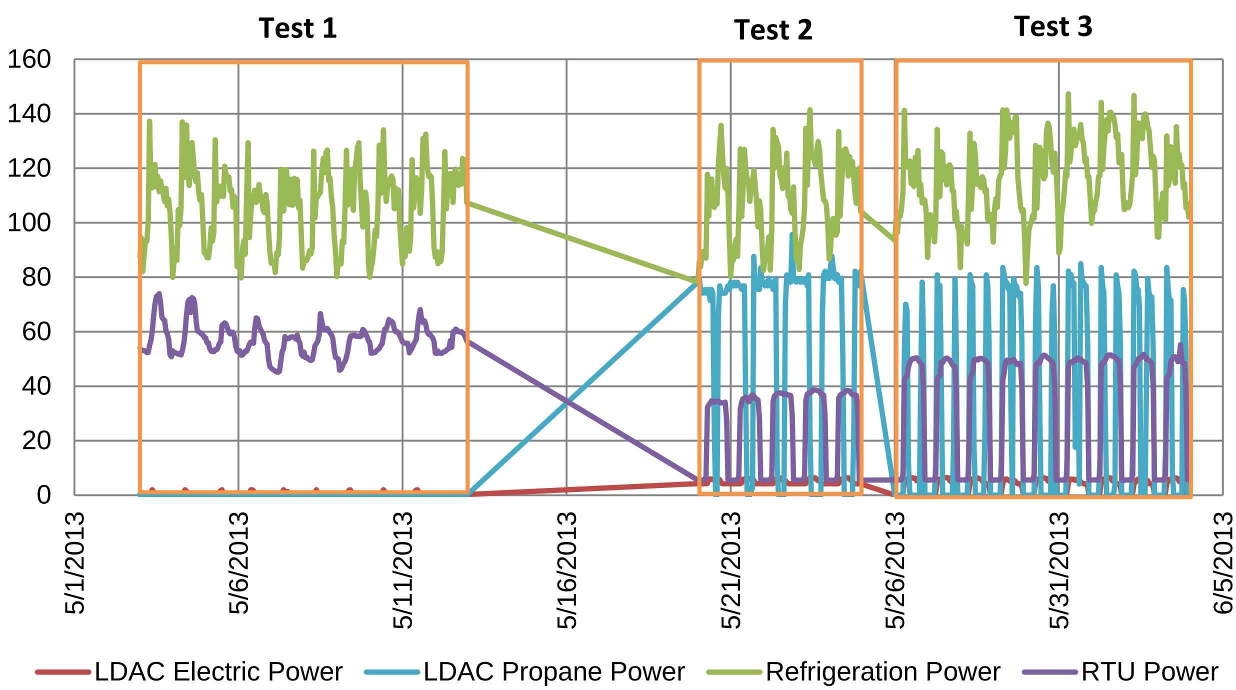

Figure 2–35 Refrigeration and HVAC systems power for tests 1–3,
Whole Foods, Kailua, Hawaii, May–June 2103
(Credit: Eric Kozubal/NREL)

Table 2–13 Whole Foods Market, Kailua, Hawaii—
Nightly Average Energy Impacts for Tests 1–3, May–June 2103

	RTU Power (kW)	Refrigeration Power (kW)	LDAC Power (kW)	LDAC Propane Power (kW)	Source Power (kW)
Test 1	52	89	0.0	0	567
Test 2	6	91	4.2	79	500
Test 3	6	102	0.0	0	431

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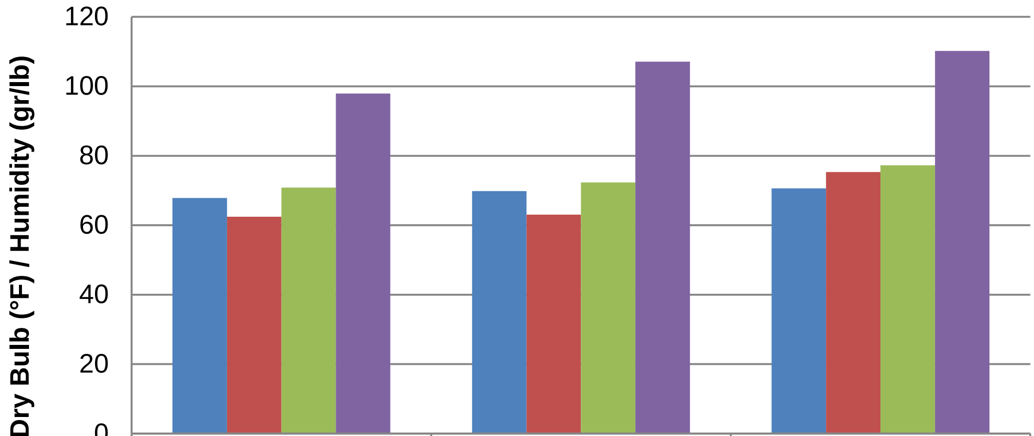
Table 2–14 Whole Foods Market, Kailua, Hawaii— Nightly Average Electric Power Savings for Tests 2 and 3, May–June, 2103

	RTU Power (kW)	Refrigeration Power (kW)	LDAC Power (kW)	Total Electric Power (kW)	Percent Savings
Test 2	46	-2	-4	41	29%
Test 3	46	-13	0	34	34%



	Test 1	Test 2	Test 3
IA Dry Bulb	67.8	69.8	70.6
IA Humidity	62.5	63.1	75.3
OA Dry Bulb	70.9	72.3	77.3
OA Humidity	97.9	107.1	110.2

Figure 2–36 Average indoor and outdoor air conditions for nighttime tests 1–3, Whole Foods, Kailua, Hawaii, May–June 2103
(Credit: Eric Kozubal/NREL)

The net source power savings for test 2 over test 1 was about 12% and similarly 24% for test 3 over test 1. This result suggests the following conclusions:

1. If the building's infiltration problem were reduced to nighttime levels for the entire day, the LDAC could maintain humidity control without the RTUs. The RTUs could then be used solely for temperature control. Such a strategy could save approximately 12% in source energy and 29% of site electric power as demonstrated by this experiment. However, this result is a reflection of a short time period and extrapolation to an entire year is only approximate.
2. Both the LDAC and RTUs should be shut off during closed hours when ventilation to the space is not necessary.

The LDAC manufacturers recommended a modification to the door operation schedule as well as a vestibule addition to the front of the building to help the store reduce infiltration and thus save energy. The building owner is currently working on a retrofit plan for the front of the store.

During these series of tests, the LDAC operated in solar mode. As expected, the regenerator's performance is nearly unaffected by the source of thermal energy, as long as each system

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provides the same temperature of water (see Figure 2–35). The average thermal RSHI in both solar mode and boiler mode averaged 1.9 kBtu/lb.

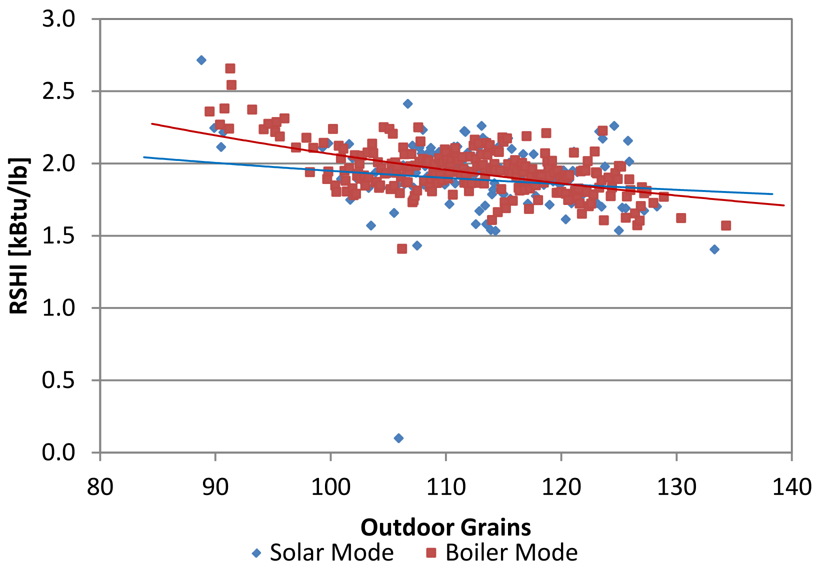

Figure 2–37 Regenerator thermal RSHI, Whole Foods, Kailua, Hawaii, May–June 2103
(Credit: Joe Ryan, with permission)

The MRC cost of the LDAC in boiler and solar modes is shown in Figure 2–36 as a function of the ambient humidity. As mentioned before, the specific cost decreases as ambient humidity increases because the unit becomes more efficient when the difference between OA and supply air HRs increases (see Equation 1). The average MRC cost is \$0.016/lb in solar mode and \$0.11/lb in boiler mode (see Figure 2–36). The total cost over this time period is \$3,038 in boiler mode and \$263 in solar mode. Note that the LDAC operated for roughly 760 hours in boiler mode and 400 hours in solar mode. The difference in total operation cost may also be impacted by variable weather conditions. This shows that the operational cost of the LDAC can be significantly reduced if thermal energy is provided by a solar system or a source of waste heat; however, the capital and maintenance costs of a “free heat” system, which are not accounted for in this analysis, must also be considered in an economic analysis.

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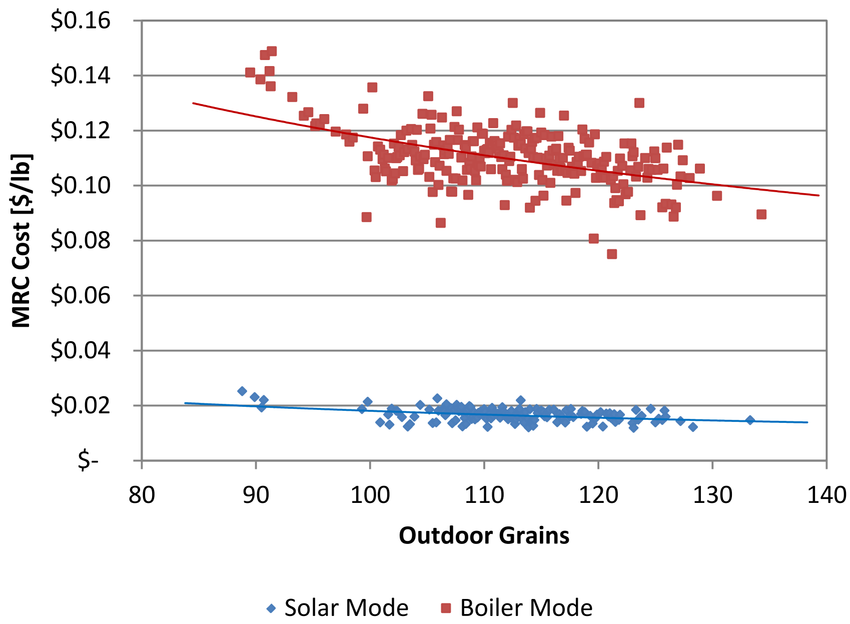

Figure 2–38 LDAC MRC cost, Whole Foods, Kailua, Hawaii, May–June 2103
(Credit: Joe Ryan, with permission)

2.2.4 Operational Issues

The primary operational issue at this site was the integration between the solar thermal system and the boiler. The solar thermal system did not always operate at its maximum capacity, which caused lower than optimal hot water temperatures. The design did not allow for a smooth transition between the solar thermal system and the boiler, so the LDAC capacity was often reduced. Future designs should allow for simultaneous solar thermal and boiler operation to provide consistently hot water temperatures in order to maximize efficiency and capacity. The system also experienced solar pump failures, hot water diverting valve leaks, and rainwater penetration into the desiccant solution, which caused the unit to trip off automatically. Many of these shutdowns could have been avoided with a more robust integration of the solar thermal system and the boiler. These observations will help the manufacturers and designers avoid these kinds of problems in the future. This unit did not experience any precipitate issues.

2.3 Schaeffer Natatorium – Stevens Institute of Technology

2.3.1 Building and Climate Description

The Schaefer Athletic and Recreation Center at the Stevens Institute of Technology in Hoboken, New Jersey, houses a 45-ft × 75-ft swimming pool (see Figure 2–37). The pool and surrounding area are generally maintained at 80°F. A major challenge in this building, as with any indoor pool facility, is maintaining sufficiently dry indoor air conditions for comfort, and to avoid mold and mildew problems from condensation. As a complicating factor, drier air increases the pool water evaporation rate, which in turn increases the need for pool water heating. The space is conditioned and dehumidified by an air handling unit (AHU) that processes return air and OA and includes chilled water coils supplied by a central plant chiller, and preheat and reheat hot water coils supplied by a central boiler. A remote relief damper allows for space exhaust. To control condensation, exterior windows were tempered with hot air jets served by a designated

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AHU. The pool was heated by a water-to-water heat exchanger connected to a natural gas cogeneration system. The original building systems are described in Table 2–14.

Figure 2–39 Schaefer Pool at Stevens Institute

(Credit: AIL Research, with permission)

Table 2–15 Schaefer Natatorium—Original HVAC System Description

Component	Specifications
Chilled water coil	11,000 cfm (5.2 m^3/s) 45°F (7.2°C) chilled water temperature 22 tons sensible cooling; 14.5 tons latent cooling
Water-to-water heat exchanger	681 kBtu/h (252 kW)
Cogeneration system	75 kW electric capacity, natural gas powered
Window air jets	3,240 cfm airflow 120°F supply air temperature 140 kBtu/h

A simple climate analysis shows that this location has a range of climate conditions, from hot and humid to cold and dry, as shown in the psychrometric chart of Figure 2–38. The RH exceeds 65% for about half the year and 80% about one third of the year. Figure 2–39 shows the distribution of latent ventilation loads in this climate. Unlike the Kailua climate, humidity loads are more varied throughout the year; maximum humidities reach higher levels than those seen in Encinitas. This analysis indicates a need to dehumidify ventilation air at this location. Assuming the same reference conditions used previously (75°F DB and 45°F DP), roughly 4,300 hours per year require dehumidification (see Table 2–15). As mentioned previously, the reference point was selected based on the ideal product air conditions from the LDAC. The LDAC will likely always be designed to deliver 45°F DP air to maintain higher delta-enthalpy, which will keep system and operation costs as low as possible. This analysis is an example of how an LDAC would treat OA to dehumidify the space. Treatment of indoor air requires a separate analysis, which has not been identified here.

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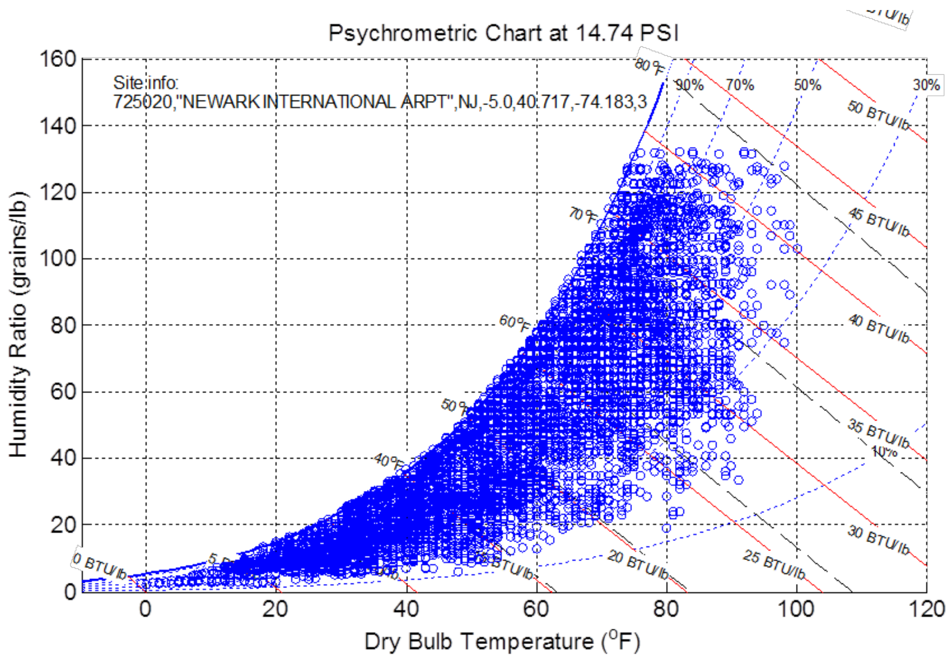

Figure 2–40 Climate analysis, Hoboken, New Jersey

(Credit: Eric Kozubal/NREL)

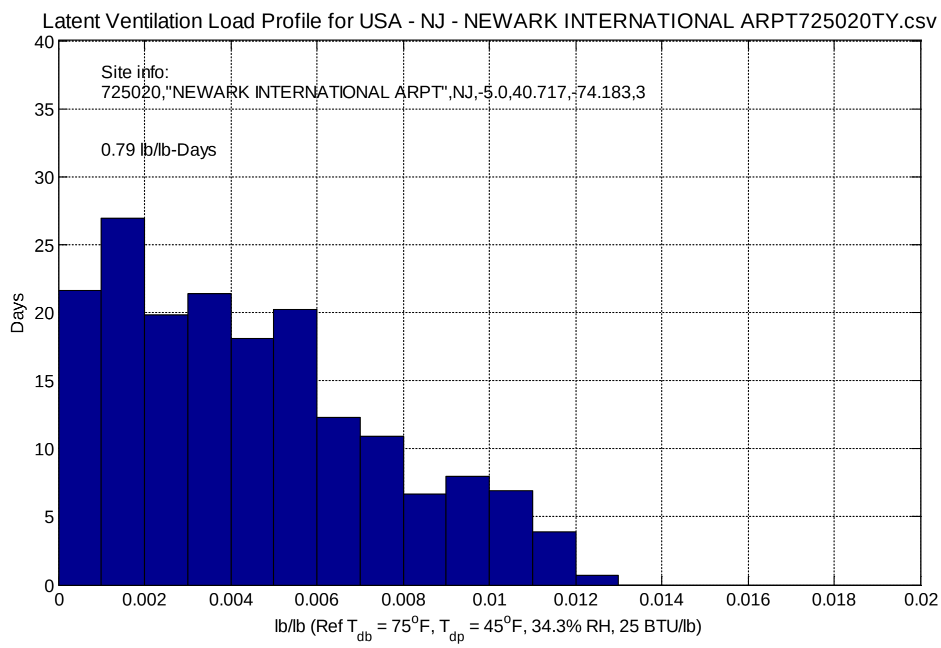

Figure 2–41 Pound-per-pound days, Hoboken, New Jersey

(Credit: Eric Kozubal/NREL)

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Table 2–16 Annual Dehumidification Loads—Newark, New Jersey (75°F DB, 45°F DP reference)

Specification	Load
Total ventilation load (Btu/lb-days)	1,791
Total moisture load (lb/lb-days)	0.79
Estimated hours of operation (h)	4,260
ASHRAE 1% design conditions (DP; HR; MCDB)	73.5°F; 124.7 gr/lb; 80.8°F

2.3.2 LDAC Design

The LDAC system at the Schaefer Center was installed as a retrofit project and was commissioned in August 2012. The system was designed to condition 100% recirculation air, which was supplied to the air jets along the perimeter windows. The LDAC ran continuously because the continuous evaporation of pool water kept the dehumidification loads relatively constant. LDAC system specifications are listed in Table 2–16.

This application offered the opportunity to evaluate the feasibility of synergistic operation of the LDAC combined with a pool heater. The LDAC requires a water sink to reject the heat of absorption and the pool requires continuous heating to maintain a comfortable temperature. At the surface of the pool, evaporation effectively creates latent loads in the air and removes sensible energy from the pool. In contrast, the LDAC removes latent loads from the air and generates sensible energy that can be directed back to the pool water. The Schaefer Pool demonstration allowed NREL to assess the relationship between these two complementary processes. The LDAC system provides all the latent cooling to the swimming pool area, which directly reduces the load on the chilled water coil that previously handled both sensible and latent cooling. Furthermore, the integrated system design uses the pool water to remove the heat of absorption in the LDAC conditioner, thereby reducing the energy to heat the pool and the energy to cool the supply air from the LDAC.

Another synergy arose from the fact that the air distribution system supplies jets of air near the external windows to prevent condensation, admit light, and allow unobstructed views. To do this, supply air in the jets near windows needs to be maintained either very dry or very hot, or a combination of both. The deeply dried air supplied by the LDAC system to the perimeter air jets effectively eliminated condensation on the windows and eliminated nearly all the AHU heating energy required by the air jets.

Lastly, this demonstration provided an opportunity to demonstrate the LDAC's ability to use waste heat or heat generated as part of a combined heat and power (CHP) cogeneration system to regenerate the desiccant. Heat recovery from a gas cogeneration system is used in this case. A heat exchanger transfers 100% of the energy needed to maintain the hot water supplied to the regenerator from the CHP system. The LDAC unit is shown in Figure 2–40 and a generalized schematic of this integrated system is shown in Figure 2–41.

The installed cost of the LDAC system is listed in Table 2–17. Because the system uses waste heat from the facility's gas cogeneration system for regeneration energy and the pool water as the source of cooling water, the conditioner and regenerator were protected from high water pressure above 30 psi by using a counterflow heat exchanger (a hot water heat exchanger is not shown in Figure 2–41).

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Figure 2–42 LDAC unit in the basement of the Schaefer Pool facility

(Credit: AIL Research, with permission)

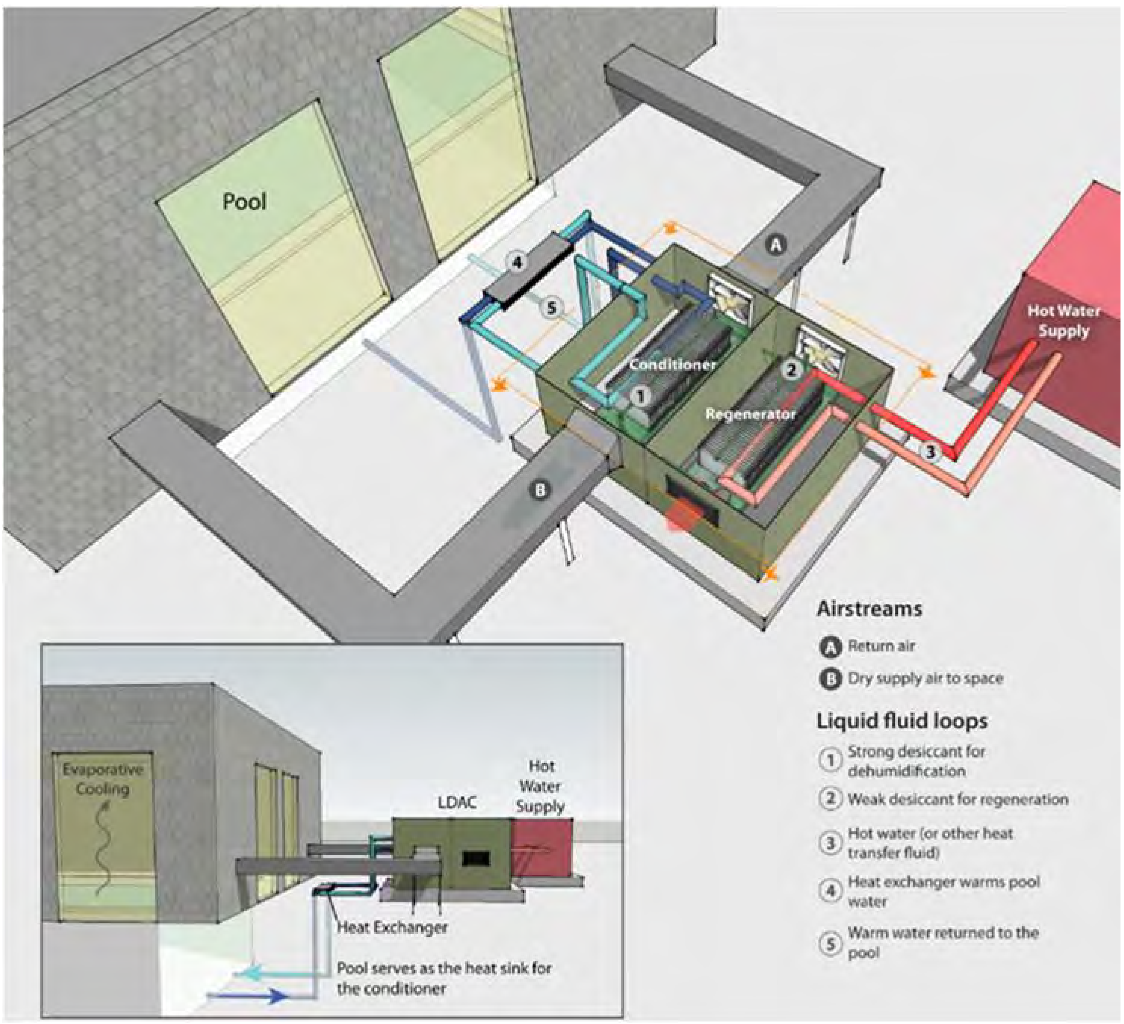

Figure 2–43 Schematic of an LDAC integrated with a pool facility

(Credit: David Goldwasser, Marjorie Schott/NREL)

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Table 2–17 Schaefer Pool—LDAC Description

Specification	Design
Latent cooling capacity	11 tons
Recirculation airflow rate	3,000 cfm
Air DP supplied by LDAC	38°F
Liquid desiccant concentration	38%–40% LiCl
LDAC latent COP (estimated)	0.76–0.79

Table 2–18 Schaeffer Pool—LDAC System Installed Cost

Component	Cost ($)
LDAC	67,000
Heat exchanger on conditioner loop	2,000
Heat exchanger on regenerator loop	2,000
Outbound freight cost	3,500
Installation (labor)	42,000
Total cost	109,000

2.3.3 Performance Results

In general, the LDAC performed very well at the Schaefer Natatorium. Pool space conditions and water temperature were maintained at desirable levels, and secondary benefits were gained from the synergistic relationship between pool operation and LDAC operation. In August 2012, the facility manager reported the LDAC system was performing better than expected. Reports from pool occupants were outstanding. The LDAC was providing all the pool heating (no auxiliary heating required), space conditions were in the desirable range, and the vapor compression AHU was doing minimal dehumidification. Outdoor conditions during this time were hot and humid. The average indoor and outdoor HR was 94 and 85 gr/lb (65°F and 63°F DP); the LDAC delivered air down to 40 gr/lb (41°F DP) at an average electrical COP equal to 8.0. Latent capacity peaked at 8.2 tons. The performance period presented below includes operation between August 12 and September 15, 2012.

Metrics used to assess the performance of the LDAC in the Schaefer Pool facility were somewhat different from those in other types of spaces:

- Although grocery stores often benefit from very dry conditions, especially near refrigeration equipment, ideal pool humidity levels need to be kept in a relatively narrow band to prevent excessive evaporation and maintain comfortable and healthy conditions. ASHRAE (2011) recommends a range of 50%–60% RH.
- Besides controlling air conditions, the pool water needed to be kept near its desired temperature of 80°F.
- Energy savings beyond those found in other building types were expected, owing to the complementary operational characteristics of the LDAC and the pool.

The psychrometric chart in Figure 2–42 shows how much latent conditioning the LDAC achieved. The system consistently provided air at 20%–30% RH; space RH was below 70% most of the time (see Figure 2–43). Space DBs were maintained at roughly 82°F. The LDAC system was designed to provide 100% of the pool water heating. Figure 2–44 shows the distribution of

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the pool water temperature over the same period; the LDAC maintained the pool water temperature at or above 80°F at all times, without an auxiliary heater.

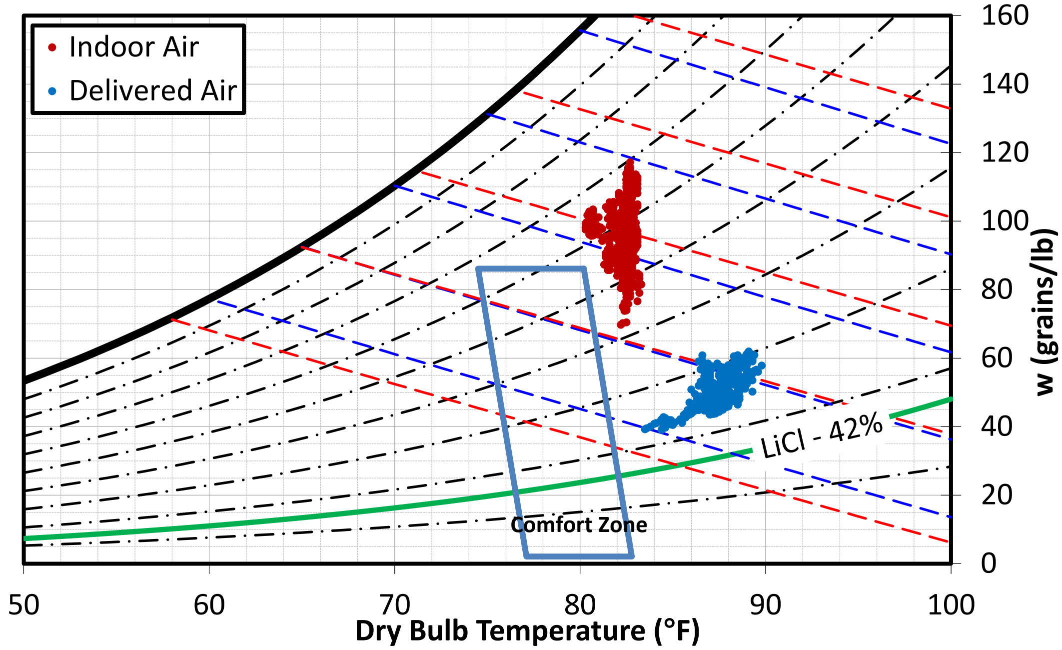

Figure 2–44 Indoor air and delivered air plotted on a psychrometric chart, Schaeffer Pool, Stevens Institute, August–September 2012

(Credit: Eric Kozubal/NREL)

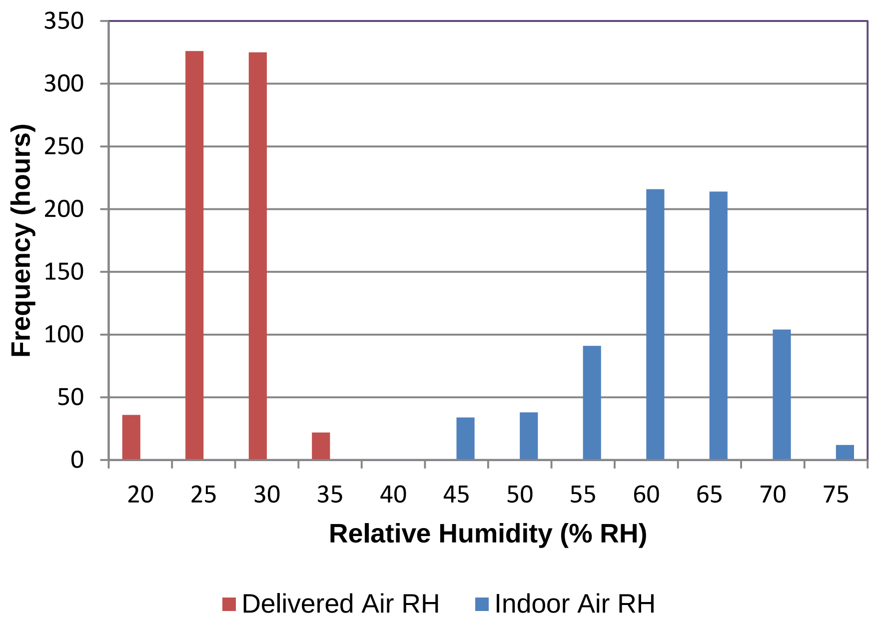

Figure 2–45 Histogram of air conditions, Schaeffer Pool, Stevens Institute, August–September 2012

(Credit: Joe Ryan, with permission)

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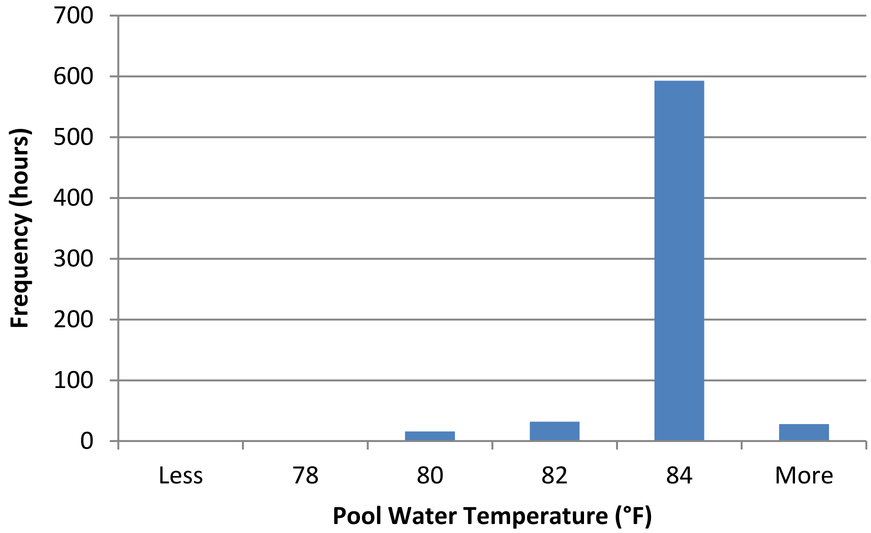

Figure 2–46 Pool water temperature, Schaeffer Pool, Stevens Institute, August–September 2012

(Credit: Jordan Clark/University of Texas, with permission)

Finally, the applicability of the CHP integration with the LDAC was tested by monitoring the water temperature delivered to the regenerator. Figure 2–45 shows that the system was able to deliver a temperature in the desirable range (180°–200°F) about 78% of the time. During other times, the other loads being served by the hot water loop issuing from the CHP system were great enough that the heat being transferred to the regenerator hot water loop wasn't sufficient to maintain a desirable temperature. The LDAC's humidity removal is degraded by about 40% when the supplied water temperature is reduced from 180°F to 120°F (see Figure 2–46). An auxiliary boiler was installed late in the 2013 cooling season to provide additional water heating. This is discussed in more detail below.

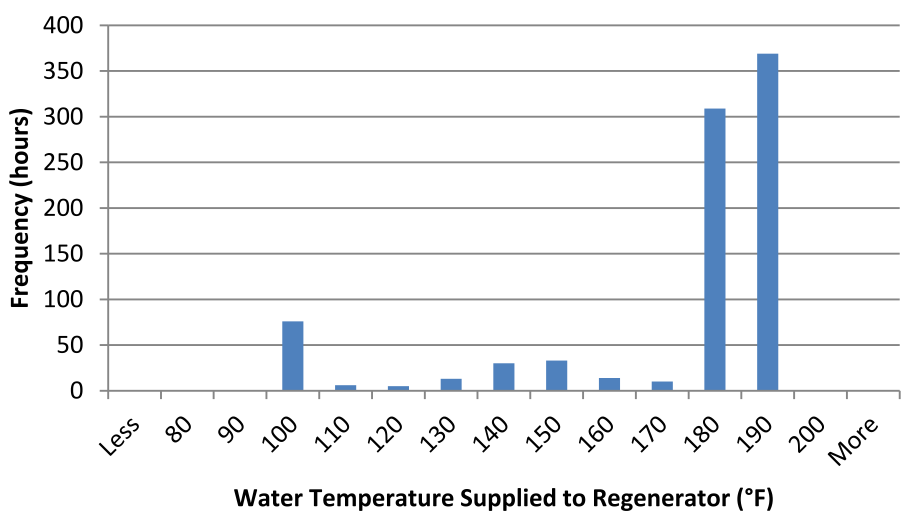

Figure 2–47 Hot water supply temperature, Schaeffer Pool, Stevens Institute, August–September 2012

(Credit: Jordan Clark/University of Texas, with permission)

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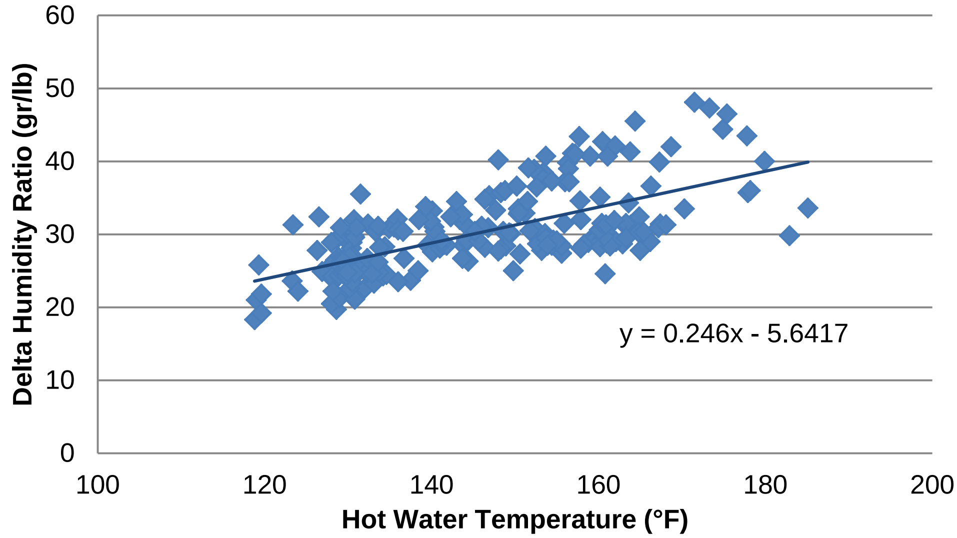

Figure 2–48 Delivered air delta humidity versus supply hot water temperature, Schaeffer Pool, Stevens Institute, August–September 2012

(Credit: Eric Kozubal/NREL)

Table 2–19 Schaeffer Pool—Key Performance Metrics, August–September 2012

Performance Metric	Average	Range
Delivered air humidity (gr/lb)	50	45–55
MRC (lb/h)	73	64–82
Electric power (kW/ton)	0.45	0.39–0.51

2.3.4 Operational Issues and Future Considerations

One issue at the Schaeffer Natatorium was that a precipitate formed in the desiccant. The precipitate led to some blockage in the heat exchanger, causing hot desiccant to bypass the heat exchanger and drain into the sump. When this happened, hot desiccant temperatures would cause the LDAC system to trip off. Once the desiccant returned to a normal operating temperature, the system could be manually restarted. The likely source of the precipitate was chlorine gas vaporizing from the pool, but the chemical makeup has not yet been verified. The data show that use of LDAC in pools is very promising; however, the precipitate problem needs to be better understood and solved before LDAC can be widely used in this application.

In response to the low humidity level, the facility manager opted to increase the space temperature in an effort to save cooling energy. This was possible because the human body feels comfortable at higher DBs when humidity is relatively lower. The facility manager could have also turned off the LDAC for periods of time and allowed the RH to float upward toward a value of around 60%, the maximum recommended (ASHRAE 2013). This would reduce the evaporation rate in the pool, and thus the heating loads on the pool and the load on the LDAC. Further research into the optimal control strategy for LDAC and associated systems in pool applications is recommended because of the potential for increased energy savings at no additional cost.

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As mentioned before, insufficient heat was sometimes transferred from the CHP system to the regenerator. This could have been eliminated with a better holistic design and control strategy. Measured data show that the water in the CHP loop during the times when it was not performing sufficiently was still hot enough to maintain acceptable regeneration temperatures (higher than 170°F). However, flow rates were set and the heat exchanger was sized to operate with CHP temperatures of around 195°F. If the flow rate in the heat exchanger were modulated in response to temperature swings in the CHP loop, a much more effective operation could have been achieved.

2.4 Lawrence T. Babbio Center – Stevens Institute of Technology

2.4.1 Building Description

The Lawrence T. Babbio Center is a six-story, 95,000-ft^2 facility at Stevens Institute of Technology that functions as the headquarters for the Wesley J. Howe School of Technology Management (2012) (see Figure 2–47). The center contains a variety of spaces, including:

- 14 classrooms
- 125-seat auditorium
- Atrium
- Six conference centers
- Business research and computer laboratory
- 10 student breakout areas
- Academic offices
- Flexible development space.

Figure 2–49 Babbio Center at Stevens Institute
(Credit: AIL Research, with permission)

The Babbio Center requires cooling for approximately 6 months of the year, which was provided by two large AHUs (designated HVAC-1 and HVAC-3) and one packaged RTU (designated HVAC-2). Table 2–19 lists the airflow rates and cooling capacities for these units. HVAC-1 serves approximately one third of the building and is located in the basement mechanical room. HVAC-2 serves the kitchen, dining area, and lounge. HVAC-3 serves the remaining two thirds of the building. The original HVAC system used overcool-and-reheat techniques to manage the entire latent load, which made this building a good candidate for LDAC. A two-stage absorption chiller running on high-pressure steam provided chilled water to the HVAC-1 and HVAC-3 cooling coils (see Table 2–20). The AHUs delivered saturated air to a variable air volume system, which reheated the air as needed to maintain DBs within a comfortable range; a gas-fired boiler provided hot water to the reheat coils in each variable air volume box.

Table 2–20 Babbio Center—HVAC System Airflows and Cooling Capacities

Component	Specification		
	Total Airflow (cfm)	**OA Flow (cfm, % total flow)**	**Cooling Capacity (tons)**
HVAC-1	21,470	8,205, 38%	85
HVAC-2	6,025	3,270, 54%	25
HVAC-3	88,510	21,085, 24%	292

Table 2–21 Babbio Center—HVAC System Design Temperatures

Specification	Design
Chilled water set point temperature	45°F
Cooling coil supply air set point temperature	51°F saturated

In the Babbio Center, the LDAC system was designed to condition about 70% of the OA supplied to the AHU designated HVAC-1, which serves office spaces, classrooms, conference rooms, restrooms, utility closets, corridor and lobby spaces, and a laboratory. The dehumidified ventilation air from the LDAC system and the return air remain unmixed through HVAC-1 to maximize the latent cooling of the return air provided by the cooling coil. This configuration maximizes the total dehumidification of the combined LDAC and AHU system. The climate analysis for this location is presented in Section 2.3.1.

The design of this system includes a single-stage regenerator, which implements a new type of experimental scavenging air regenerator called a wicking fin regenerator (see Figure 2–48). The technology removes the plastic flow passages for the heat exchanger fluids and replaces them with a eutectic copper-nickel alloy, which is resistant to the long-term corrosion effects of halide salt liquid desiccant (LiCl). The wicking fin regenerator uses a wicking medium between tube rows that provides more surface area for heat and mass transfer. Because the fluids are in tubes that can withstand a pressure much greater than 100 psi, these exchangers can more easily be placed in buildings with a central chilled, cooling tower, or hot water system without the need to install a liquid pressure isolating heat exchanger. The more expensive tube material is offset by a more compact design and simpler balance of system components. This regenerator design is expected to perform like the standard regenerator, except that it should eliminate leakage problems and lower costs.

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Figure 2–50 Wicking fin regenerator
(Credit: AIL Research, with permission)

Design specifications are listed in Table 2–21. Figure 2–49 and Figure 2–50 show the regenerator and conditioner in the mechanical room. The installed costs of the LDAC system are listed in Table 2–22. Because the system uses the facilities central cooling and boiler system, the installation was simplified by using a heat exchanger between the chilled water source and the conditioner loop, and the hot water source and the regenerator loop and reduced the overall cost of the system.

Table 2–22 Babbio Center—LDAC Description

Specification	Design
Latent cooling capacity	25 tons
OA flow rate for space ventilation	6,000 cfm
Air RH supplied by LDAC	20%
Desiccant concentration	36%–40% LiCl
Single-stage LDAC latent COP (gas basis)	0.7

Figure 2–51 LDAC regenerator and conditioner in the mechanical room of the Babbio Center
(Credit: AIL Research, with permission)

Figure 2–52 LDAC conditioner in the mechanical room of the Babbio Center
(Credit: AIL Research, with permission)

Table 2–23 Babbio Center—LDAC System Installed Costs

Component	Cost ($)
LDAC	41,000
Heat exchanger on conditioner loop	2,000
Heat exchanger on regenerator loop	2,000
Outbound freight cost	3,500
Installation (labor)	45,000
Total cost	86,000

2.4.2 Performance Results

Installation of the LDAC system at the Babbio Center was delayed until July of 2013, too late to collect performance data for this report. Delays were caused by Hurricane Sandy and unanticipated high-priority repair and maintenance needs that temporarily diverted resources from the LDAC installation project.

2.4.3 Operational Issues and Future Considerations

The LDAC manufacturers and facility managers reported that the unit at the Babbio Center had successfully completed startup during the last week of July 2013. Shortly thereafter, the manufacturers discovered the unit was underperforming and could not meet the target airflow and RH goals. The system was designed to supply 6,000 cfm of air at 19% RH, but the unit was supplying roughly 4,000 cfm of air at 20%–28% RH. The source of this problem was diagnosed as an undersized heat exchanger installed between the LDAC and hot and cold water supplies. The LDAC manufacturer is working with the facility manager to replace the heat exchanger with one of the proper size.

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3 Analysis of LDAC Applicability for Supermarkets by Climate

3.1 Background

Energy modeling of a typical supermarket was used to estimate the general energy savings potential of LDAC systems in seven regions of the United States. Of the three LDAC demonstration categories (supermarkets, pool facilities, and multipurpose campus facilities), the supermarket building type was chosen for analysis because of its high potential for energy savings, and its broad applicability nationwide.

A variation of the “new construction” supermarket reference building model for EnergyPlus version 8.0 was used as the starting point for model development. This model was previously created based on 2003 Commercial Buildings Energy Consumption Survey data and additional research carried out by NREL and Pacific Northwest National Laboratory; other inputs refer to ASHRAE Standards 90.1and 62.1 (Deru et al. 2011). Several modifications made for this analysis, including: (1) the refrigeration system was replaced with one developed by NREL based on measured data from an existing Walmart located in Centennial, Colorado; (2) the HVAC system was replaced with one including dehumidification capabilities; and (3) ventilation requirements and exhaust flow rates were updated to conform with ASHRAE Standard 62.1-2007. The following sections detail the LDAC modeling approach, the building model inputs, and the results of the energy savings and economic feasibility study for seven relevant climate zones.

3.2 Energy Modeling Approach

3.2.1 Relevant Climate Zones for Modeling

The United States can be divided into eight climate zones ranging from hot (zone 1) to severe cold (zone 8). Figure 3–1 shows the seven primary ASHRAE climate zones (excluding extreme zone 8 regions in Alaska). The zones are based on a range of heating and cooling degree days and are divided further into three subcategories: moist (A), dry (B), and marine (C). LDAC technology is potentially applicable in A- and C-type climate subcategories. Humid climate subcategories (A-type subcategories) and one marine climate were selected for the energy and economic analysis because dehumidification is generally required in these subcategories for part or all of the year. The representative cities for these climate zones are listed in Table 3–1.

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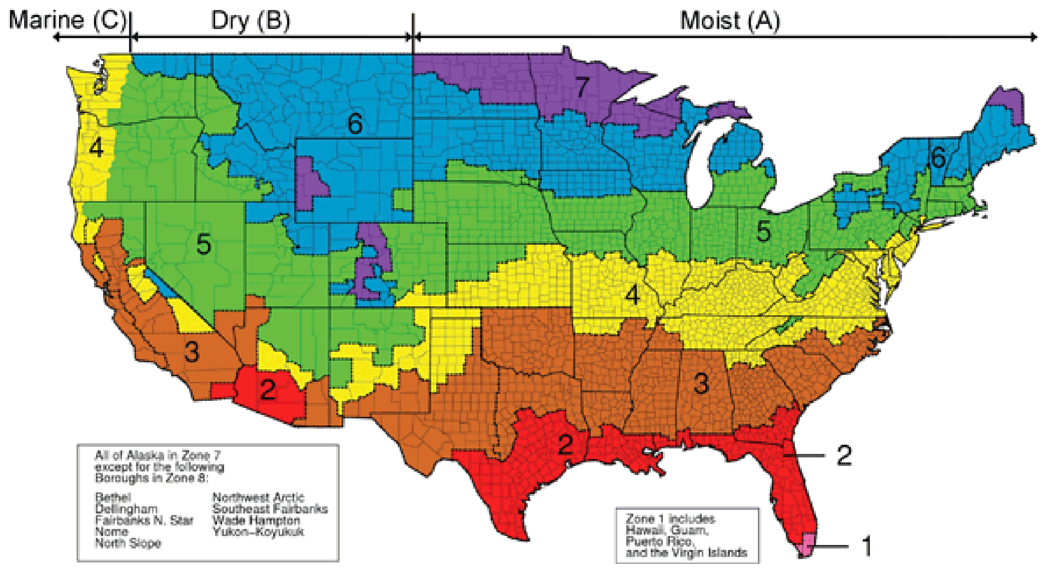

Figure 3–53 U.S. climate zone map
(Credit: DOE 2004)

Table 3–24 Relevant U.S. Climate Zones

Climate Zone and Subcategory	Representative City	Climate
1A	Miami, Florida	Hot-humid
2A	Houston, Texas	Hot-humid
3A	Atlanta, Georgia	Hot-humid
3B	Long Beach, California	Marine
4A	Baltimore, Maryland	Mild-humid
5A	Chicago, Illinois	Cold-humid
6A	Minneapolis, Minnesota	Cold-humid

Total and latent loads for these locations were estimated with metrics similar to cooling degree days that use enthalpy and HR. Whereas cooling degree days use a balance temperature representative of a building's total sensible cooling load to estimate the cooling requirement, enthalpy-days (Btu/lb-days) and HR-days (lb/lb-days) address the latent cooling load. In this analysis we considered only the latent load imposed by the ventilation air, because in supermarkets internally generated latent load is limited to that from occupants and produce. These metrics were calculated using TMY3 weather data to determine the magnitude of the annual difference between the outdoor conditions and a particular reference point of 75°F DB and 45°F DP (NREL 2008). The reference point refers to the condition of the air as it leaves the LDAC conditioner component. The resulting Btu/lb-days and lb/lb-days are listed in Table 3–2. The estimated hours of dehumidification, also listed in Table 3–2, indicate the number of hours ambient humidity levels exceed the reference condition.

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Table 3–25 Calculations of Total Load and Ventilation Load for Representative Cities

Climate Zone, Subcategory, and Representative City	Total Ventilation Load (Btu/lb-days)	Total Moisture Load (lb/lb-days)	Estimated Hours of Operation
1A: Miami	8,003	3.01	8,347
2A: Houston	7,331	2.72	8,753
3A: Atlanta	5,520	2.13	6,912
3B: Long Beach	3,088	1.28	5,408
4A: Baltimore	1,359	0.94	7,213
5A: Chicago	2,144	0.92	4,258
6A: Minneapolis	1,791	0.79	4,260

3.2.2 Baseline Building and HVAC Description

The supermarket model is a 45,000-ft^2, single story, six-zone building and includes a sales floor, bakery, deli, produce section, dry storage area, and office space (see Figure 3–2 and Figure 3–3). The envelope construction and fenestration comply with ASHRAE Standard 90.1-2004 for each humid climate subcategory (ASHRAE 2004) (see Table 3–4 and Table 3–5 for construction and fenestration properties, respectively). Building loads in each zone include people, lights, and electrical equipment; the deli and bakery zones also include gas-use equipment (see Table 3–6).

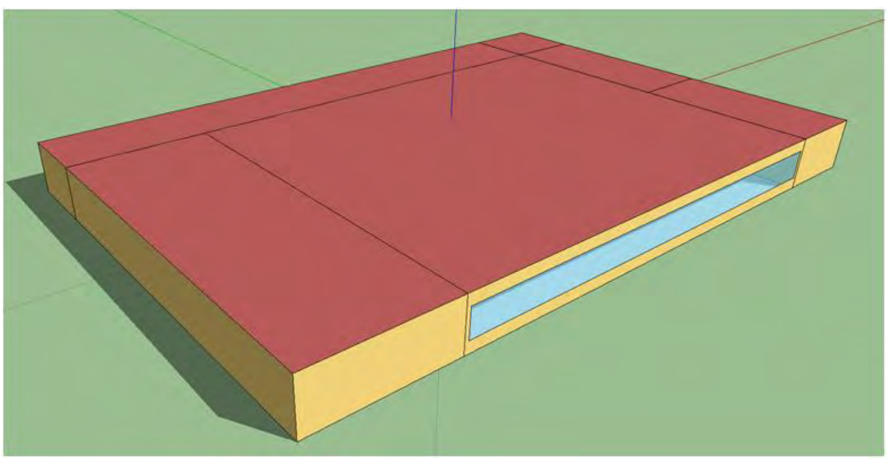

Figure 3–54 Supermarket model rendering

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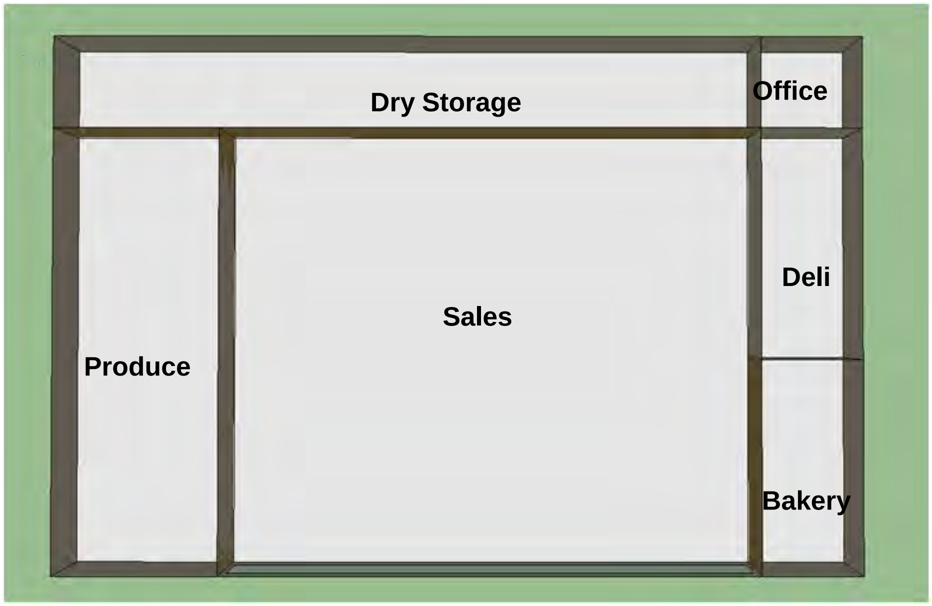

Figure 3–55 Supermarket model floor plan

Table 3–26 Zone Area

Zone	Area ft^2	Percent of Total
Office	956	2
Dry storage	6,694	15
Deli	2,418	5
Sales	25,025	56
Produce	7,657	17
Bakery	2,250	5
Total	45,000	100

Table 3–27 Construction Types and R-Values ($h{\cdot}ft^2{\cdot}°F/Btu$)

Location	Roof Insulation Above Deck	Mass Walls	Slab-on-Grade Floor
1A: Miami	15.8	2.4	1.8
2A: Houston	15.8	2.4	1.8
3A: Atlanta	15.8	6.6	1.8
3B: Long Beach	15.8	6.6	1.8
4A: Baltimore	15.8	6.6	1.8
5A: Chicago	15.8	8.1	1.8
6A: Minneapolis	15.8	9.6	1.8

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Table 3–28 Fenestration Properties

Location	U-Value (Btu/h·ft^2·°F)	SHGC* (%)
1A: Miami	1.0	0.25
2A: Houston	1.0	0.25
3A: Atlanta	0.56	0.25
3B: Long Beach	0.56	0.25
4A: Baltimore	0.56	0.39
5A: Chicago	0.56	0.39
6A: Minneapolis	0.56	0.39

* Solar heat gain coefficient

Table 3–29 Internal Loads

Space Type	People (ft^2/person)	Lighting (W/ft^2)	Electric Plug and Process Loads (W/ft^2)	Gas (Btu/ft^2)
Office	200	1.1	0.75	0
Dry storage	300	0.8	0.75	0
Deli	125	1.7	5	8.53
Sales	125	1.7	0.5	0
Produce	125	1.7	0.5	0
Bakery	125	1.7	5	8.53

In the baseline building model, each zone is equipped with a unitary packaged RTU, which includes an electric DX cooling coil and a gas heating coil. Humidity is controlled by cooling the zone supply air to saturation and reheating it to an appropriate zone supply air temperature when the space calls for dehumidification. Four reheat strategies are compared to identify a range of potential savings, including:

- Case 1: Natural gas reheat coils
- Case 2: RTU condenser hot-gas reheat with auxiliary natural gas reheat
- Case 3: Electric reheat coils
- Case 4: RTU condenser hot-gas reheat with auxiliary electric reheat.

See Table 3–7 for HVAC inputs. OA supply and exhaust are operated during building occupied hours (6:00 a.m. to 10:00 p.m.). Two zones, the deli and the bakery, have exhaust requirements because they contain exhaust hoods for cooking equipment (Table 3–8). About 70% of the makeup air for the deli and bakery is transferred from the sales zone; the remainder is brought in through the unitary systems that serve the deli and the bakery. (This makeup air is an addition to the ventilation air provided by the unitary systems.)

The produce and sales floor includes 1,064 linear ft of refrigerated cases; walk-in freezers are located in the dry storage area (see Table 3–9). There are four racks, each with four compressors (see Table 3–10). The case ASH power is controlled as a function of ambient air DP. This

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control method varies the ASH power linearly based on the ambient air DP, the case operating temperature, and the ambient DP at which the case was rated (DOE 2012b).

Table 3–30 HVAC Properties

HVAC Property	Model Value
Average cooling coil energy efficiency ratio (Btu/W·h)	10.7
Average cooling coil COP (W/W)	3.14
Compressor/condenser combined COP	3.67
Natural gas heating coil efficiency	80%
Reheat options	
• **Natural gas reheat coil efficiency**	80%
• **RTU compressor hot-gas reheat coil utilization**	25%
• **Electric reheat coil efficiency**	99%

Table 3–31 OA Supply and Exhaust Flow Rate Requirements

	OA Supply		Exhaust
Space Type	cfm	% OA	cfm
Office	82	12	–
Dry storage	575	7	–
Deli	487	15	1800
Sales	3090	22	–
Produce	946	21	–
Bakery	487	16	1800

Table 3–32 Refrigeration System Case Length and Capacities

	Case		Walk-in Freezers	
Zone	Length (ft)	Capacity (kW)	Length (ft)	Capacity (kW)
Produce	72	29	–	–
Sales	992	247	–	–
Dry storage	–	–	5,400	92

Table 3–33 Refrigeration Rack Compressors

Rack	Number of Compressors	Evaporator Temperature
Rack A	4	Low (–29° to –9°F)
Rack B	4	Low (–29° to –9°F)
Rack C	4	Medium (5° to 25°F)
Rack D	4	Medium (5° to 25°F)

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3.2.3 LDAC Model

EnergyPlus can model a wide variety of building systems, but in cases where technologies are newer or underutilized, EnergyPlus must often be coupled with external modeling software to add customized systems. The Dymola simulation environment was used to develop the LDAC model (Dassault Systèmes 2012). This model was based on the configuration of an installed LDAC system at Tyndall Air Force Base, in Panama City, Florida (Dean et al. 2012). The mathematical algorithms used to predict the performance of the conditioner and regenerator were based on a particular LDAC system and are not necessarily expected to predict the performance of systems built by other manufacturers. The model is also flexible enough to predict system performance in a variety of climates with different specifications for pumps; fans; control strategies; heating and cooling sources; and sizes for the desiccant storage, regenerator, and conditioner.

The LDAC system and its effects on building performance and energy usage were modeled in three phases (for more details on the component- and system-level procedures, refer to Appendix A):

1. **Component-level modeling:** The heat and mass transfer in the conditioner and regenerator was modeled differently than the LDAC. The processes occurring within the plates of the conditioner are well-defined and understood, which allowed for the development of a rigorous physical model. This model was first validated with laboratory measurements and then used to generate a performance map over the entire range of operating conditions. The processes occurring in the regenerator are more complex, making it difficult to predict performance using a purely physical model. Therefore, an empirical model was developed using a map of laboratory performance data over a full range of operating conditions. Both the conditioner and regenerator performance maps agree well with laboratory data (refer to Appendix A). Other components of the LDAC system were taken from either the Modelica Standard Library or the open source Modelica Buildings Library created by the Simulations Research Group at Lawrence Berkeley National Laboratory.
2. **System-level modeling:** The performance maps of the conditioner and regenerator were then input into a system-level model containing all other necessary components using the Dymola environment (see Appendix A). Here, the annual performance of the system was modeled using TMY3 weather data for each site in the analysis. LiCl was used as the desiccant solution at a 40%–42% concentration. The hot and cold water was supplied by a natural gas boiler and a variable-flow cooling tower (with a variable-speed fan). An interchange heat exchanger exchanged sensible heat between the weak and strong desiccant streams and a stratified desiccant sump was used to allow the conditioner and regenerator to operate at different flow rates to accommodate the demand for dehumidification. Table 3–11 lists the inputs and characteristics of the system-level model of the LDAC. The LDAC system was bypassed when the ambient DB was lower than 41°F or RH was lower than 15%. The conditioner fan and pump did not operate during this time. The regenerator shut off when desiccant concentration reached 0.42 kg salt/kg solution. For further explanation, refer to the appendix.

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Table 3–34 LDAC Model Inputs

LDAC Model Input	Details
OA flow rate	• 4,036 cfm (6857 m^3/h)
Desiccant	• 40%–42% $LiCl\text{-}H_2O$
Cooling tower	• Approach: 7°F at site's design conditions • Range: 10°F at design conditions • Maximum fan power: 250 W (850 Btu/h)
Natural gas boiler	• Efficiency: 0.8 • Capacity: 110 kW (375 kBtu/h) • Hot water temperature: 176°–194°F
Interchange heat exchanger	• Effectiveness of 0.8

3. **Building-level modeling:** Output values for the processed air conditions (DB and wet bulb temperature) were fed into the EnergyPlus building model using the energy management system. The processed air temperatures replaced the OA node temperatures for the RTUs serving the produce and the sales zones. The reheat coils and humidistats were removed, as the LDAC provides all of the latent cooling.

Control strategies implemented in the system-level and building-level modeling represent the likely mode of operation, rather than those that create space conditions identical to the baseline model. Thus, space DB and RH often differ slightly between the baseline and the LDAC models as they would in an actual retrofit situation. In nearly all situations, this leads to more comfortable indoor air conditions in the LDAC model, and provides the energy and cost savings presented in Section 3.2.4. These cost savings are conservative because the baseline system was not forced to provide indoor air conditions as comfortable as the LDAC system, which would have caused the baseline system to use more energy. Further discussion follows in Section 3.3.

3.3 Economics

Energy and economic assessments of the LDAC were conducted by combining model results with pricing data. Utility tariffs were based on the average national monthly rates from January 2010 through September 2012 for electricity (EIA 2013a) and from January 2010 through July 2012 for natural gas (EIA 2013b). This strategy is used to account for price volatility rather than referring to last year's average. National average electricity and natural gas tariffs are listed in Table 3–12.

Table 3–35 National Average Electricity Tariffs ($/kWh) (EIA 2013 a,b)

Month	Electricity ($/kWh)	Natural Gas ($/1000 ft^3)
Annual average	0.102	8.84

3.3.1 Performance and Cost Analysis Results by Climate Zone

Figure 3–4 through Figure 3–7 show the annual ventilation and air conditioning source energy consumption and savings for the four variations of baseline reheat strategies. As expected, the highest source energy savings are seen in the hot-humid climate zones (1A and 2B), where humidity control is required during much of the year and where latent cooling makes up a

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significant portion of the overall energy consumption. Savings are also greatest where electric reheat is used in the baseline and least where gas reheat is used. Where gas reheat is used the natural gas consumption for desiccant regeneration negates the savings in many climate zones. Appendix B provides more detail on the end use energy breakdown for each baseline case.

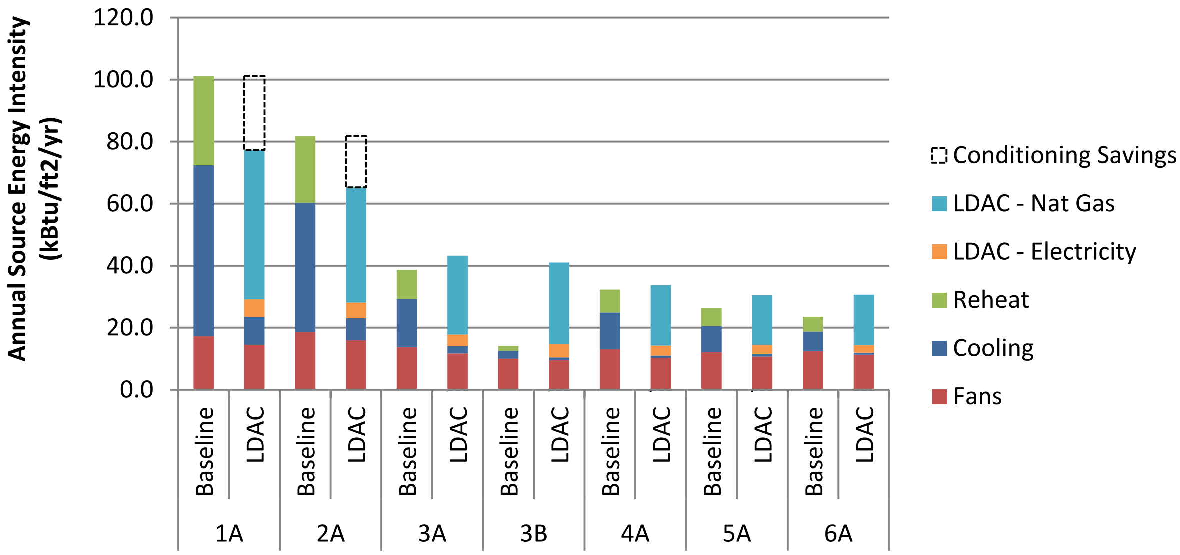

Figure 3–56 Annual ventilation and air conditioning source energy intensity and savings – natural gas reheat coils – single-stage regenerator
(Credit: Lesley Herrmann/NREL)

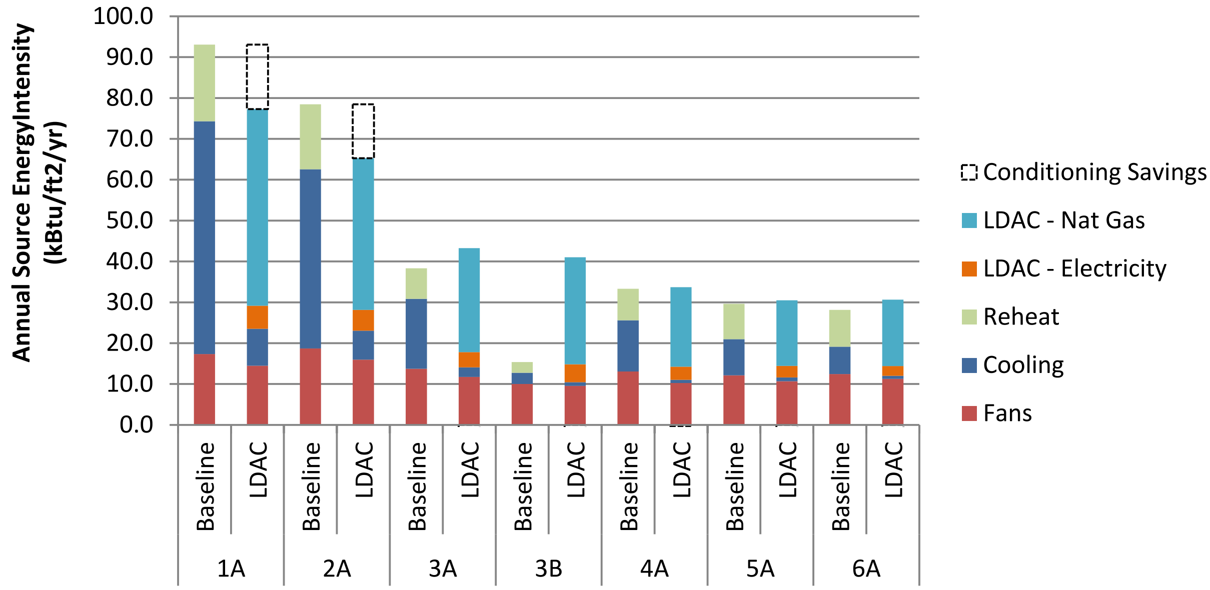

Figure 3–57 Annual ventilation and air conditioning source energy intensity and savings – RTU condenser hot-gas reheat with auxiliary natural gas reheat coils – single-stage regenerator
(Credit: Lesley Herrmann/NREL)

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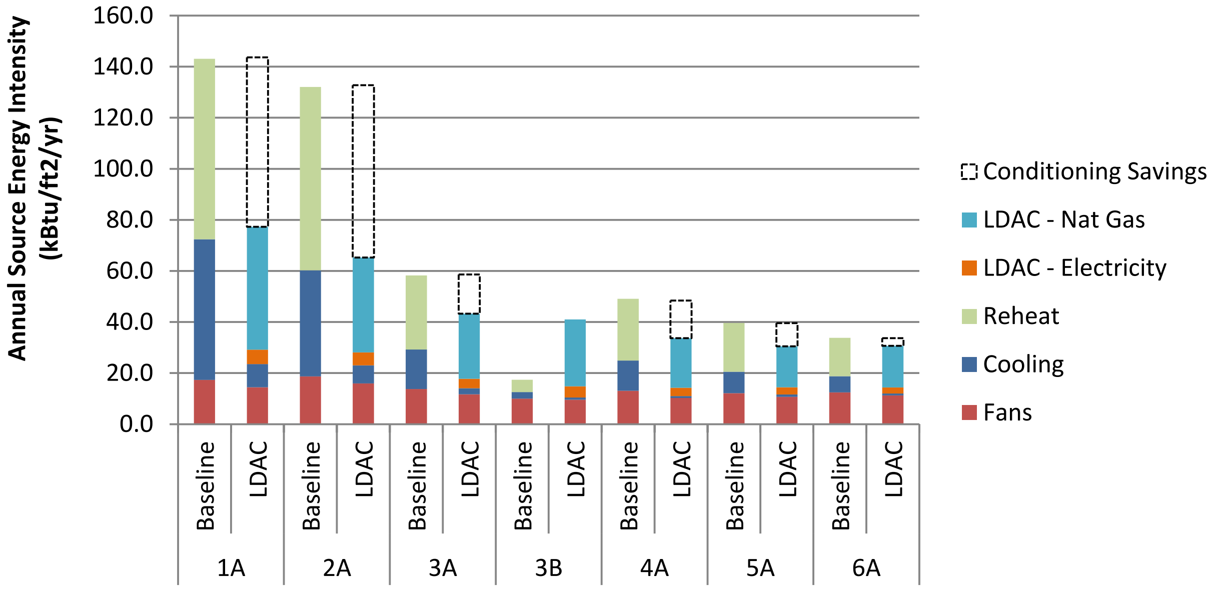

Figure 3–58 Annual ventilation and air conditioning source energy intensity and savings – electric reheat coils – single-stage regenerator
(Credit: Lesley Herrmann/NREL)

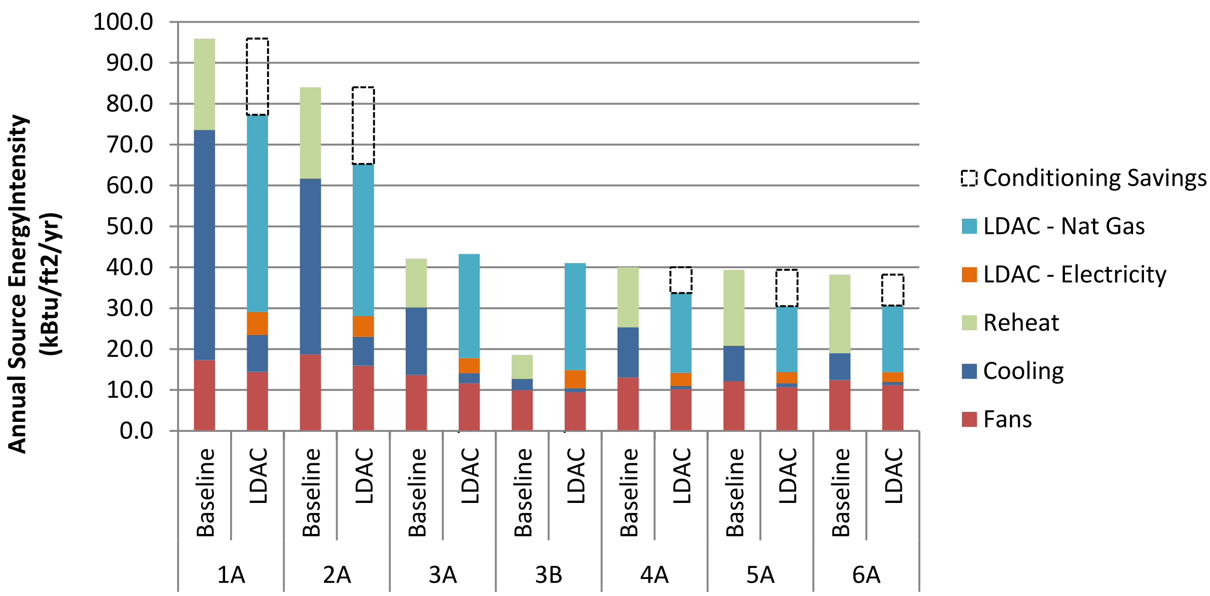

Figure 3–59 Annual conditioning source energy intensity and savings – RTU condenser hot-gas reheat with auxiliary electric reheat coils – single-stage regenerator
(Credit: Lesley Herrmann/NREL)

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A discussion of the breakdown of the energy, cost, and comfort benefits follows.

- **Heating, cooling, and fan energy savings:** Heating energy savings were negligible. Some small benefit may be gained by adding complex control strategies, which take advantage of the latent heat of vaporization generated in the LDAC conditioner during the heating season; however, this was not modeled in this work and thus heating savings are minimal. Cooling energy is reduced because the need for overcooling with the vapor compression system is eliminated by removing the latent load from the ventilation air upstream of the cooling coils. Fan energy savings are realized from fewer cooling runtime hours.
- **Lower average RH:** Table 3–13 shows the annual average RH in the zones treated by the LDAC for the case with electric reheat coils. This example shows that the RH levels are lower in the LDAC models because the LDAC was controlled to provide the driest air possible so that the refrigeration system does not waste energy dehumidifying the zone air. Although a control strategy could be devised to maintain similar RH levels for the two models, we elected to use realistic and different control strategies for the baseline and the LDAC. This resulted in conservative savings estimates because the baseline systems were not forced to produce the low humidity levels achieved with the LDAC. Lower RH levels lead to better product preservation by avoiding frost buildup on frozen foods and moisture collection in packaged baked goods.

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Table 3–36 Comparison of Average RH in the Sales and Produce Zones for the Baseline Using Electric Reheat Coils (%) (Single-Stage Regenerator)

1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
Baseline	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**
51.1	42.8	46.5	39.0	40.0	34.0	43.6	35.5	35.8	33.7	34.1	29.1	30.4	26.7

Table 3–37 Comparison of Average DBs in the Sales and Produce Zones for the Baseline Using Electric Reheat Coils (°F) (Single-Stage Regenerator)

1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
Baseline	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**	**Baseline**	**LDAC**
20.7	22.8	20.5	21.9	20.5	21.2	20.7	20.9	20.0	20.0	20.0	20.4	20.0	20.3

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- **Refrigeration energy savings:** Overall refrigeration energy savings as a percentage of whole-building energy usage are small, ranging from 1% to 4% of source energy savings, but the defrost and anti-sweat components show significant savings; defrost savings range from 12% to 23% and anti-sweat savings range from 5% to 11% for the four reheat strategies (refer to Appendix B for a breakdown of refrigeration energy savings in the Annual Whole Building Source Energy Intensity tables). The refrigeration compressor energy consumption, which dominates the total refrigeration energy use, is shown to increase in in climate zones 1A and 2A with the LDAC. This is a result of higher average indoor air temperatures (see Table 3–14). Therefore, some of the LDAC savings are negated as a result of an increase in refrigeration energy, but leads to more comfortable space conditions. The impact of drier space conditions on compressor energy use is a complex nonlinear function of several variables, and its effect was minimal for the conditions modeled. Had we forced the baseline systems to the same control strategy as the LDAC system, we would have seen refrigeration savings similar to those demonstrated in previous studies. For example, Faramarzi et al. (2000) reported a 3%–18% savings in compressor energy, a 4–5% reduction in defrost energy and a 1%–15% reduction in total refrigeration energy for a decrease in space RH from 55%–35%. Instead we opted to model in a more realistic and conservative manner by letting the baseline systems follow a typical control strategy. The baseline systems in humid climates had zone air temperatures that were somewhat chilly; the LDAC system maintained zone air temperatures that were warmer and more comfortable. When uncertainty, case-type differences, humidity level differences, and zone air temperature differences are taken into account, the savings realized in this study are comparable to the previous studies.
- **Utility cost savings:** A simple utility cost savings estimate was done using a flat rate for electricity and natural gas, which were based on the national averages of \$0.102/kWh for electricity and \$8.84/100 ft^3 for natural gas (EIA 2013 a,b). The resulting total annual cost savings are shown in Table 3–15 and are a result of load shifting from electricity to natural gas, as the cost per normalized unit of electricity is near 3.5 times more than that for natural gas. Electricity and natural gas cost savings are broken down for each baseline model in Table B–18 and Table B–19 in Appendix B. Cost savings will vary depending on local utility charges; maximum savings will occur in locations where the ratio of electricity price to natural gas price is highest. Similar to the pattern seen for source energy savings, the utility cost savings are greatest in the most humid climates and for the baseline where electricity is used for reheat. The source energy savings and utility cost savings are greater for an LDAC with a two-stage regenerator as seen in Figure 3–5 and Table 3–17.

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Table 3–38 Annual Energy Cost Savings ($1,000/yr) (Single Stage Regenerator)

Reheat Strategy	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
Case 1: Nat Gas	11	8	2	-4	5	1	0
Case 2: RTU Cond. + Nat Gas	3	3	0.2	-4	4	2	1
Case 3: Electricity	29	27	10	-2	11	6	4
Case 4: RTU Cond. + Electricity	10	9	3.8	-2	8	6	6

- **Incremental cost analysis:** Low-flow LDAC is an emerging rapidly evolving technology and is therefore not yet mature enough to allow a detailed economic analysis. However, the incremental cost target for the LDAC was determined based on 3-year, 5-year, and 10-year simple payback periods. Table 3–16 lists the incremental costs associated with the baseline case using hot-gas reheat from the RTU with auxiliary natural gas reheat. This case results in the lowest annual energy cost savings (see Table 3–15), so the incremental costs listed in Table 3–16 are the most conservative of the four cases. The negative values for Long Beach indicate that LDAC is not a favorable option for that climate. Some coastal microclimates have higher humidity and therefore may be appropriate for an LDAC. However, many California coastal climates are similar in humidity to Long Beach. Table 3-17 below shows the incremental cost targets for an LDAC with a two-stage regenerator. These incremental cost targets are higher than for the single-stage regenerator and show the benefit of developing the two-stage regenerator for the LDAC. Refer to Appendix B for other incremental cost values.

Table 3–39 Incremental LDAC Cost Compared to Baseline With RTU Hot-Gas and Auxiliary Natural Gas Reheat ($) (Single-Stage Regenerator)

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	8,522	9,652	657	(10,515)	11,215	4,569	1,995
5-year	14,204	16,086	1,095	(17,524)	18,691	7,615	3,325
10-year	28,408	32,172	2,191	(35,049)	37,382	15,229	6,649

3.4 Discussion

Overall applicability of the LDAC for a particular climate can be understood as one of four situations:

1. In hot-humid climates such as 1A and 2A, baseline cooling energy is dominated by latent loads, including a large penalty for reheat (20%–48% of HVAC energy). In these climates, the LDAC is particularly well suited, because it removes 100% of the latent loads upstream of the cooling coil and eliminates the large overcool-reheat energy used by the vapor compression system. Additional natural gas required to regenerate the desiccant is more than compensated by cooling energy savings, and HVAC source energy

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savings are on the order of 12%–40% for each baseline reheat strategy (see Figure 3–4 through Figure 3–7). Additional, smaller savings are realized in refrigeration, nearly all of which come from reductions in defrost and anti-sweat energy usage (refer to Appendix B). Additional benefits from reduced frost buildup in cases and better comfort from changes in space conditions are also realized, but harder to quantify in terms of energy and costs. Because such a great quantity of energy usage is shifted from electricity to gas in these climates, large cost savings are achievable: 1%–9% of the yearly energy cost expenditure of the whole building.

2. The use of RTU condenser hot gas with auxiliary natural gas reheat in these climate zones most efficiently meets the humidity set point.
3. Cold-humid climates such as 5A and 6A are less applicable for the LDAC, as the sensible heating dominates the HVAC energy expenditure. The LDAC retrofitted system, however, minimally reduced energy usage and costs in the baseline case using electric reheat coils (with and without the use of RTU condenser hot-gas reheat). See Appendix B for details.
4. Although mild, marine climates such as 3B have some latent loads, the LDAC was not applicable in these climates under any baseline scenario. Considerable savings were shown in refrigeration energy (more than any other climate modeled) owing to the greatest reduction in space humidity; however, the amount of natural gas needed to regenerate the desiccant outweighed the benefits. This confirms NREL’s previous understanding that the LDAC is most beneficial in climates with large latent loads and low sensible HRs during the cooling season rather than smaller, constant latent loads throughout the year (as in Long Beach).

The energy and utility cost savings presented here are conservative. Several improvements may be made to the design and control of the LDAC system to realize additional benefits:

- The thermal energy requirement for regeneration can be provided more efficiently, thus reducing the LDAC’s natural gas consumption. This analysis assumes an 80% efficient natural gas boiler and a single-stage regenerator. Greater energy savings are achievable with a two-stage regenerator, which is predicted to use 40% less natural gas in the regeneration process (Lowenstein 2013). In this case, HVAC savings in hot humid climates are 34%–57% (see Figure 3–8) (assuming the same 80% efficient natural gas boiler is used) with corresponding net utility cost savings of $3,000–$36,000/year, depending on reheat strategy type (minimal economic savings are realized in Long Beach) (see Table 3–16). As a result of the reduction in natural gas consumption, the incremental cost of the LDAC with a two-stage regenerator increases compared to the LDAC with a single-stage regenerator (see Table 3–18). Refer to Appendix B for a breakdown of energy and cost savings for each reheat strategy.

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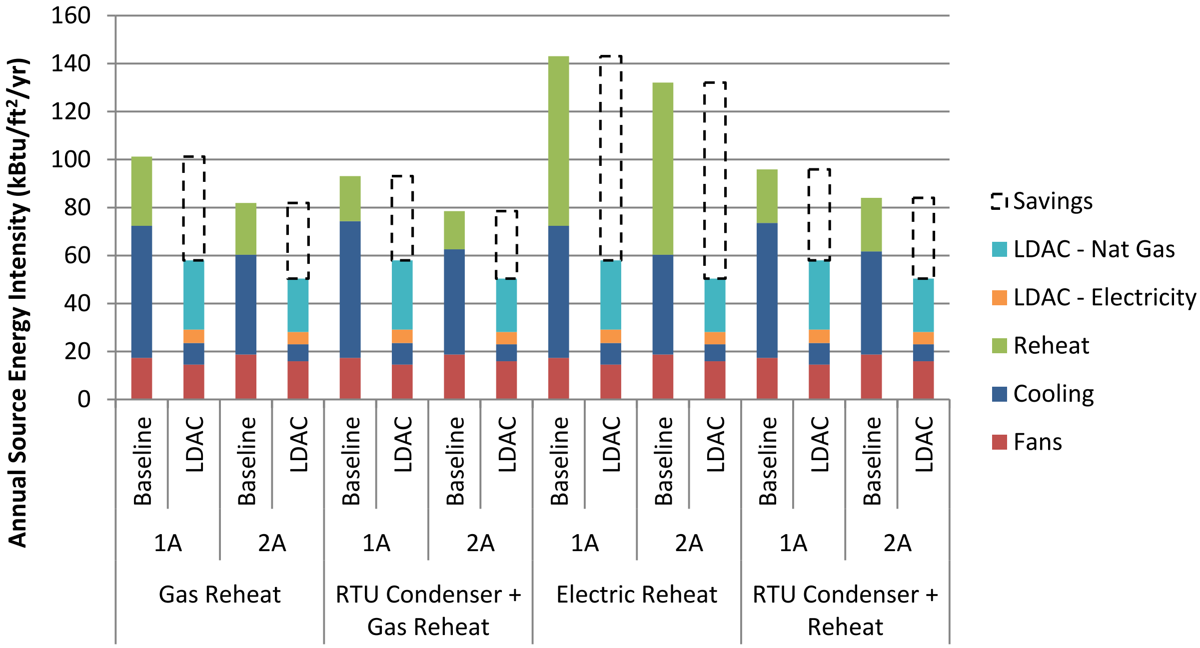

Figure 3–60 Annual source ventilation and air conditioning energy consumption and savings – two-stage regenerator (kBtu/ft^2/yr)
(Credit: Lesley Herrmann/NREL)

Table 3–40 Annual Energy Cost Savings ($1,000/yr) (Two-Stage Regenerator)

Reheat Strategy	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
Case 1: Nat Gas	18	13	6	0	7	3	2
Case 2: RTU Cond. + Nat Gas	10	9	4	0	7	4	3
Case 3: Electricity	36	32	14	1	14	9	6
Case 4: RTU Cond. + Electricity	22	19	9	2	12	10	9

Table 3–41 LDAC Incremental Cost Target – RTU Condenser Hot-Gas Reheat With Auxiliary Electric Reheat Coils – Two-Stage Regenerator

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	50,180	42,898	22,422	6,151	31,702	26,924	25,112
5-year	83,633	71,497	37,370	10,252	52,837	44,873	41,853
10-year	167,266	142,994	74,739	20,503	105,675	89,745	83,706

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- Some situations present options for using alternative sources of thermal energy that would reduce or eliminate natural gas consumption for regeneration. These include solar heaters, waste heat from HVAC and refrigeration condensers, waste heat from other sources such as kitchen exhaust, auxiliary supermarket equipment, and CHP.
- Control of the LDAC system is complicated and beyond the scope of this work, but many improvements may be realized by optimizing the interactions of the various components. For example, operation of the LDAC without liquid cooling (such as from a cooling tower) that allows the LDAC to dehumidify and heat the air during times when latent loads exist and the building requires heating, may eliminate some of the heating energy requirements. Conversely, modulating cooling tower fan power and altering cooling tower size to take maximum advantage of free cooling (and ensuring this is not negated by additional fan power) may result in additional cooling savings. More sophisticated strategies for desiccant regeneration may yield incremental benefits as well, including running the conditioner at lower desiccant concentrations and regenerating at lower temperatures when OA humidity is lower. Lastly, load-shifting by regenerating at times when free energy is available or utility costs are lower may be a means of saving additional energy and money. Additional modeling studies are recommended to gain a better understanding of the impact of these control strategies.

At this time, an accurate market cost for LDAC systems of this type cannot be determined, as only a few of the systems exist, so a more in-depth economic analysis is not available. However, the cost savings from reduced cooling coil sizes and the elimination of reheat coils can help to offset the capital cost of the LDAC system.

In retrofit situations, building owners should include estimates for minor modifications to the existing RTUs. Refer to *Low-Flow Desiccant Air-Conditioning: General Guidance and Site Considerations* for suggested modifications to existing systems for optimal performance (NREL 2014).

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Conclusions

Low-flow LDAC is an emerging, rapidly evolving technology, and the case studies presented in this report form a snapshot in time of the state of the art as of roughly 2012. The three demonstrations provided the opportunity to evaluate the performance of the low-flow LDAC technology and to identify potential problems and means for improvement. Modeling of the interactions of the LDAC with a typical supermarket provided a conservative comparison of estimated LDAC energy cost and performance with a set of conventional DX systems.

In general, the LDAC's measured performance in the case studies was near expectations during the periods analyzed in this report. The LDAC technology proved capable of providing a large latent capacity and low DP air that is required to provide comfortable and desirable space conditions for supermarkets and the challenging environment of a natatorium. Indoor conditions at Whole Foods, Encinitas were consistently maintained within acceptable humidity levels (35%–55% RH) without conventional overcool-and-reheat strategies. The RSHI was higher than expected, because of a suspected water leak in the system. This finding has spurred industry to accelerate development of the wicking fin technology, which will eliminate this problem. The wicking fin design is less prone to water leaks due to its well-established and conventional metallic construction. For more information on wicking fin technology, refer to NREL (2013). RH levels at Whole Foods, Kailua were kept at 50%–75%, owing mainly to the effects of large unanticipated infiltration rates due to entry and loading dock doors being kept open during store operation. This appears to be a cultural response to a warm humid environment where air velocity across the human body was a traditional passive comfort strategy. However, in a grocery store, this is counterproductive from an energy and a comfort perspective. Such an operating schedule would be extremely rare on the U.S. mainland. Natatorium humidity was maintained within 10% of the ideal condition of 60% RH for 90% of the time. Humidity was controlled with a regeneration efficiency near expected values in most cases. The average RSHI for Whole Foods, Kailua and the Schaeffer Pool was 1.5 kBtu/lb water removed, which is near expected values. The average electricity consumption at all demonstrations was 0.32–0.45 kW/ton compared to a typical vapor compression system (not counting reheat) values of about 0.8–1.0 kW/ton.

There were periods when some of the LDAC systems were not fully functional because of mechanical issues. Observations during these periods were quite useful in showing how LDAC systems and installations can be further improved, and prepared for mass market readiness. Key lessons from the demonstrations were:

- The Encinitas grocery store and the Schaeffer natatorium both experienced precipitate formation in the desiccant resulting from air contaminants. At Encinitas, the scavenging air intake for the regenerator was located at a loading dock with heavy diesel exhaust. A carbon filter added to this airstream appears to have solved the problem. For the natatorium, the reason is less clear and needs to be understood before widespread use of LDAC in this application. The precipitate problem is also an indicator of the potential for liquid desiccant to operate as an air cleaning agent for both biological and chemical contaminants, thus potentially adding to the value proposition for LDAC.
- The Schaeffer Pool LDAC system design integrated a CHP system to utilize waste heat for desiccant regeneration. The CHP system did not always provide 160°F hot water, thus

reducing LDAC capacity. LDACs using hot water from a CHP system or using other sources of thermal energy such as solar or waste heat require thoughtful integrated system design to ensure that delivered temperatures always exceed 160°F.

- The Whole Foods grocery in Kailua was operated in a manner atypical for grocery stores on the U.S. mainland. The main entrance doors and the doors to the loading dock were kept open creating a strong cross-ventilation airflow, which greatly increased the infiltration load. The LDAC system was not sized to accommodate such a large unanticipated load and indoor RH drifted up to as high as 75%. It is important to understand any special operational conditions that will increase latent load when sizing an LDAC. This is especially critical in supermarkets, where sufficiently dry air enables more efficient operation of the refrigeration equipment.
- Operation of the LDAC systems at the Babbio Center was delayed beyond the time frame for this report because of installation problems, showing the need for proper training of designers and installers for this emerging technology. Modeling tools that simplify the design process would also be helpful.

Energy modeling provided estimates of the savings available with an LDAC in supermarket applications across the U.S. climate zones. The baseline models included four reheat options: (1) natural gas reheat coils; (2) RTU condenser hot-gas reheat with auxiliary natural gas reheat; (3) electric reheat coils: and (4) RTU condenser hot-gas reheat with auxiliary electric reheat. For a supermarket in a climate with high latent loads requiring 4,000 cfm of ventilation, we calculated energy cost savings of $3,000–$30,000 in the hot-humid climate zones of 1A and 2A, with corresponding source energy savings of 1%–6% of the supermarket's whole building energy expenditure. For climate zones 1A–2A, the estimated space conditioning source energy savings in grocery stores are 12%–40% for the four reheat strategies modeled. Space conditioning savings are realized because the large expenditure for overcooling and reheat in a DX system was eliminated. Supermarkets in mixed-humid climates (3A and 4A) are projected to show savings of around 1%–4% of building source energy, and utility cost savings of $200–$13,000. Cold-humid climates and marine climates are expected to show minimal differences in energy use, although some cost savings may be possible due to the shifting of energy consumption from electricity to gas where RTU condenser hot-gas reheat and/or electric reheat coils were used. Additional savings can be achieved with the use of a two-stage regenerator, which is estimated to save 40% of the thermal energy required for regeneration. With a two-stage regenerator, total building source energy savings are estimated to be 4%–8% in hot humid climate zones, with corresponding annual energy cost savings of $10,000–$36,000. HVAC savings in hot humid climates are 34%–57%. We chose to model the LDAC conservatively by not accounting for savings from improved control strategies and waste heat integration, and by not forcing the conventional baseline systems to maintain humidity conditions as low as the LDAC.

The industry is currently modifying LDAC to improve energy efficiency and reliability. These improvements include: (1) two-stage regeneration, which improves LDAC regenerator efficiency by about 40%; (2) wicking fin design, which solves the leak problems with the current LDAC element design; (3) membrane-based LDAC unit, which completely eliminates carryover; (4) improved LDAC control strategies, which increase energy savings; (5) better integration of alternative heat sources, which saves regenerator energy; and (6) integration of heat pumps with LDAC systems, which enable all-electric systems. These hardware improvements could benefit

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from laboratory research to better characterize the thermodynamics, improve the models, and optimize the system designs and building interactions. LDAC technology has promise as an effective means to save energy in applications where humidity control is essential and energy intensive; however, further development is needed for increased energy savings and improved reliability.

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Appendix A: Component- and System-Level LDAC Model Details

The following assumptions were used in the physical modeling of the LDAC conditioner:

1. Steady-state operation.
2. Laminar developing flow transfer coefficients for both heat and mass transfer from the bulk air to the air-desiccant interface, assuming a smooth surface and constant temperature within each cell at the interface and no fluid-fluid interaction.
3. Developing flow falling film transfer coefficients for mass transfer modeling in the desiccant, taken from Grossman (1982).
4. Estimations of heat transfer resistance in the desiccant film showed that this resistance was less than 1% of the overall heat transfer resistance and justified an assumption of negligible heat transfer resistance in the desiccant.
5. The flocking on the plate surface uniformly distributed the desiccant over the plate surface but negligibly affected heat and mass transfer within the desiccant layer. Neglecting the effect of the flocking on transport is justified by Lund and Knowles (2001), which shows a less than 5% effect on Nusselt number under the operating conditions of the LDAC.
6. The desiccant-plate interface was assumed to be impermeable to moisture transfer.
7. Conduction shape factors were used to model thermal conductance between the desiccant-plate interface and the water-plate interface. These were calculated with the correlation given in Ganzevles and Geld (1996).
8. Conduction and diffusion were assumed to occur in one dimension only (perpendicular to the plates).
9. Heat transfer coefficients describing heat transfer from the plate-water interface to the bulk water were taken from fully developed correlations for laminar pipe flow. This resistance was estimated at 2%–3% of the overall heat transfer resistance; thus, any error in this assumption should be negligible.
10. All desiccant properties were taken from Conde (2009) except for enthalpy, which was calculated with a correlation provided by AIL Research.

Half of a single LDAC plate, one desiccant film, and half of the adjacent air gap was represented in the component-level model. The plate was divided into eight elements in each direction and the mass and energy conservation equations were solved in each element. Increasing grid resolution beyond this point was shown to negligibly affect the results (less than 1% change in relevant quantities). A Newton solver was used to adjust state variables until normalized residuals were below 10^{-7}, at which time energy balances were accurate within 0.015% and mass balances within machine precision. With the preceding assumptions and methods employed, the modeled moisture removal rate compared well with the 39 lab conditions tested as shown in Figure A–1. Outlet temperatures of the three fluids were predicted with an average residual of less than 0.9°F.

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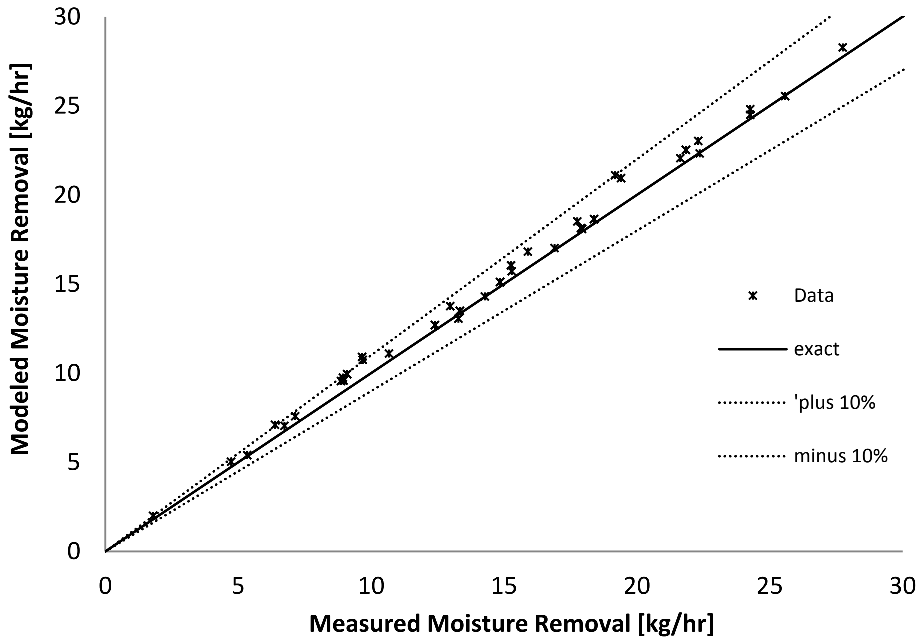

Figure A–1 Comparison of conditioner model and laboratory data showing good agreement

A polynomial mapping of the outlet variables to nine inlet variables was generated for a sample of 500 data points spanning the entire expected and modeled operating range for all inlet variables. The fits of these maps for heat removed from the heating water, heat added to the scavenging air, and moisture removed from the desiccant were 0.999, 0.998, and 0.999, respectively. This mapping was used as an input for the system-level model.

Processes inside the regenerator are more complex and could not be modeled to the desired level of accuracy with a purely physical approach. However, laboratory data are available for the regenerator over the entire expected operating range. Therefore, an empirical correlation of the laboratory data was used as an input to the system level model. Three quantities (heat removed from the heating water, heat added to the scavenging air, and moisture removed from the desiccant) were mapped as a function of the nine governing input variables. Two other output variables needed to fully define operation were fixed by energy and mass balances. The three quantities were predicted by the empirical correlations with coefficients of determination (R^2) of 0.981, 0.981, and 0.976, respectively.

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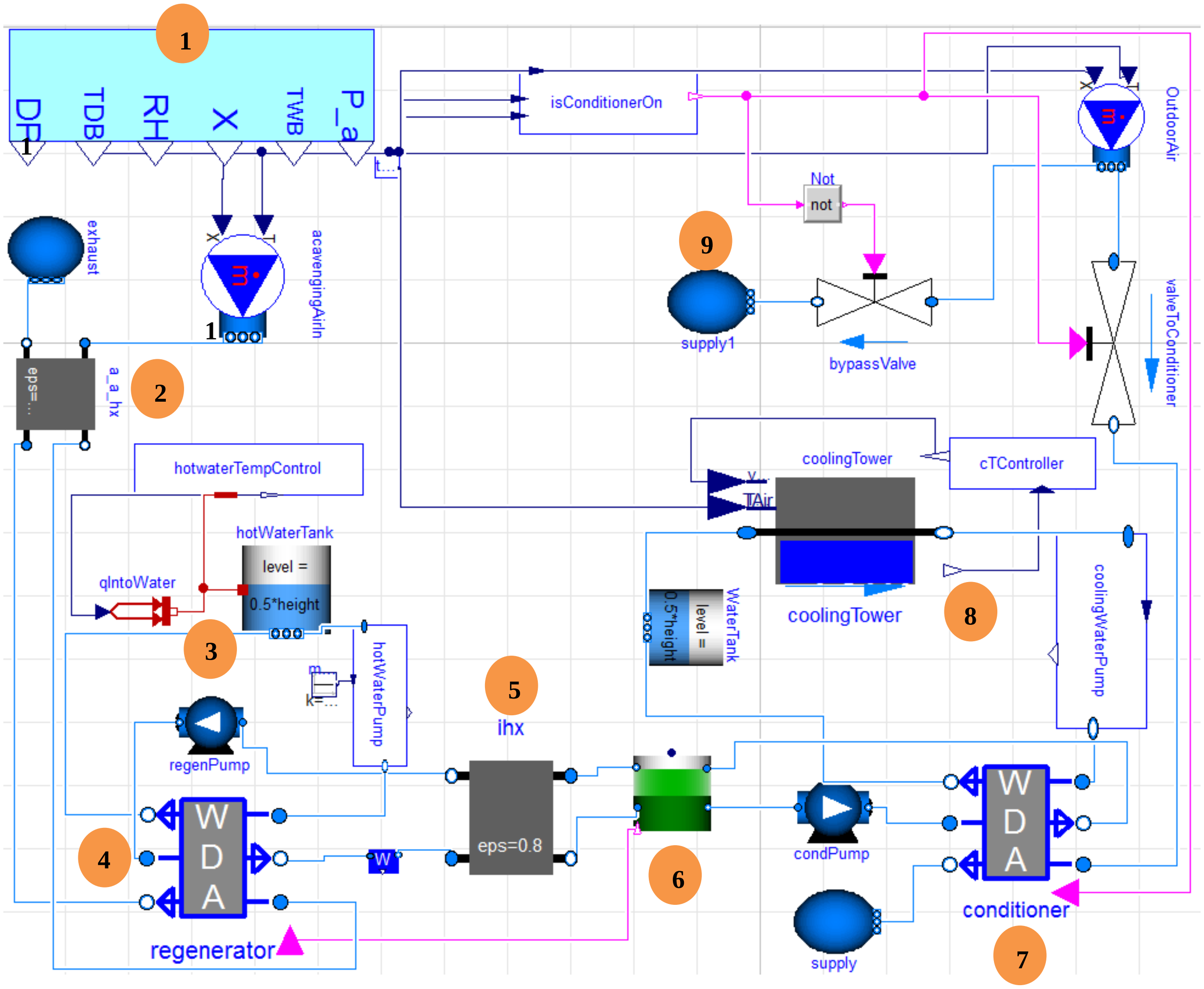

Figure A–2 Schematic of system-level LDAC model

A schematic of the system-level model is shown in Figure A-2. Inputs and assumptions for each component of the model are described in the following. The schematic is explained by beginning at the upper left corner and proceeding in a counter-clockwise direction.

1. Starting from the upper left corner of the schematic, TMY3 weather data are represented by the light blue box (including, from right to left, atmospheric pressure, wet bulb temperature, HR, RH, DB, and DP). The software linearly interpolates between the hourly TMY3 data points to give fully dynamic boundary conditions.
2. Directly below, a constant flow rate scavenging air input is modeled. The air is preheated with an air-to-air heat exchanger with a constant effectiveness of 0.55. This effectiveness was chosen to prevent condensation in the heat exchanger at the worst operating conditions.
3. Directly below the air-air heat exchanger is the hot water loop, which supplies heating water to the regenerator. This includes a 110-kW constant-rate heat input, a hot water storage tank with a capacity of 2.7 ft^3, and a controller, which maintains the temperature

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of the hot water at 144°–176°F. The boiler efficiency is assumed to be 0.8. The pump is modeled as a constant-flow rate device. The small tank is assumed to be insulated well enough to prevent appreciable heat transfer to the environment.

4. The regenerator is shown below the hot water loop, which treats the three fluid streams (water, desiccant, and air, labeled as W, D, and A) according to the procedure discussed above.
5. To the right of the regenerator is an interchange heat exchanger, which exchanges sensible heat between weak and strong desiccant streams with an assumed constant effectiveness of 0.8.
6. To the right of this is a model of a completely stratified desiccant tank. In this model, the strong and weak desiccant regions of the tank are modeled as two individual tanks, except that weak desiccant can be pulled into the strong tank if the conditioner is running at a higher flow rate than the regenerator. This captures the stratification that occurs in the field caused by density differences between weak and strong desiccant. This is the only element whose operation is fully transient. Desiccant concentration and temperature in the tank are calculated continuously by applying energy and mass balances on the tank volume.
7. In the bottom right corner is the conditioner model.
8. Above the conditioner is the cooling water loop. This includes a model of a York cooling tower with a variable-speed fan previously implemented in Dymola by the Simulation Research Group at Lawrence Berkeley Laboratory. The cooling tower is sized to provide a 7°F approach at design conditions and a 10°F range. Cooling tower performance is given by a performance map of the York cooling tower. This cooling tower model is also implemented in the building simulation program EnergyPlus. A controller adjusts fan speed to one of three speeds according to delivered water temperature; natural convection operation of the cooling tower is also modeled when the fan is off. At design conditions, desired water temperature is set to be 7°F above the site's design DP for all sites. The pump is modeled as a constant-flow rate device.
9. The upper right corner of the schematic represents the constant supply airflow of the LDAC. OA is delivered directly to the conditioner when the conditioner is in operation. When the LDAC conditioner is shut off, OA is sent through a bypass valve to the existing vapor compression system.
10. (not shown) A new class was implemented for the LiCl solution used as the liquid desiccant in this system, which extends the Partial Medium model included in the Modelica Standard Library. This model implements all properties contained in Conde (2009) with two exceptions: specific heat capacity is modeled as constant value rather than a function of temperature, which results in less than 5% discrepancy at the extremes of the operating range, and density is modeled as a function of concentration only (not temperature) resulting in negligible discrepancy with the Conde (2009) relations. Specific enthalpy is also modeled with correlations developed by AIL Research.

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Appendix B: Modeling Results for Alternative Reheat Strategies

Case 1a: Natural Gas Reheat Coils (Single-Stage Regenerator)

Table B–1 Annual Whole Building Source Energy Intensity – Natural Gas Reheat Coils – Single-Stage Regenerator

End Use Category (kBtu/ft^2)	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	BL	LD	BL	LD	BL	LD	BL	LDAC	BL	LD	BL	LD	BL	LD
Cooling	55.1	9.0	41.6	7.1	15.5	2.4	2.5	0.9	11.8	0.8	8.4	0.9	6.3	0.7
Space heating	20.2	19.6	39.1	38.4	54.1	53.7	42.9	42.2	71.4	72.0	84.5	84.5	94.7	94.8
Reheat	28.8	0.0	21.6	0.0	9.4	0.0	1.6	0.0	7.4	0.0	5.9	0.0	4.8	0.0
Refrigeration	462.2	462.2	461.3	457.2	387.8	380.0	379.6	365.4	384.7	371.4	365.9	359.4	345.2	340.1
• Electric defrost	10.9	9.6	10.3	8.9	7.7	6.4	8.4	6.5	6.9	5.9	6.3	5.3	5.2	4.4
• Anti-sweat	36.0	33.7	36.7	34.0	30.8	28.3	32.6	28.9	30.2	28.2	28.8	26.6	26.3	24.8
• Compressor Rack	230.7	233.6	234.3	234.5	198.7	196.2	188.8	183.7	199.4	193.4	190.7	188.6	181.1	179.4
• Condenser fan	81.9	82.5	67.5	67.3	46.4	42.0	45.6	42.0	37.3	33.0	30.2	29.1	26.2	25.1
Fans	17.3	14.5	18.7	15.9	13.7	11.7	10.0	9.6	13.1	10.3	12.1	10.7	12.4	11.3
Base load – electricity	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4
Base load – nat gas	5.3	5.3	5.4	5.4	5.4	5.4	5.4	5.4	5.5	5.5	5.5	5.5	5.5	5.5
LDAC – electricity		5.6		5.1		3.7		4.4		3.2		2.8		2.4
LDAC – gas		48.2		37.1		25.4		26.2		19.4		16.1		16.3
Source savings	24.5		21.4		3.6		–11.9		11.3		2.4		–2.0	

*Baseline Case

**Liquid Desiccant Air Conditioner Case

Table B–2 LDAC Incremental Cost Target – Natural Gas Reheat Coils – Single–Stage Regenerator

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	32,364	24,051	6,223	(11,336)	13,514	3,415	(1,345)
5-year	53,940	40,084	10,372	(18,894)	22,524	5,692	(2,241)
10-year	107,881	80,168	20,745	(37,788)	45,047	11,383	(4,483)

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Case 1b: Natural Gas Reheat Coils (Two-Stage Regenerator)

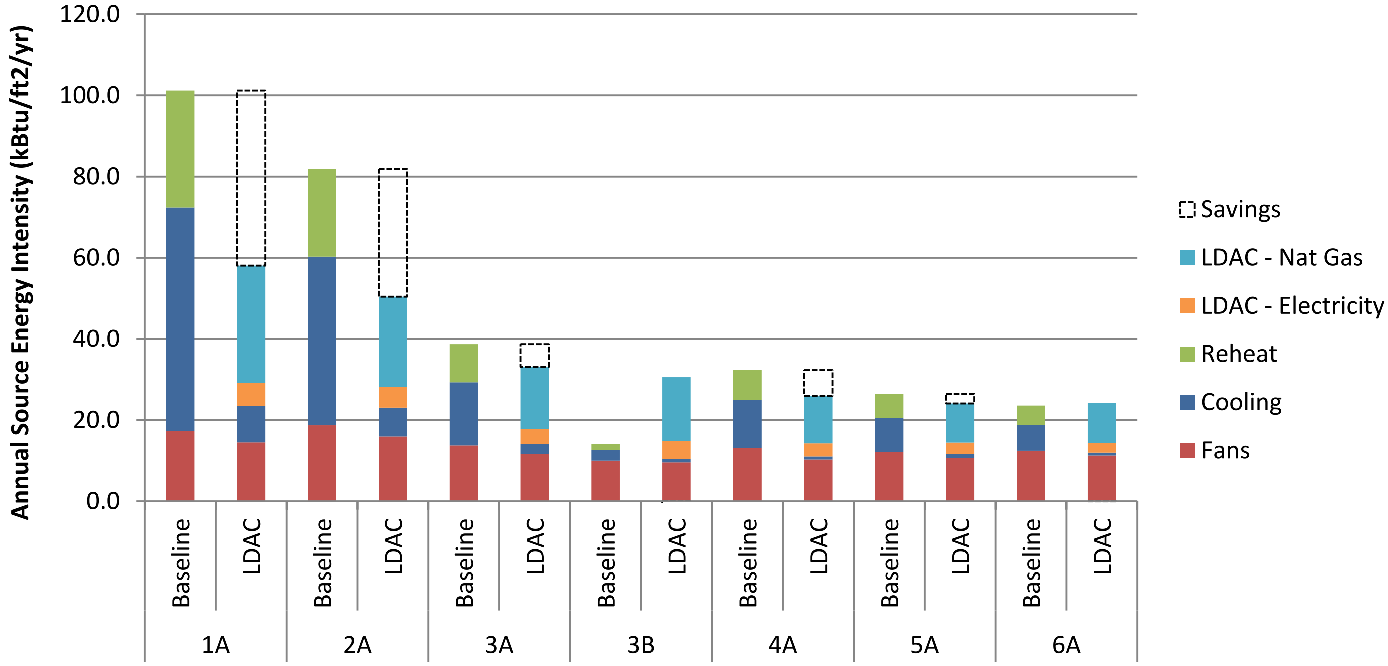

Figure B–1 Annual ventilation and air conditioning source energy intensity and savings – natural gas reheat coils – two-stage regenerator
(Credit: Lesley Herrmann/NREL)

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Table B–3 Annual Whole Building Source Energy Intensity – Natural Gas Reheat Coils – Two-Stage Regenerator

End Use Category (kBtu/ft^2)	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	BL	LD	BL	LD	BL	LD	BL	LDAC	BL	LD	BL	LD	BL	LD
Cooling	55.1	9.0	41.6	7.1	15.5	2.4	2.5	0.9	11.8	0.8	8.4	0.9	6.3	0.7
Space heating	20.2	19.6	39.1	38.4	54.1	53.7	42.9	42.2	71.4	72.0	84.5	84.5	94.7	94.8
Reheat	28.8	0.0	21.6	0.0	9.4	0.0	1.6	0.0	7.4	0.0	5.9	0.0	4.8	0.0
Refrigeration	462.2	462.2	461.3	457.2	387.8	380.0	379.6	365.4	384.7	371.4	365.9	359.4	345.2	340.1
• Electric defrost	10.9	9.6	10.3	8.9	7.7	6.4	8.4	6.5	6.9	5.9	6.3	5.3	5.2	4.4
• Anti-sweat	36.0	33.7	36.7	34.0	30.8	28.3	32.6	28.9	30.2	28.2	28.8	26.6	26.3	24.8
• Compressor rack	230.7	233.6	234.3	234.5	198.7	196.2	188.8	183.7	199.4	193.4	190.7	188.6	181.1	179.4
• Condenser fan	81.9	82.5	67.5	67.3	46.4	42.0	45.6	42.0	37.3	33.0	30.2	29.1	26.2	25.1
Fans	17.3	14.5	18.7	15.9	13.7	11.7	10.0	9.6	13.1	10.3	12.1	10.7	12.4	11.3
Base load – electricity	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4
Base load – nat gas	5.3	5.3	5.4	5.4	5.4	5.4	5.4	5.4	5.5	5.5	5.5	5.5	5.5	5.5
LDAC – electricity		5.6		5.1		3.7		4.4		3.2		2.8		2.4
LDAC – gas		28.9		22.3		15.3		15.7		11.7		9.6		9.8
Source savings	43.8		36.3		13.7		–1.4		19.0		8.8		4.5	

Table B–4 LDAC Incremental Cost Target – Natural Gas Reheat Coils – Two-Stage Regenerator

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	52,988	39,986	17,129	(120)	21,823	10,299	5,629
5-year	88,313	66,643	28,548	(199)	36,371	17,166	9,381
10-year	176,626	133,285	57,095	(398)	72,743	34,331	18,762

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Case 2a: RTU Condenser Hot-Gas Reheat With Auxiliary Natural Gas Reheat Coils (Single-Stage Regenerator)

Table B–5 Annual Whole Building Source Energy Intensity – RTU Condenser Hot-Gas Reheat With Auxiliary Natural Gas Reheat Coils – Single-Stage Regenerator

End Use Category (kBtu/ft^2)	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	BL	LD	BL	LD	BL	LD	BL	LDAC	BL	LD	BL	LD	BL	LD
Cooling	57.0	9.0	43.9	7.1	17.1	2.4	2.7	0.9	12.5	0.8	8.8	0.9	6.7	0.7
Space heating	20.2	19.6	39.2	38.4	54.1	53.7	42.9	42.2	71.4	72.0	84.5	84.5	94.7	94.8
Reheat	18.8	0.0	15.9	0.0	7.5	0.0	2.6	0.0	7.7	0.0	8.7	0.0	9.0	0.0
Refrigeration	452.8	462.2	454.5	457.2	384.6	380.0	379.6	365.4	383.1	371.4	365.5	359.4	345.5	340.1
• **Electric defrost**	10.7	9.6	10.2	8.9	7.7	6.4	8.4	6.5	6.9	5.9	6.3	5.3	5.2	4.4
• **Anti-sweat**	35.9	33.7	36.6	34.0	30.7	28.3	32.6	28.9	30.2	28.2	28.8	26.6	26.3	24.8
• **Compressor rack**	229.8	233.6	233.7	234.5	198.4	196.2	189.1	183.7	199.8	193.4	191.5	188.6	182.1	179.4
• **Condenser fan**	73.6	82.5	61.5	67.3	43.7	42.0	45.3	42.0	35.4	33.0	29.0	29.1	25.4	25.1
Fans	17.3	14.5	18.7	15.9	13.7	11.7	10.0	9.6	13.1	10.3	12.1	10.7	12.4	11.3
Base load – electricity	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4
Base load – nat gas	5.3	5.3	5.4	5.4	5.4	5.4	5.4	5.4	5.5	5.5	5.5	5.5	5.5	5.5
LDAC – electricity		5.6		5.1		3.7		4.4		3.2		2.8		2.4
LDAC – gas		48.2		37.1		25.4		26.2		19.4		16.1		16.3
Source savings	7.0		11.3		0.1		–10.7		10.7		5.2		2.8	

Table B–6 LDAC Incremental Cost Target – RTU Condenser Hot-Gas Reheat With Natural Gas Reheat Coils – Single-Stage Regenerator

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	8,522	9,652	657	(10,515)	11,215	4,569	1,995
5-year	14,204	16,086	1,095	(17,524)	18,691	7,615	3,325
10-year	28,408	32,172	2,191	(35,049)	37,382	15,229	6,649

This report is available at no cost from the National Renewable Energy Laboratory (NREL) at www.nrel.gov/publications.

Case 2b: RTU Condenser Hot-Gas Reheat With Auxiliary Natural Gas Reheat Coils (Two-Stage Regenerator)

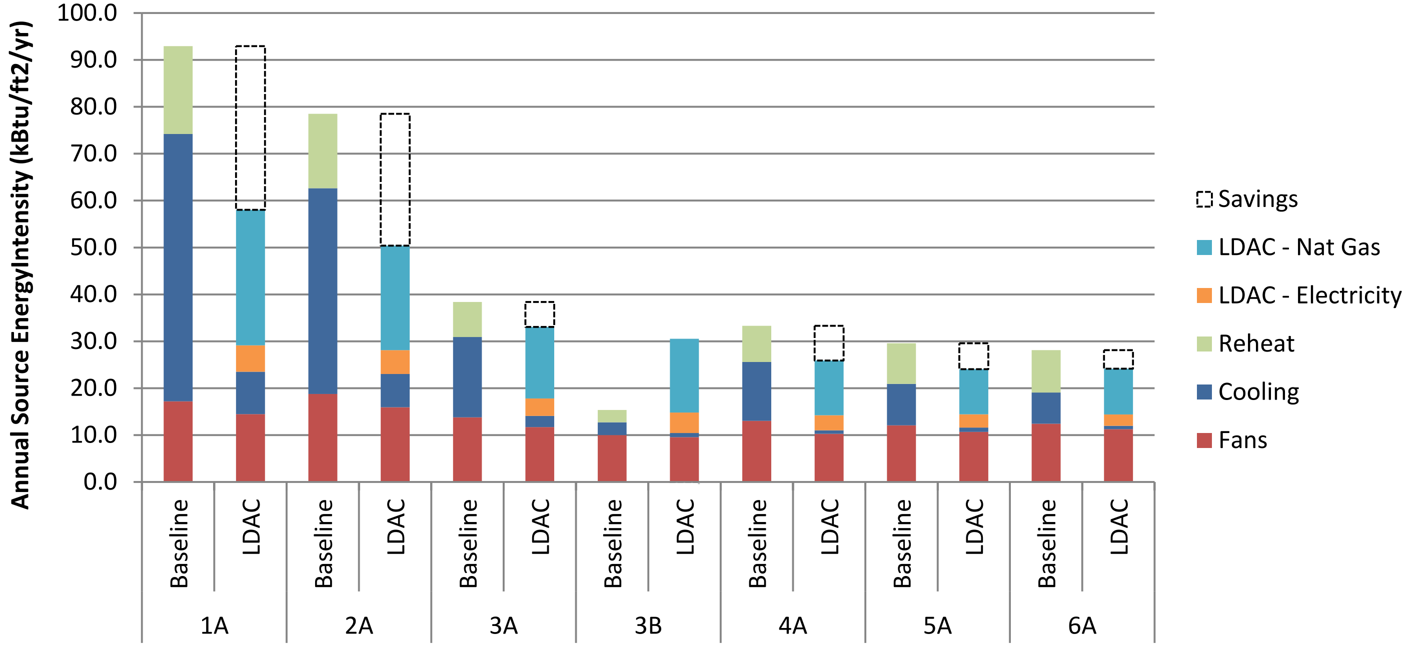

Figure B–2 Annual ventilation and air conditioning source energy intensity and savings – RTU condenser hot-gas reheat with auxiliary natural gas reheat coils – two-stage regenerator
(Credit: Lesley Herrmann/NREL)

This report is available at no cost from the National Renewable Energy Laboratory (NREL) at www.nrel.gov/publications.

Table B–7 Annual Whole Building Source Energy Intensity – RTU Condenser Hot-Gas Reheat With Auxiliary Natural Gas Reheat Coils – Two-Stage Regenerator

End Use Category (kBtu/ft^2)	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	BL	LD	BL	LD	BL	LD	BL	LDAC	BL	LD	BL	LD	BL	LD
Cooling	57.0	9.0	43.9	7.1	17.1	2.4	2.7	0.9	12.5	0.8	8.8	0.9	6.7	0.7
Space heating	20.2	19.6	39.2	38.4	54.1	53.7	42.9	42.2	71.4	72.0	84.5	84.5	94.7	94.8
Reheat	18.8	0.0	15.9	0.0	7.5	0.0	2.6	0.0	7.7	0.0	8.7	0.0	9.0	0.0
Refrigeration	452.8	462.2	454.5	457.2	384.6	380.0	379.6	365.4	383.1	371.4	365.5	359.4	345.5	340.1
• Electric defrost	10.7	9.6	10.2	8.9	7.7	6.4	8.4	6.5	6.9	5.9	6.3	5.3	5.2	4.4
• Anti-sweat	35.9	33.7	36.6	34.0	30.7	28.3	32.6	28.9	30.2	28.2	28.8	26.6	26.3	24.8
• Compressor rack	229.8	233.6	233.7	234.5	198.4	196.2	189.1	183.7	199.8	193.4	191.5	188.6	182.1	179.4
• Condenser fan	73.6	82.5	61.5	67.3	43.7	42.0	45.3	42.0	35.4	33.0	29.0	29.1	25.4	25.1
Fans	17.2	14.5	18.8	15.9	13.8	11.7	10.0	9.6	13.0	10.3	12.1	10.7	12.4	11.3
Base load – electricity	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4
Base load – nat gas	5.3	5.3	5.4	5.4	5.4	5.4	5.4	5.4	5.5	5.5	5.5	5.5	5.5	5.5
LDAC – electricity		5.6		5.1		3.7		4.4		3.2		2.8		2.4
LDAC – gas		28.9		22.3		15.3		15.7		11.7		9.6		9.8
Source savings	26.2		26.2		10.3		–0.2		18.5		11.6		9.3	

Table B–8 LDAC Incremental Cost Target – Refrigeration Condenser Hot-Gas Reheat With Natural Gas Reheat Coils – Two-Stage Regenerator

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	29,146	25,587	11,562	702	19,523	11,453	8,968
5-year	48,577	42,644	19,271	1,170	32,539	19,088	14,947
10-year	97,153	85,289	38,541	2,341	65,078	38,177	29,894

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Case 3a: Electric Reheat Coils (Single-Stage Regenerator)

Table B–9 Annual Whole Building Source Energy Intensity – Electric Reheat Coils – Single-Stage Regenerator

End Use Category (kBtu/ft^2)	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	BL	LD	BL	LD	BL	LD	BL	LDAC	BL	LD	BL	LD	BL	LD
Cooling	55.1	9.0	41.6	7.1	15.5	2.4	2.5	0.9	11.8	0.8	8.4	0.9	6.3	0.7
Space heating	20.2	19.6	39.1	38.4	54.1	53.7	42.9	42.2	71.4	72.0	84.5	84.5	94.7	94.8
Reheat	70.7	0.0	71.8	0.0	29.0	0.0	4.8	0.0	24.2	0.0	19.1	0.0	15.0	0.0
Refrigeration	462.2	462.2	461.3	457.2	387.8	380.0	379.6	365.4	384.7	371.4	365.9	359.4	345.2	340.1
• Electric defrost	10.9	9.6	10.3	8.9	7.7	6.4	8.4	6.5	6.9	5.9	6.3	5.3	5.2	4.4
• Anti-sweat	36.0	33.7	36.7	34.0	30.8	28.3	32.6	28.9	30.2	28.2	28.8	26.6	26.3	24.8
• Compressor rack	230.7	233.6	234.3	234.5	198.7	196.2	188.8	183.7	199.4	193.4	190.7	188.6	181.1	179.4
• Condenser fan	81.9	82.5	67.5	67.3	46.4	42.0	45.6	42.0	37.3	33.0	30.2	29.1	26.2	25.1
Fans	17.3	14.5	18.7	15.9	13.7	11.7	10.0	9.6	13.1	10.3	12.1	10.7	12.4	11.3
Base load – electricity	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4
Base load – nat gas	5.3	5.3	5.4	5.4	5.4	5.4	5.4	5.4	5.5	5.5	5.5	5.5	5.5	5.5
LDAC – electricity		5.6		5.1		3.7		4.4		3.2		2.8		2.4
LDAC – gas		48.2		37.1		25.4		26.2		19.4		16.1		16.3
Cooling	66.4		71.6		23.1		–8.7		28.0		15.6		8.2	

Table B–10 LDAC Incremental Cost Target – Electric Reheat Coils – Single-Stage Regenerator

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	87,487	80,675	30,892	(7,236)	32,872	18,841	11,146
5-year	145,812	134,459	51,486	(12,061)	54,786	31,401	18,577
10-year	291,625	268,918	102,972	(24,121)	109,573	62,803	37,154

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Case 3b: Electric Reheat Coils (Two-Stage Regenerator)

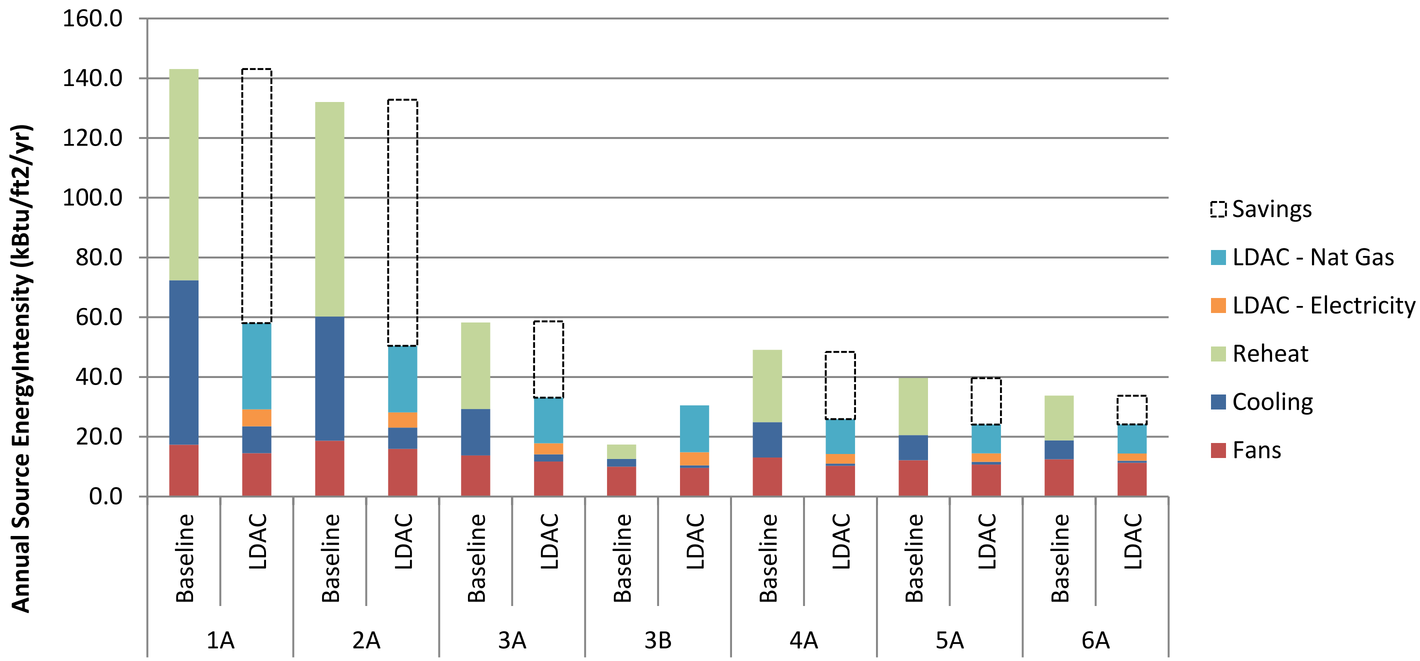

Figure B–3 Annual Ventilation and air conditioning source energy intensity and savings – electric reheat coils – two-stage regenerator

(Credit: Lesley Herrmann/NREL)

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Table B–11 Annual Whole Building Source Energy Intensity – Electric Reheat Coils – Two-Stage Regenerator

End Use Category (kBtu/ft^2)	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	BL	LD	BL	LD	BL	LD	BL	LDAC	BL	LD	BL	LD	BL	LD
Cooling	55.1	9.0	41.6	7.1	15.5	2.4	2.5	0.9	11.8	0.8	8.4	0.9	6.3	0.7
Space heating	20.2	19.6	39.1	38.4	54.1	53.7	42.9	42.2	71.4	72.0	84.5	84.5	94.7	94.8
Reheat	70.7	0.0	71.8	0.0	29.0	0.0	4.8	0.0	24.2	0.0	19.1	0.0	15.0	0.0
Refrigeration	462.2	462.2	461.3	457.2	387.8	380.0	379.6	365.4	384.7	371.4	365.9	359.4	345.2	340.1
• **Electric defrost**	10.9	9.6	10.3	8.9	7.7	6.4	8.4	6.5	6.9	5.9	6.3	5.3	5.2	4.4
• **Anti-sweat**	36.0	33.7	36.7	34.0	30.8	28.3	32.6	28.9	30.2	28.2	28.8	26.6	26.3	24.8
• **Compressor rack**	230.7	233.6	234.3	234.5	198.7	196.2	188.8	183.7	199.4	193.4	190.7	188.6	181.1	179.4
• **Condenser fan**	81.9	82.5	67.5	67.3	46.4	42.0	45.6	42.0	37.3	33.0	30.2	29.1	26.2	25.1
Fans	17.3	14.5	18.7	15.9	13.7	11.7	10.0	9.6	13.1	10.3	12.1	10.7	12.4	11.3
Base load – electricity	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4
Base load – nat gas	5.3	5.3	5.4	5.4	5.4	5.4	5.4	5.4	5.5	5.5	5.5	5.5	5.5	5.5
LDAC – electricity		5.6		5.1		3.7		4.4		3.2		2.8		2.4
LDAC – gas		28.9		22.3		15.3		15.7		11.7		9.6		9.8
Cooling	85.7		86.5		33.3		1.8		35.8		22.0		14.7	

Table B–12 LDAC Incremental Cost Target – Electric Reheat Coils – Two-Stage Regenerator

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	108,111	96,610	41,797	3,980	41,181	25,725	18,120
5-year	180,185	161,017	69,662	6,634	68,634	42,875	30,199
10-year	360,370	322,034	139,323	13,268	137,269	85,751	60,398

Case 4a: RTU Condenser Hot-Gas Reheat With Auxiliary Electric Reheat Coils (Single-Stage Regenerator)

Table B–13 Annual Whole Building Source Energy Intensity – RTU Condenser Hot-Gas Reheat With Auxiliary Electric Reheat Coils – Single-Stage Regenerator

End Use Category (kBtu/ft²)	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	BL	LD	BL	LD	BL	LD	BL	LDAC	BL	LD	BL	LD	BL	LD
Cooling	56.3	9.0	43.0	7.1	16.5	2.4	2.7	0.9	12.3	0.8	8.7	0.9	6.6	0.7
Space heating	20.2	19.6	39.2	38.4	54.1	53.7	42.9	42.2	71.4	72.0	84.5	84.5	94.7	94.8
Reheat	22.3	0.0	22.3	0.0	11.9	0.0	5.9	0.0	14.7	0.0	18.6	0.0	19.2	0.0
Refrigeration	461.7	462.2	460.9	457.2	387.7	380.0	380.2	365.4	385.4	371.4	367.3	359.4	346.8	340.1
• **Electric defrost**	10.8	9.6	10.2	8.9	7.7	6.4	8.4	6.5	6.9	5.9	6.3	5.3	5.2	4.4
• **Anti-sweat**	35.9	33.7	36.6	34.0	30.7	28.3	32.6	28.9	30.2	28.2	28.8	26.6	26.4	24.8
• **Compressor rack**	230.5	233.6	234.1	234.5	198.7	196.2	189.2	183.7	199.9	193.4	191.6	188.6	182.2	179.4
• **Condenser fan**	81.8	82.5	67.4	67.3	46.4	42.0	45.8	42.0	37.5	33.0	30.6	29.1	26.5	25.1
Fans	17.3	14.5	18.7	15.9	13.7	11.7	10.0	9.6	13.1	10.3	12.1	10.7	12.4	11.3
Base load – electricity	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4
Base load – nat gas	5.3	5.3	5.4	5.4	5.4	5.4	5.4	5.4	5.5	5.5	5.5	5.5	5.5	5.5
LDAC – electricity		5.6		5.1		3.7		4.4		3.2		2.8		2.4
LDAC – gas		48.2		37.1		25.4		26.2		19.4		16.1		16.3
Cooling	18.8		23.3		7.0		–6.9		19.6		16.7		14.1	

Table B–14 LDAC Incremental Cost Target – RTU Condenser Hot-Gas Reheat With Auxiliary Electric Reheat Coils – Single-Stage Regenerator

	1A: Miami	2A: Houston	3A: Atlanta	3B: Long Beach	4A: Baltimore	5A: Chicago	6A: Minneapolis
3-year	29,556	26,963	11,517	(5,066)	23,394	20,039	18,138
5-year	49,260	44,939	19,194	(8,443)	38,989	33,399	30,231
10-year	98,521	89,878	38,389	(16,886)	77,979	66,798	60,461

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Case 4b: RTU Condenser Hot-Gas Reheat With Auxiliary Electric Reheat Coils (Two-Stage Regenerator)

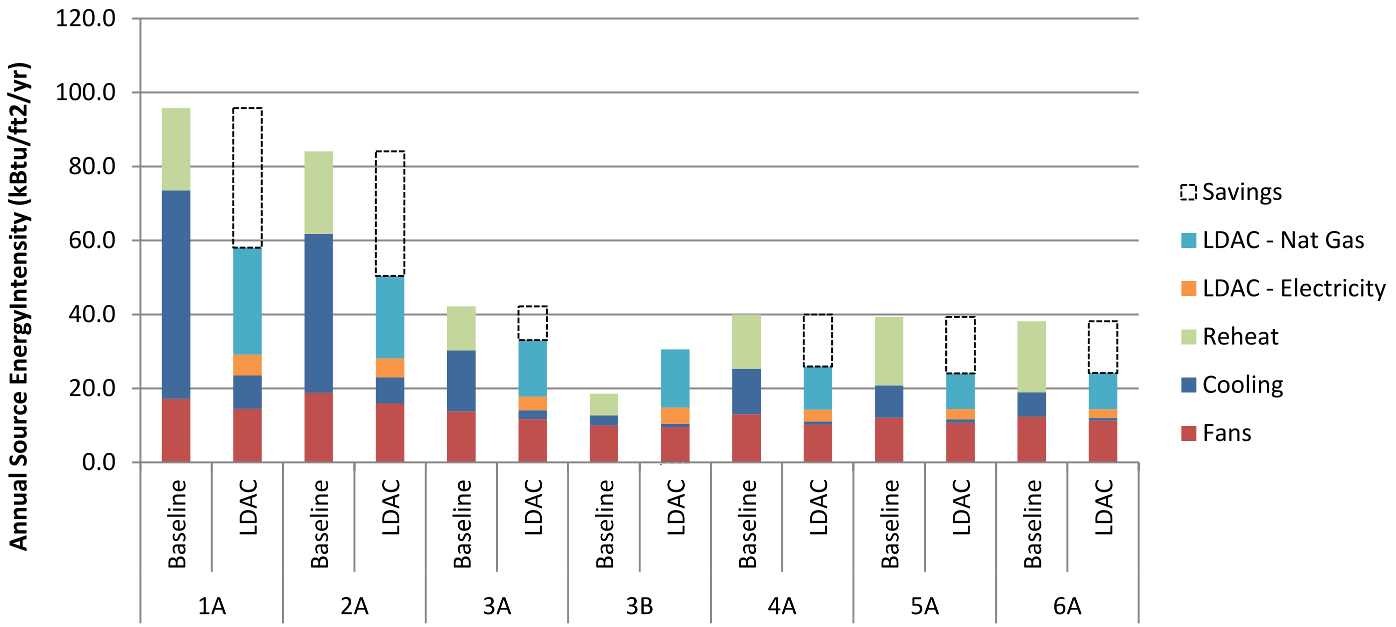

Figure B–4 Annual conditioning source energy intensity and savings – RTU condenser hot-gas reheat with auxiliary electric reheat coils – two-stage regenerator
(Credit: Lesley Herrmann/NREL)

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Table B–15 Annual Whole Building Source Energy Intensity – RTU Condenser Hot-Gas Reheat With Auxiliary Electric Reheat Coils – Two-Stage Regenerator

End Use Category (kBtu/ft^2)	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	BL	LD	BL	LD	BL	LD	BL	LDAC	BL	LD	BL	LD	BL	LD
Cooling	56.3	9.0	43.0	7.1	16.5	2.4	2.7	0.9	12.3	0.8	8.7	0.9	6.6	0.7
Space heating	20.2	19.6	39.2	38.4	54.1	53.7	42.9	42.2	71.4	72.0	84.5	84.5	94.7	94.8
Reheat	22.3	0.0	22.3	0.0	11.9	0.0	5.9	0.0	14.7	0.0	18.6	0.0	19.2	0.0
Refrigeration	461.7	462.2	460.9	457.2	387.7	380.0	380.2	365.4	385.4	371.4	367.3	359.4	346.8	340.1
• Electric defrost	10.8	9.6	10.2	8.9	7.7	6.4	8.4	6.5	6.9	5.9	6.3	5.3	5.2	4.4
• Anti-sweat	35.9	33.7	36.6	34.0	30.7	28.3	32.6	28.9	30.2	28.2	28.8	26.6	26.4	24.8
• Compressor rack	230.5	233.6	234.1	234.5	198.7	196.2	189.2	183.7	199.9	193.4	191.6	188.6	182.2	179.4
• Condenser fan	81.8	82.5	67.4	67.3	46.4	42.0	45.8	42.0	37.5	33.0	30.6	29.1	26.5	25.1
Fans	17.2	14.5	18.8	15.9	13.8	11.7	10.0	9.6	13.0	10.3	12.1	10.7	12.4	11.3
Base load – electricity	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4	145.4
Base load – nat gas	5.3	5.3	5.4	5.4	5.4	5.4	5.4	5.4	5.5	5.5	5.5	5.5	5.5	5.5
LDAC – electricity		5.6		5.1		3.7		4.4		3.2		2.8		2.4
LDAC – gas		28.9		22.3		15.3		15.7		11.7		9.6		9.8
Cooling	37.9		38.2		17.2		3.6		27.4		23.0		20.6	

Table B–16 Total Building Annual Energy Percent Savings (%) (Single-Stage Regenerator)

Reheat Strategy	1A Miami		2A Houston		3A Atlanta		3B Long Beach		4A Baltimore		5A Chicago		6A Minneapolis	
	Site	Source	Site	Source	Site	Source	Site	Source	Site	Source	Site	Source	Site	Source
Case 1: nat gas	–2%	3%	–1%	3%	–4%	1%	–9%	–2%	–2%	2%	–2%	0%	–3%	0%
Case 2: RTU cond. + nat gas	–6%	1%	–4%	2%	–5%	0%	–8%	–2%	–2%	2%	–1%	1%	–2%	0%
Case 3: electricity	–4%	9%	–1%	9%	–4%	4%	–9%	–1%	–2%	4%	–2%	2%	–3%	1%
Case 4: RTU cond. + electricity	–3%	5%	–1%	5%	–2%	3%	–4%	1%	0%	4%	0%	4%	0%	3%

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Table B–17 Total Building Annual Energy Percent Savings (%) (Two-Stage Regenerator)

Reheat Strategy	1A: Miami		2A: Houston		3A: Atlanta		3B: Long Beach		4A: Baltimore		5A: Chicago		6A: Minneapolis	
	Site	Source	Site	Source	Site	Source	Site	Source	Site	Source	Site	Source	Site	Source
Case 1: nat gas	5%	6%	4%	5%	0%	2%	–4%	0%	1%	3%	0%	1%	–1%	1%
Case 2: RTU Cond. + nat gas	1%	4%	1%	4%	–1%	2%	–4%	0%	1%	3%	1%	2%	1%	1%
Case 3: electricity	3%	11%	4%	11%	0%	5%	–4%	0%	1%	5%	0%	3%	–1%	2%
Case 4: RTU Cond. + electricity	–1%	7%	0%	7%	–1%	3%	–4%	1%	0%	5%	0%	4%	0%	4%

Table B–18 Annual Energy Cost Savings ($1,000/year) – Single-Stage Regenerator

Reheat Strategy	1A Miami		2A Houston		3A Atlanta		3B Long Beach		4A Baltimore		5A Chicago		6A Minneapolis	
	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas
Nat gas	17	–7	13	–5	8	–6	5	–9	9	–5	5	–4	4	–4
Hot-gas + nat gas	14	–12	12	–8	7	–7	5	–8	9	–5	5	–3	4	–3
Electricity	46	–17	40	–13	19	–9	7	–9	18	–7	12	–6	10	–6
Hot-gas + electricity	27	–17	22	–13	13	–9	7	–9	15	–7.2	12	7	12	6

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Table B–19 Annual Energy Cost Savings ($1,000/year) – Two-Stage Regenerator

Reheat Strategy	1A Miami		2A Houston		3A Atlanta		3B Long Beach		4A Baltimore		5A Chicago		6A Minneapolis	
	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas	Elec	Nat Gas
Nat gas	17	0	13	0	8	–2	5	–5	9	–2	5	–1	4	–2
Hot-gas + nat gas	14	–5	12	–3	7	–3	5	–5	9	–2	5	–1	4	–1
Electricity	46	–10	40	–8	19	–5	7	–5	18	–4	12	–3	10	–4
Hot-gas + electricity	27	–10.1	22	–14	13	–5	7.4	–5	15	–4.4	12	–4	12	–4