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DESIGN STUDY OF A PACKAGED LIQUID DESICCANT (LD) SYSTEM

Task 7

Prepared For
Hawaii Natural Energy Institute

Prepared By
Sustainable Design & Consulting LLC & HNEI

November 2017

DESICCANT DEHUMIDIFICATION TO SUPPORT ENERGY EFFICIENT SPACE CONDITIONING SYSTEMS FOR HAWAII

Project Phase 1: Design Study and Project Site Selection

Project Deliverable 4:

"DESIGN STUDY OF A PACKAGED LIQUID DESICCANT (LD) SYSTEM IN A TEST FACILITY TO CARRY OUT A TEST PROGRAM UNDER LAB CONDITIONS AND SUBSEQUENT ON-SITE TEST OPERATION OF THE LD SYSTEM"

November 22, 2017

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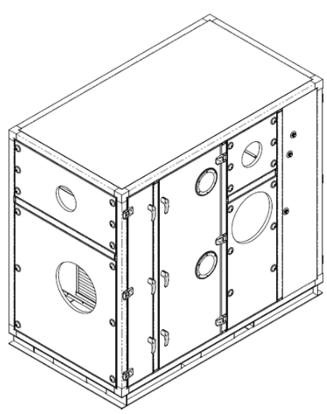

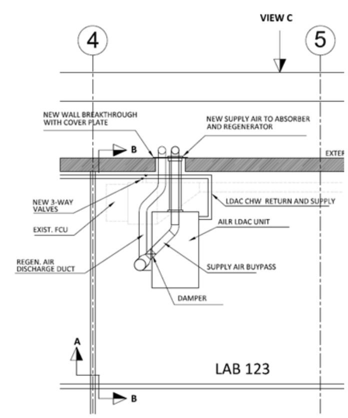

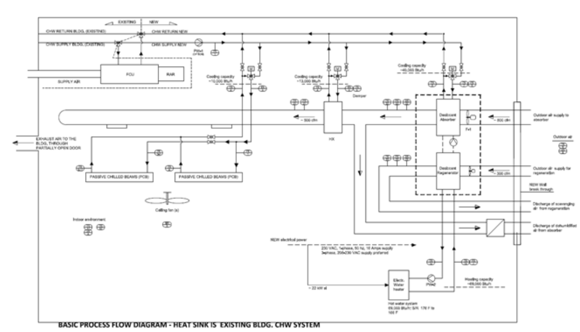

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PROJECT PHASE 1: DESIGN STUDY AND PROJECT SITE SELECTION

PROJECT DELIVERABLE NO. 4

"DESIGN STUDY OF A PACKAGED LIQUID DESICCANT (LD) SYSTEM IN A TEST FACILITY TO CARRY OUT A TEST PROGRAM UNDER LAB CONDITIONS AND SUBSEQUENT ON-SITE TEST OPERATION OF THE LD SYSTEM”

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Prepared for
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November 22, 2017

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ABBREVIATIONS AND UNITS

AC	Air-conditioning
AHU	Air-handling Unit
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
COP	Coefficient of performance
DOAS	Dedicated Outdoor Air Supply
DP	Dew point
DEAC	Direct Evaporative Air Coolers
DEC	Direct evaporative cooler
EAC	Evaporative Air Cooling external source of energy (Q)
HNEI	Hawaii Natural Energy Institute
HR	Humidity Ratio
HVAC	Heating Ventilation Air Conditioning
IEAC	Indirect Evaporative Air Coolers
LD	Liquid Desiccant
LDAC	Liquid Desiccant Air-Conditioning
IEC	Indirect Evaporative Cooler
LiCl	Lithium Chloride
LCST	Lower Critical Solution Temperature
OA	Outdoor (or outside) Air
PV	Photovoltaic System, also solar-PV Power System
RH	Relative Humidity
SHR	Sensible Heat Ratio
SDC	Sustainable Design & Consulting LLC
SI	International System of Units
VAC	Vapor-Compression Air Conditioning
VC	Vapor Compression
WBT	Wet-bulb temperature
VAV	Variable Air Volume

UNITS

atm	Atmosphere pressure
BTU	British Thermal Unit
°C	Degree Celsius
CFM	Cubic feet per minute
°F	Fahrenheit
fpm	Feet per minute
kJ	kIlo Joule
MPa	Mega Pascale

EXECUTIVE SUMMARY

This report presents the preliminary design of a liquid desiccant air conditioning (LDAC) system which will be used for air dehumidification application tests.

The basic premise of testing a liquid desiccant dehumidification system as part of an advanced HVAC system is to prove that the LD dehumidification technology will provide tangible benefits regarding energy savings and providing a high quality indoor environmental quality (IEQ). While IEQ includes a wider range of indoor environmental aspects, this report focuses on IEQ aspects that are generated by the type of space conditioning, and this includes mainly providing good thermal comfort and indoor air quality.

Though sensible cooling loads in high performance buildings are significantly reduced by reducing external and internal heat gains, latent loads remain relatively constant. Therefore, a greater portion of the overall cooling requirement in energy efficient buildings designs shifts to latent loads, particularly in humid climates such as in Hawaii, thus lowering the sensible heat ratio (SHR). Conventional vapor compression HVAC systems face challenges at low SHR values since they operate by suppressing the supply air temperature to below-dew point values to remove humidity contained in the supply air as condensate on cooling coils. This cooling based dehumidification approach results in overcooling buildings, necessitates reheating, and in short increases energy use.

Assuring optimally combined sensible and latent heat load removal, as well as providing required amounts of outside air for space ventilation with the conventional “all-air” HVAC systems, is next to impossible. Overcooling of buildings often creates the need for reheating the supply air. A suitable mitigation is to decouple sensible and latent heat removal by making the removal of humidity independent of cooling the supply air.

The LD dehumidification technology provides the process characteristics for “decoupled” HVAC where humidity control is carried out by the LD system and sensible cooling by a separate HVAC technology. The sensible cooling approach selected for the proposed pilot integrated LDAC system is the passive chilled beam technology, which offers energy efficient operation but requires precise indoor humidity control to avoid condensation and associated humidity related problems.

Previous technology demonstrations tests conducted by NREL and using the AILR LD technology, which is used for the present design study, indicate that significant energy savings can be realized when compared to conventional vapor compression HVAC systems.

An evolving aspect in space conditioning is the focus on benefits from increased health and productivity provided by advanced HVAC applications. Advanced HVAC applications provide financial benefits through increased productivity and reduced illness and absenteeism that may be significantly larger than the financial benefits of energy savings through efficient HVAC systems.

For this report, anticipated benefits of improved IEQ from better indoor humidity control and higher ventilation rates were considered and quantified for a sample office space. The results of this assessment suggest that the financial benefits of improved IEQ due to the proposed LDAC systems significantly outperform the estimated energy savings for the example office space. In the example assessment, increased revenues from more healthy and productive offices spaces and achievable energy savings had an approximately 25:1 ratio. In addition to the financial benefits of the proposed LDAC system over conventional HVAC systems are the reductions in carbon emissions. When using the proposed LDAC system with a thermally-driven rather than an electric chiller the resulting carbon emission are reduced from about 60% to less than 20% of the baseline emission of a conventional HVAC system.

The successful introduction of LDAC technology requires field demonstration and performance testing in a conditioned, smaller-sized space such as office, classroom or library space. The selected approach of proving the validity of using LDAC technology in Hawaii is to a pilot a lab installation to test performance in a controlled lab environment. The initial tests in and the subsequent pilot installation are referred to as Project Phase II/A and II/B, respectively.

For the initial tests in Project Phase II/A, a liquid desiccant (LD) dehumidification unit acquired from AIL Research (AILR) will be used to gain operational experience and to derive important design intelligence for a subsequent pilot installation in Project Phase II/B, at a yet to be selected location. The AILR LD dehumidification unit represents a significant investment and will be used in both Project Phases II/A and II/B.

The AILR LD dehumidification unit for project Phases II/A and B is the same design as a recent product designed and delivered by AILR to a German University, for a similar performance test approach. By using the existing design of the relatively small LDAC system approach, significant costs are saved by avoiding a new design and manufacturing technology. Furthermore, the capacity of the selected AILR LD dehumidification unit is suitable for the planned pilot installation as it is not too large or expensive, and not too small to obtain meaningful practical operational process data. It must be noted that for the initial test in LAB 123 (Phase II/A) the capacity of the ALIR LD unit is exceeds the actual load and therefore a significant portion of the dehumidified supply air must bypass the indoor LAB 123 space and discharge to the exterior. While this is not an energy efficient process, the objective of the initial tests in LAB 123 are to obtain operational experience and design performance data for the subsequent LDAC plot installation.

For test set-up in LAB 123 the heat sink and source for cooling the LD conditioner and heating the LD regenerator convenient, respectively, are provided by the building chilled water supply and an industrial electric water heater. For the pilot installation in Project Phase II/B more energy efficient evaporative cooling and solar heating will be use as desiccant heat sink and heat source.

After an extensive concept design phase, it was decided to install the AILR LD unit inside the room LAB 123 and convey the various air intakes and discharges through a new wall breakout in the exterior LAB

123 wall. This report presents the final preliminary system set-up for the initial test series in LAB 123. The report provides a preliminary cost estimate and compilation of all systems components. The present report also presents the proposed system configuration for the pilot installation.

The report clearly shows the significant benefits of innovative HVAC systems, using liquid desiccant technology, which decouples sensible and latent load removal. These benefits become especially relevant for Hawaii's hot and humid climate. Such innovative HVAC systems avoid energy penalties of overcooling typically associated with conventional cooling coil dehumidification HVAC systems. By avoiding humidity related problems through dedicated humidity control, avoiding overcooling by decoupled sensible cooling load control and providing increased space ventilation for better IEQ the proposed LDAC technology under the proposed system configuration represents a significant advance in HVAC applications.

SECTION 1 - BENEFITS OF LDAC TECHNOLOGY FOR HAWAII

This section provides qualitative and quantitative assessments of the value proposition provided by the proposed LDAC system to Hawaii. The discussion presented in this section stresses the importance of considering both indoor environmental quality and energy savings when designing preferred HVAC systems that comply with Hawaii's sustainability goals. High indoor environmental quality (IEQ) is an important underpinning for the more comprehensive indoor wellness and comfort conditions in high performance or "green" building.

1.1 Emergence of Wellness and Indoor Environmental Quality as Important Metrics of Building Performance

Before the terms "*wellness in buildings* and *healthy buildings*" became widely used in describing building performance, Indoor Environmental Quality (IEQ) was recognized as an important but not yet quantified decision factor in determining performance of buildings. IEQ has been typically overshadowed by other performance metrics in assessing buildings, such as energy performance, first costs and O&M costs for building, outfitting and maintaining a structure.

Over the past several years, high quality IEQ and quantifiable wellness parameters have emerged as more important criteria to assess indoor conditions. Figure 1.1.1 illustrates a typical financial scenario for company office expenditures.

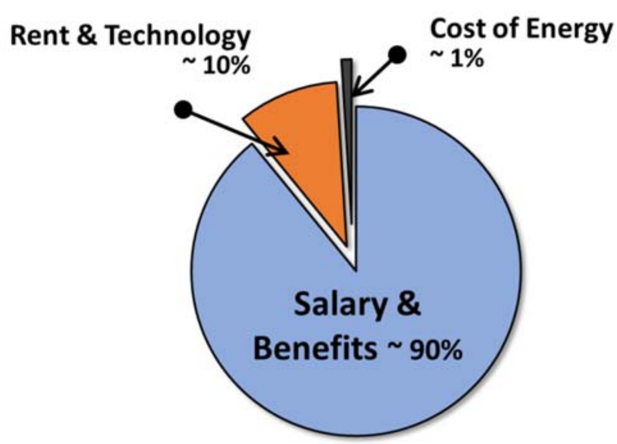


Figure 1.1.1: Typical cost of doing business

The figure was created using data that describe the typical percentages of a company spending for their office operation. The suggested amount for energy "~1%" in the figure might be higher by a factor of 3 to 4 in Hawaii due to the higher energy costs and lower salaries when compared to the rest of the country.

The illustration points out the relative importance of increased productivity of employees compared to the importance of saving energy.

For companies, costs for employees and the revenue which employees can generate are determined by the following main performance elements:

- Salary commensurate with the qualification of the employee
- Quality, quantity and timeliness of employee work output; is determined by the employee's job qualification to carry out the desired work product in a certain time.
- Workplace attendance and absenteeism and the ability to produce on time and budget
- Retention of employees and talent and avoidance of training of new employees
- Cost of hiring new employees
- Socio economic aspects with a new generation (millennials) representing an increasing portion of the workforce

The above-mentioned performance elements are directly and significantly affected by the work environment that is offered to employees. Increasing employee acceptance of the healthy and conducive workplace is directly linked to the financial bottom line of companies. Reduced absenteeism avoids costs and increases revenues. An attractive work environment becomes an essential offering of companies to employees as it supports retention of valuable talent and the acquisition of motivated employees; such employees who based on their qualification are the most mobile part of the work force.

For educational institutions, good indoor environmental conditions are essential for providing a productive learning environment and promoting good test scores and lowers absentees as a result. Both metrics, high academic achievement and high attendance, guarantee the school's standing and funding.

For commercial building owners and operators, a high quality indoor environment facilitates attracting long-term tenants and increases net operating income (NOI) and the cap rate.

Figure 1.1.2 illustrates that IEQ is a subset of health buildings or "Wellness". The present project focuses on the contribution of quality space conditioning on the quality of the built environment. Figure 1.1.2 therefore suggests IEQ being supported by space conditioning and ventilation as a subset of a more comprehensive scope of wellness in buildings.

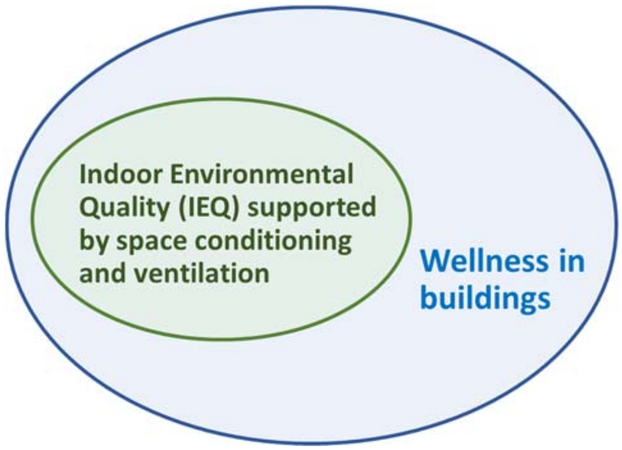


Figure 1.1.2: Interrelationship between IEQ supported by space conditioning / ventilation and a more comprehensive scope of wellness in buildings.

Table 1.1.1. provides a description of the different reaches of IEQ and Wellness initiatives in building. The concepts of “healthy buildings or wellness” have a significantly larger reach than providing good IEQ. The broader reach of wellness and healthy buildings includes psychological and social aspects in addition to the physical IEQ elements. The different wellness elements listed in here are based on the WELL Building Standard®.

Elements of Indoor Environmental Quality (IEQ) are essential to achieve wellness in buildings and healthy buildings as they guarantee that key indoor environmental parameters are satisfied. It can be stated that the presence of favorable IEQ elements in the building alone does not necessarily guarantee a high level of wellness and health indoor living and working conditions; but it can be stated that in the absence of good IEQ, elements wellness cannot be achieved.

Table 1.1.1 further emphasizes five out of six IEQ functions which a quality space conditioning system must supply. In this sense, space conditioning, which includes temperature, humidity and ventilation control, is a key component for an indoor environment that supports wellness.

Table 1.1.1: Comparison between IEQ and Indoor Wellness and Healthy Buildings Strategies

Typical building measures implemented under the following initiatives			
1. Indoor Environmental Quality (IEQ)		**2. Indoor Wellness and Health Interventions**	
*Functions typically inclided in IEQ; functions that are affected by of space conditoing and ventilation are **bolded and underlined**)*		*(A typical scope of indoor measures that supports physical (e.g. indoor environmental quality) AND psychological interventions for benefit of the building occupant)*	
1.1	**Thermal Comfort**	2.1	Thermal Comfort
1.2	**Humidity Control**	2.2	Air Quality
1.3	**Air Quality**	2.3	Acoustic Quality
1.4	**Acoustic Quality**	2.4	Lighting Quality
1.5	Lighting Quality	2.5	Humidity Control
1.6	**Olfactury Comfort**	2.6	Olfactory Comfort
		2.7	Construction Pollution Management
		2.8	Cleaning Protocol
		2.9	Air Quality (advanced filtering & ventilation)
		2.10	Pest Control
		2.11	Water quality
		2.12	Drinking water quality and supply
		2.13	Nourishment - promotion of good food
		2.14	Food Advertising and Offerings
		2.15	Advanced Lighting Design
		2.16	Circadian Lighting design
		2.17	Maximizing Daylighting
		2.18	Right to light (and views)
		2.19	Glare control (solar and light)
		2.20	Promotion of Fitness
		2.21	Ergonomics
		2.22	Noise masking & reduction
		2.23	Beauty and Design
		2.24	Employee & Family Support
		2.25	Biophilia
		2.26	Artwork
		2.27	Operational Transparency

The following briefly describes IEQ elements listed in Table 1.1.1, which are directly affected and controlled by the HVAC system. Only five are briefly described hereafter, whereas the sixth IEQ element, lighting quality, is typically only marginally affected by the type of space conditioning.

1.1 Thermal Comfort: Good thermal comfort is essential for occupants to retain individual thermal equilibrium of their metabolic heat rejection and the surrounding environment. In the absence of thermal comfort, occupants experience physiological stress, which translates into dissatisfaction or, in severe cases, in unhealthy conditions for the individual. The level of thermal indoor comfort in conditioned spaces can be described and qualified by Predicted Mean Value (PMV) and a series of local comfort metrics. Under the PMV model, thermal comfort has two categories of controlling parameters: the objective room specific environmental conditions of temperatures, humidity and air speed, and individual conditions of metabolic rate and clothing insulation.

1.2 Humidity Control: Effective humidity control becomes increasingly important as the sensible heat ratio (SHR) declines because of reduced indoor sensible heat gain as the latent load basically remains constant. Uncontrolled and excessive indoor humidity can damage the building itself, but health risks to occupants are even more dangerous. Humidity problems are increased when liquid water is introduced into the building from either free-running, capillary or diffusion water sources. Humidity problems also occur due to water vapor, generated indoors or introduced through ventilation air. The literature states slightly deviating ranges of “safe” humidity levels, typically given as relative humidity (RH), between 40% and 60% RH.

Building material is best protected against humidity damages by very low RH, which is not healthy for occupants. For occupants, the best ranges of relative humidity are cited typically between 40% and 55%, with 45% reported as an optimum “sweet spot” (Taylor, 2017). Mold, a significant occupant health concern, develops with prolonged surface-RH at or above 80%. It must be noted that surface-RH and representative RH in a space are not the same, and high surface-RH can occur with insufficient ventilation at space-RH levels that are generally considered as “safe”.

1.3 Air Quality: The quality of indoor air is significant to occupant well-being. External and internal sources of air pollutants are responsible for the quality of air. External pollutants enter the building with the ventilation outdoor air (OA) supply, which should be filtered to mitigate problems. Internal air pollutants are either volatile chemical compounds or biological organisms, typically transported as air aerosol or deposited aerosol on surfaces. Elevated humidity levels play a significant role in outgassing of harmful compounds, when airborne transport occurs and the ability of harmful pathogens to grow and attain activity levels increases. While minimum ventilation rates are prescribed in building industry standards (e.g. ASHRAE 61.2) and are often adopted a local code, increased ventilation rates are frequently adopted to increase indoor air quality. Conventional HVAC mostly includes recirculated air systems to enhance the sensible and latent load removal. In this case, a large proportion of the air flow entering the zones is recirculated air (e.g. return air) whereas the proportion of fresh outdoor air is held at a smaller percentage of total air flow. The recirculated indoor air can be a transport vehicle to distribute unwanted air components throughout the ventilation zones. A remedy against unwanted distribution of pollutants of return air is the Dedicated Outdoor Air Supply (DOAS) system, where only fresh outdoor air (e.g. 100% of total air) enters the indoor space and no air is recirculated.

1.4 Acoustic Quality: Indoor acoustic quality is a result of the type of source and propagation of sound. Detrimental sound is referred to as noise, which can cause physical and psychological stress. Noise emitted by HVAC systems are machinery and air flow noise. HVAC machinery noise can be from the cooling equipment, air fans and/or chilled water pumps. While chillers and water pumps are typically located outside the conditioned spaces, air delivery fans are operated inside the zone, potentially generating disturbing noise. Noise of forced air in ducts or at diffusers and registers is typically the most noticeable type of air flow noise. When the air flow rates are high, because of significant sensible and latent loads that need to be removed from a conditioned space, flow noises can be significant. In DOAS systems the amount of air flow and therefore the generated indoor noise is typically much less than in HVAC systems using recirculated air.

1.5 Olfactory Comfort: Olfaction is the sense of smell and olfactory comfort is the perception of air quality based on the experience of smell. Unpleasant smell can have a significant adverse effect on the occupant indoor environmental experience, from mental distraction and stress to unhealthy symptoms such as headaches, skin irritations or asthma attacks. The sources of the smell can be external or internal. Conventional HVAC which recirculates air can worsen olfactory discomfort by distributing impaired air within ventilation zones. Higher outdoor ventilation rates help to dilute odor sources and accumulations. DOAS systems, especially with higher outdoor ventilation rates, are the most effective remedy, provided that the sources of odor are contained or minimized.

The capability of the HVAC system to control indoor temperature and humidity as well as provide sufficient fresh and outdoor air supply, while avoiding exposure to harmful air pollutants and detrimental noise and noise is perhaps the most important assurance of good indoor environmental quality.

1.2 Value Proposition of Increased Indoor Wellness and Comfort

The level of wellbeing found in residential, commercial and institutional buildings can be best described by the benefits it provides to occupants, and the functions occupants carry out.

- Occupants of commercial or institutional office buildings are in working functions and their better health and productivity conditions directly provides a positive financial effect on the companies who employ them. Direct measurable benefits of a more healthy and productive work environment include the avoided cost of health-related absenteeism, and losing and replacing of talent and increased revenues through higher productivity and motivation. Owners of buildings using wellness as a marketing differentiator increase their Net Operating Income (NOI), by retaining quality long-term tenants and avoiding costs for tenant turnover and vacancies.
- Occupants of educational facilities (students, teaching staff, management) benefit through a better learning environment with noticeable increase of academic achievement and attendance.
- Occupants of medical facilities (patients, medical staff, admin.) benefit from less hospital-acquired infection (HAI) cases through a better indoor environment quality. Problems with HAI have been increasing in recent years and they represent a significant financial burden on the health institutions.
- Occupants of residential buildings with a range of wellness and IEQ elements are provided a healthier indoor environment, with the attributable health and comfort benefits.

To provide suitable metrics of indoor wellness and healthy IEQ, the WELL Building Standard has been introduced, which offers a ranking of buildings similar to the well-established LEED Green Building Standard. While the WELL Building Standard® is the occupant focused ranking of the building, the LEED standard is concerned with environmental impacts and their mitigation in the built environment. The WELL Building Standard® provides a point based ranking procedure which leads to several levels of certification. The WELL Building Standard® has seven concepts, which are subdivided into 102 features. Figure 1.2.1 illustrates the WELL Building Standard® and the seven governing concepts.

Figure 1.2.1: WELL Building Standard® with seven concepts (International WELL Building Institute)

Another approach to describe the required elements which produce wellness and a healthy and productive indoor environment is provided by the Healthy Buildings Program at the Harvard T.H. Chan School of Public Health. Figure 1.2.2 illustrates the "9 Foundations of a Health Building" as defined by the Harvard Healthy Buildings Program.

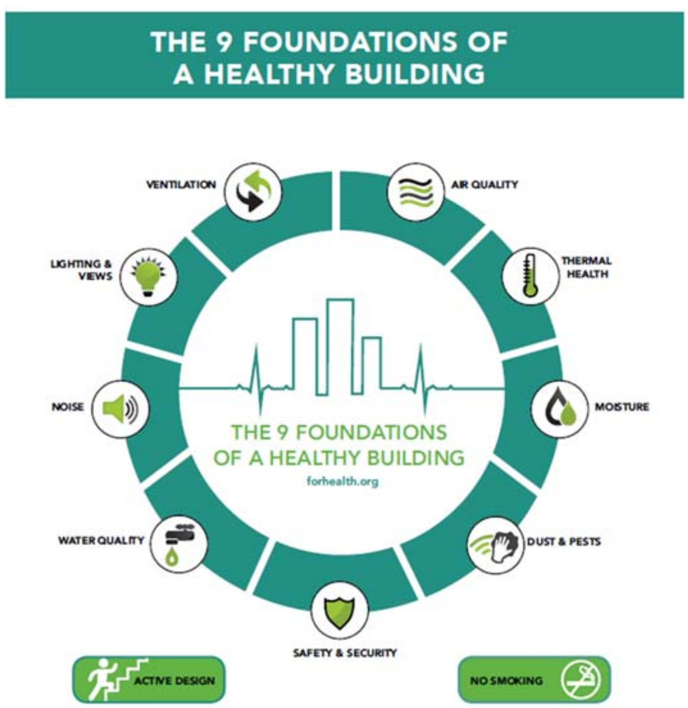


Figure 1.2.2:"9 Foundations of a Health Building" as defined by the Harvard Healthy Buildings Program

Source: Healthy Buildings Program at the Harvard T.H. Chan School of Public Health

Because of the numerous and interacting elements of wellness and good IEQ in buildings, it has been a challenge to clearly attribute certain quantifiable improvement of occupant health and productivity with specific wellness and IEQ elements and derive a framework of financial gain from wellness. There have been several recent publications which have provided the mapping of tangible financial benefits with specific elements of wellness and IEQ in buildings.

Table 1.2.1 shows suggested costs associated with inadequate Indoor Environmental Quality.

Table 1.2.1: Potential Cost Benefit Categories for Different Building Types (Vivian (2016), modified)

Type of costs	Offices	Schools	Hospitals
Facility related costs	O&M, Energy & Water	O&M, Energy & Water	O&M, Energy & Water
Type of health-related costs	Worker Health	Teacher Health Student Health	Patient Health/ Recovery Rates; Staff Health
Metric for health-related loss in staff / employees	Attraction/ Retention	Teacher Turnover	Staff Turnover
Metric for productivity / funding	Individual Productivity	Student Test Scores College Placement	
Metric for health-related employee costs	Absenteeism/ Presenteeism	Absenteeism/ Presenteeism	Absenteeism/ Presenteeism
Measure of organizational losses	Organizational Productivity Market Share/ Customer Speed to Market	Drop-out Rates No Child Left Behind	Bed Vacancies Cost/Bed Profit/Bed
Liability issues	Litigation Insurance/ Tax		Medication Errors

The following examples of increases in tangible health and productivity benefits and/or related financial benefits from improvements of IEQ and wellbeing are quoted from papers by Allen, J. et al (2017), Loftness, Vivian (2016) and World Green Building Council (2016):

Cost associated with <u>inadequate ventilation, thermal comfort and humidity control</u> was describes by the following categories:

- Separate ventilation and thermal conditioning
- Increase outside air, including natural ventilation
- Provide task air and Individual control
- Control pollutant sources
- Improve air filtration
- Control humidity and moisture

- Researchers have linked improved ventilation to ranges of gains in productivity between 8% to 11%, because of increasing outside air ventilation rates about 3 times, dedicated delivery of fresh air to the workstation, and reduced levels of pollutants.
- Short term sick leave was found to be 35% lower in offices by doubling outdoor air supply through increased ventilation.
- As study by Carnegie Mellon concluded that substandard ventilation rates (at half the standard rate) negatively impacted higher order cognitive function of office workers.
- A study in 17 hospitals in Canada identified a 71% reduction in specific infections for employees who worked in patient rooms with a ventilation rate greater than 2 air changes per hour. A 69% reduction in hospital acquired infections (HAIs) among orthopedic and general surgery patients following a ventilation system upgrade that increased outside air exchange rates by 5-9 air changes per hour. The study indicated an annual health savings of about $5,000 per bed while the energy cost increase was $80 per bed.
- For a call center in a tropical climate, 9% improvement (approximately) in operator performance and a 19.5% reduction in headache intensity was identified when increasing the ventilation rates by a factor of 2.5 The resulting annual productivity savings was assessed at $4,000, with an annual energy cost increase of only $9 / employee.
- The number of cases of hospital acquired infections (HAI) decreased significantly when the indoor RH was adjusted to about 42-44% providing the 250 bed hospitals a reduced preventable patient cost of about 7 million USD. (Taylor, 2017). Figure 1.2.3 illustrates the findings of Taylor (2017), which indicates a too low RH, which is frequently found in hospitals, as being very detrimental to patient's health in hospitals.
- A research carried out by World Green Building Council indicated that productivity of office workers was found to be reduced by 4% and 6% at colder and warmer than average room temperatures, respectively.

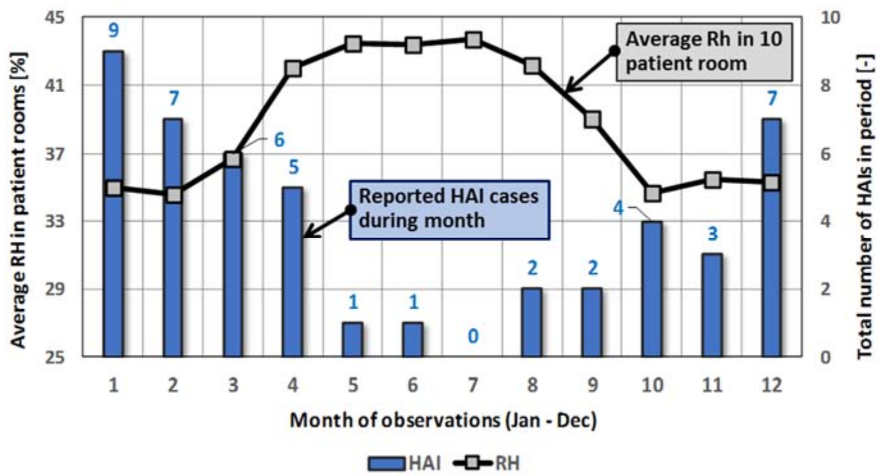


Figure 1.2.3: Relationship between indoor RH and cases of HAI

HAI= hospital acquired infections. The data indicates number of HAI in 10 patient room correlated to average RH.

(Based on Taylor, 2017, recreated by author)

While HVAC related improvements to IEQ and wellbeing of building occupants are of primary importance, other wellbeing elements, such as external views and daylighting, were also proven to positively improve productivity and absenteeism.

- A study indicated that the quality of outside view and daylighting was correlated with a variation of 6.5% of sick leave of employees.
- Workers in a Californian call center were found to increase their productivity in the range of 7% to 12% by having better outside views. Computer programmers with views spent 15% more time on their primary task, while equivalent workers without views spent 15% more time talking on the phone or to one another.
- An 2.7% increase in productivity was identified using environmentally responsive workstations, which provide occupant control of temperature, air supply, task lighting, and sound masking. Annual productivity savings was assessed as about $1,200 per employee.

The following two statements provide an appropriate summary of financial gains through improvement of IEQ and wellbeing in buildings.

A recent Harvard study suggest that substandard ventilation rates negatively impact cognitive function while a doubling in ventilation rates significantly increase higher order cognitive function of office workers. The study concluded the increase in performance is equivalent to *a $6,500 increase in salary per person per year*, while the energy costs of achieving the same change in ventilation were less than $40 per person per year, and down to $1 per person per year when energy efficient systems are used. ***Harvard Health Building Program (Allen, 2017)***

People, through salaries and benefits, represent 90% of an organization's expense and well exceed building costs and energy costs, therefore a small improvement in employee productivity can yield significant value. ***World Green Building Council (2016)***

1.3 Special Consideration of Latent Loads due to Increased Ventilation Rates

The previous section delineates tangible benefits arising from increased ventilation rates. The downside of increased ventilation rates are potentially negative impacts on the indoor environmental quality through increased introduction of external harmful airborne pollutants and increased humidity rates. Humidity related problems, including the presence of indoor mold, are intensified in hot and humid climates with high latent load. Energy for dehumidification must be used to mitigate interior moisture to maintain healthy indoor humidity levels.

According to Kozubal, et al (2014) the shift toward higher latent heat loads 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 to reduce moisture content in the air stream, then reheating it to an appropriate supply temperature, requiring additional energy.

Papers published over the past 15 years, such as Odom (2009) and TIAX (2003), emphasized the shortcomings of conventional HVAC systems to handle larger latent loads and decreasing sensible loads in more energy efficient buildings. A metric to describe the shift in latent load is the sensible heat ratio (SHR), which is the ratio of sensible cooling capacity to total (sensible and latent) cooling capacity. A lower SHR indicates that more cooling capacity is needed for dehumidification. TIAX (2003) suggested that for conventional unitary AC-equipment, SHR-related performance characteristics have changed little, while the building SHR has changed significantly, with latent cooling loads increasing. Conventional unitary AC systems tend to satisfy sensible loads before the latent loads are met.

In Figure 1.3.1 TIAX (2003) illustrates a basic relationship between the building cooling load SHR of conventional unitary AC-system and indoor RH. The authors suggested a maximum indoor RH of 65% to avoid mold growth and a preferred range of 40% to 50% for comfort. (*Note: More recent findings suggest that the surface-RH levels, not the average indoor RH-levels, are the representative RH metric for the likelihood of indoor mold growth. This suggests that, under specific conditions on the surface which support humidity accumulation, mold can grow even at RH-levels below the so-called "safe" RH of 65%*). When the sensible heat ratio of the AC-system does not match the building, the desired indoor RH-levels cannot be met.

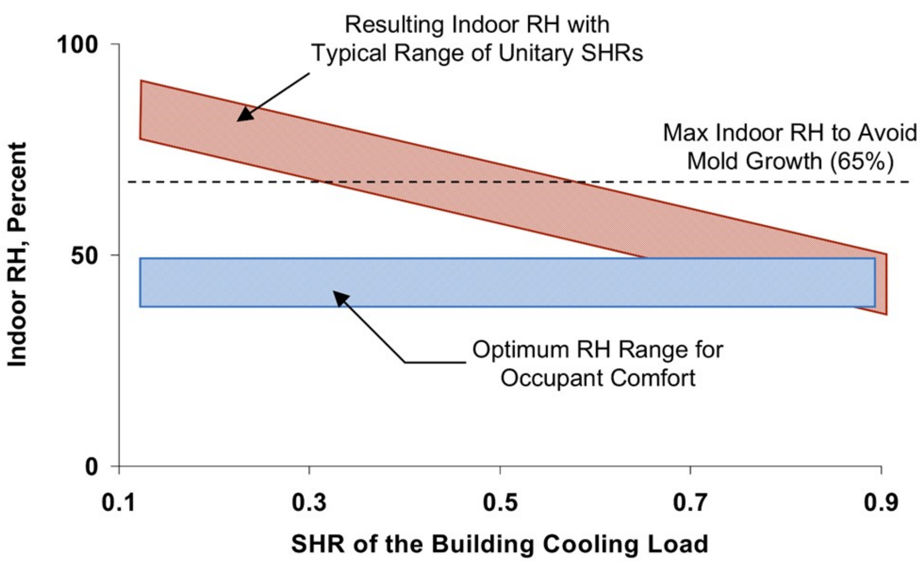


Figure 1.3.1: Indoor RH versus Building Cooling Load SHR with conventional unitary AC equipment

A strategy to reduce the latent loads is to reduce ventilation rates, since most of the latent load is the humidity in outdoor air supply (assuming the building envelope does not allow free flowing water or water through diffusion to enter the building). Between 2001 and 2007, for example, the required ASHRAE minimum outdoor ventilation rates have been significantly reduced; in the case of office and lecture spaces by 75% and 50%, respectively.

In the following section, a basic calculation model is used to illustrate the reduction of SHR with larger ventilation rates. As noted before, the latent load is primarily supplied by humidity contained in the outdoor air supply, and secondarily by indoor humidity sources, with occupants being the main contributors. Table 1.3.1. shows the results of the calculation model, using a generic 25,000 sqft office building and average sensible and latent cooling load assumptions reported by the Florida Solar Energy Center as the baseline building. In Table 1.3.1 two scenarios were reported, (A) baseline cooling loads and (B) a 30% reduction in sensible cooling loads was applied. The sensible and latent cooling loads were distinguished as being either independent or a function of ventilation.

Table 1.3.1: SHR in example building, baseline sensible cooling loads and 30% reduction in sensible cooling loads, values shown represent a baseline ventilation rate

	Typical office building (*) baseline sensible energy use				Typical office building (*) 30% improvements in sensible energy use			
	Ratios	Total	Independent of ventilation	Function of ventilation	Ratios	Total	Independent of ventilation	Function of ventilation
		kBTU	kBTU	kBTU		kBTU	kBTU	kBTU
Sensible cooling load envelope Internal load Ventilation (**)	**65%**	530	490	40	**57%**	380	340	40
Latent cooling load Internal load Ventilation (**)	**35%**	280	70	210	**43%**	280	70	210
sum >>>		810	560	250		660	410	250
SHR >>>	65%			**SHR >>>**	57%			

(*) *Example office building of 25,000 sqft*
(**) *Some infiltration independent of ventilation was considered*

Kozubal, et al (2014 suggested that alternative dehumidification strategies have been previously employed to a certain extent in the HVAC industry. These systems incorporate the actively or passively regenerated solid desiccant rotors (enthalpy wheel) to enhance the removal of water from the air. The downside of these solid desiccant systems is their large dimensions, which limit spatial flexibility, and the relatively high fan energy consumption due to the increased airside pressure drop of the rotors. Another alternative dehumidification strategy used for several decades uses high-flow liquid desiccant systems. These high-flow LD systems require high-maintenance mist eliminators to protect the air distribution system from corrosive desiccant droplet carryover. These high maintenance costs are the reason why their use have been typically restricted to specific industrial applications, where low dew point ventilation is required.

The more recent development and commercialization of low-flow liquid desiccant air-conditioning (LDAC) technology for general HVAC applications 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 under development. The common key feature among all of the designs is that LDAC effectively dry supply air without the need for overcooling and reheating.

As concluded in Section 1.2, increasing ventilation rates is beneficial to the Indoor Environmental Quality (IEQ) and, therefore, create a more healthy and productive environment. While buildings with energy conservation measures can significantly reduce the sensible cooling loads, latent cooling loads are not reduced to the same extent, since most humidity is admitted to the indoors via the ventilation air. This decreases the sensible heat ratio (SHR). Higher ventilation rates further decrease the SHR values. Conventional unitary AC systems are typically not designed to operate at such low SHR values. Between 2001 and 2007, a remedy was introduced by significantly lowering the required ASHRAE minimum ventilation rates to keep the SHR at a higher level.

A decrease of ventilation rates does not help to achieve high IEQ or create healthy and productive indoor conditions. A remedy for curbing high indoor humidity rates at a low SHR is the use of dedicated dehumidification systems, where sensible and latent cooling loads can be decoupled and their individual removal can be precisely controlled. At low SHR values, the conventional cooling based dehumidification approach requires reheating the supply air or risking overcooling of the space. The use of desiccant systems provides an efficient approach to control indoor humidity independently from the sensible cooling loads.

There are several technical considerations that negatively affect the performance conventional cooling coil based dehumidification, especially under the premise of low Sensible Heat ratio (SHR).

- In conventional AC application units, the sensible cooling load is served first, and the cooling load modulation is controlled by means of temperature set-points, which function as under a min/max thermostatic control function. This control procedure causes ON-OFF cycles of the cooling capacity as well as part-loads conditions of the cooling coils (Shirey, 2006).
- In energy efficient buildings, where sensible heat gain is reduced by means of better heat envelope performance, the reduced sensible cooling load causes longer ON-OFF cycle periods. These ON-OFF cycles degrade the dehumidification performance as in the OFF cycle since wetted coils release moisture from water surface films into the passing ventilation air. In addition, the dehumidification effectiveness is further reduced by moist ventilation entering the conditioned space without having passed over below-dewpoint coils. Figure 1.3.3 illustrates the interaction of condensation and subsequent evaporation of wet cooing coils operating in ON-OFF cycles. Shirey suggested that this process can greatly reduce dehumidification effectiveness and can cause moisture related problems in buildings, thus resulting in health risks and potential damage to the building itself.

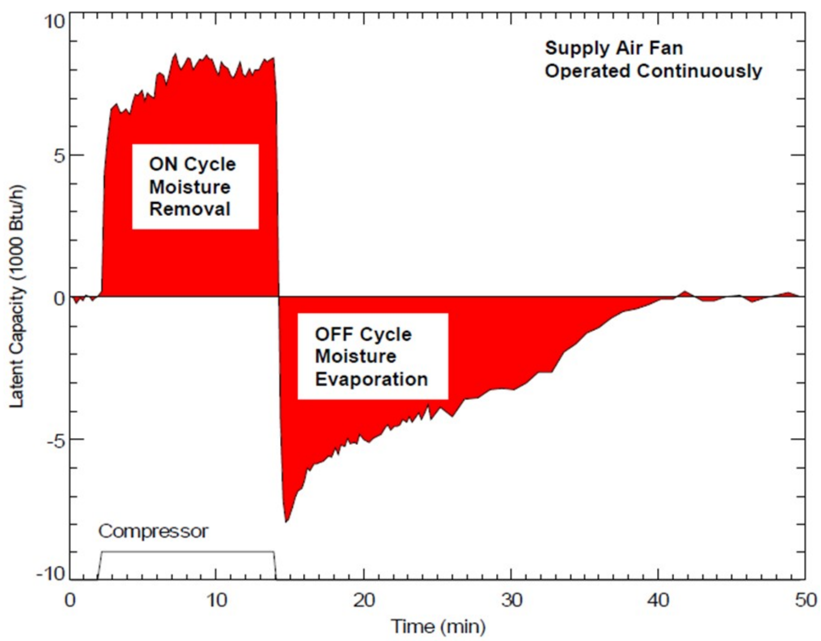


Figure 1.3.3.: the interaction of condensation and subsequent evaporation of wet cooing coils operating in ON-OFF cycles (Shirey,2006).

- Since cooling based dehumidification requires low air temperatures for effective removal of moisture, reheating the supply air becomes a requirement when the internal sensible heat gain cannot accommodate low temperature supply air temperatures through mixing. Reheating is energy intensive and expensive if no low-costs heat sources are available, such as natural gas or sufficient waste heat. Several recuperative heat exchange technologies are available that can save energy and reduce cooling as well as increase dehumidification capacity (Gatley, 2000). The recuperative technologies and processes are typically working best at sensible heat ratios (SHR) above 0.7. In applications with lower SHR values, which means higher latent cooling loads, reheating or active regenerated desiccant systems become a necessity.

Conventional cooling based dehumidification HVAC systems face technical challenges with compensating the cooling load for dehumidification and overcoming dehumidification degradation under part load conditions. These challenges are increased significantly with low SHR values caused by declining sensible and increasing latent cooling loads. With higher ventilation rates required for healthy and productive indoor conditions, there is a point when conventional HVAC can no longer effectively provide simultaneous dehumidification and sensible cooling. Dedicated outside air systems (DOS) with desiccant based dehumidification is a mitigating technology whose approach provides decoupled sensible and latent cooling loads in the appropriate proportions to safeguard energy efficient building HVAC operations.

1.4 Description of the Energy Performance of the Proposed Liquid Desiccant System

This section discusses a qualitative comparison of the energy performance of the proposed liquid desiccant (LD) cooling system with a conventional HVAC system using standard cooling coil dehumidification. Figures 1.4.1 and 1.4.2 show the conventional and the advanced desiccant HVAC system, respectively.

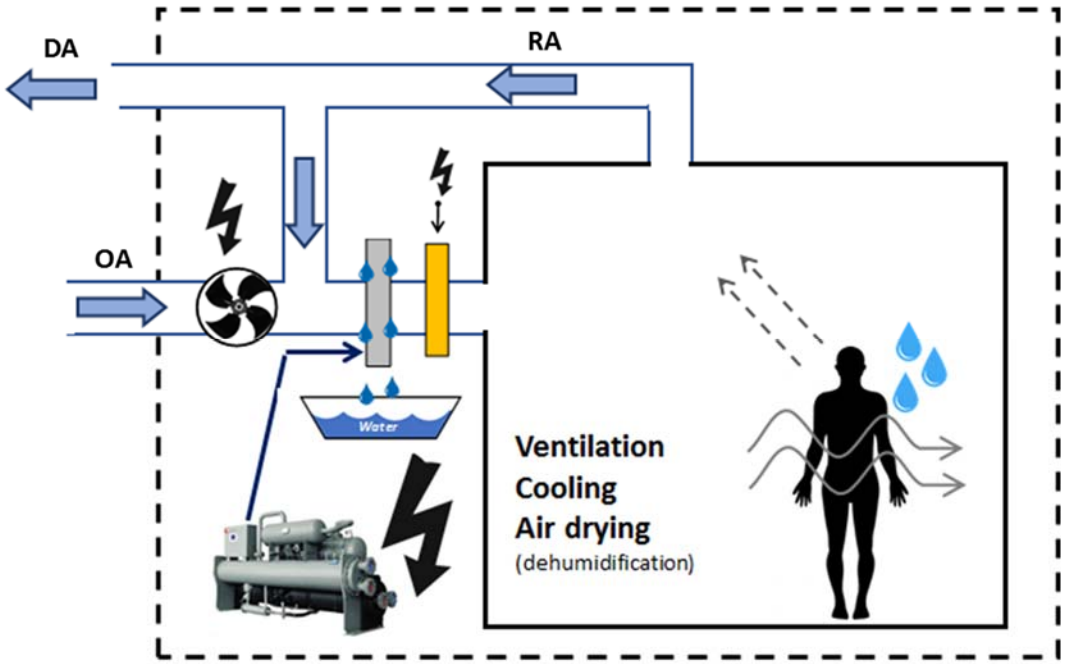


Figure 1.4.1: Conventional HVAC system – sensible cooling and humidity reduction through chilled air

Figure 1.4.1 shows a basic process where outdoor air (OA) is mechanically forced (e.g. fan) over cooling coils which extract sensible heat and remove humidity at below-dewpoint temperatures. Since the sensible heat ratio (SHR) is low, reheat must be provided to avoid overcooling. About 80% to 90% of the return air (RA) is flowing back to the cooling coil. The remaining part of RA is expelled from the building as discharge air (DA). Energy is used for the chiller function and the supply fan. It is assumed that the depicted HVAC system is a unitary AC system.

Figure 1.4.2 depicts the proposed liquid desiccant system. Outdoor air (OA) is mechanically forced (e.g. by fan) through the internally cooled desiccant absorber ("desiccant drying"), where humidity is removed from the supply air. During desiccant drying of the supply air, absorption heat is released, which is rejected through a small evaporative cooling tower. The saturated desiccant is pumped to the regenerator, where the absorbed humidity is released through applying heat energy.

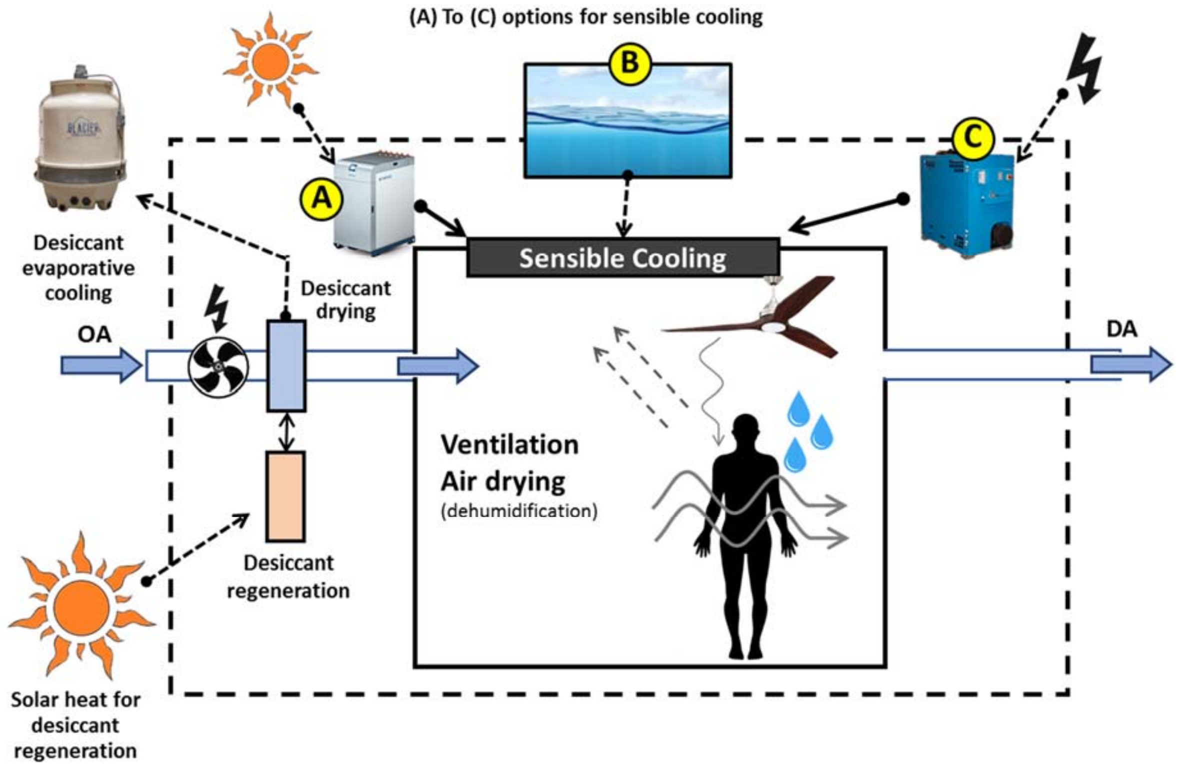


Figure 1.4.2: Advanced AC – Desiccant dehumidification & separate sensible cooling; options for sensible cooling can be (A) thermally operated adsorption chillers, (B) cold deep seawater cooling, or (C) conventional vapor compression chillers.

The system uses a dedicated outdoor air supply (DOAS) without return air. The discharge air (DA) leaves the conditioned space and removes quantities of the sensible heat as well as all internal latent loads through convection. The indoor sensible heat can be rejected to heat sinks using a range of sensible heat sinks (A) through (C), as follows:

(A) To further reduce energy consumption, sensible heat rejection can be provided through **thermally operated adsorption chillers**. Adsorption chillers have lower maintenance requirements and a longer operational lifetime than the more common absorption chillers. The preferred heat source for the adsorption chillers is solar heat, but waste process heat can also be used.

(B) Sensible heat rejection can be provided **through seawater AC systems (SWAC**). In SWAC applications, cold deep seawater is pumped from a depth that corresponds to the required temperature for the heat rejection. The heat gains to the cold deep seawater along the transport through pipes and heat exchangers must be accounted for. Figure 1.4.3 shows a typical relationship between seawater temperature and depth; the temperature versus depth distribution depicted in the figure is characteristic for low latitude ocean regions such as Hawaii. The figure indicates a required depth of about 2,000 to 2,500 feet (*a)* for cold seawater delivered to a district or institutional system where cooling coil based dehumidification is used in buildings. The cooling system which could serve decoupled sensible and latent heat rejection systems (e.g. desiccant dehumidification) can work with significantly shallower seawater intakes, about 600 to 800 feet depth (*b*). This lower intake depth saves significant costs for the cold seawater pipe since these costs are strongly dependent on the depth and length of the pipe to reach cold seawater temperatures. For some situations, using a solar heat driven desiccant dehumidification system with sensible cooling derived from deep seawater is may be a promising and cost-effective HVAC system for Hawaii.

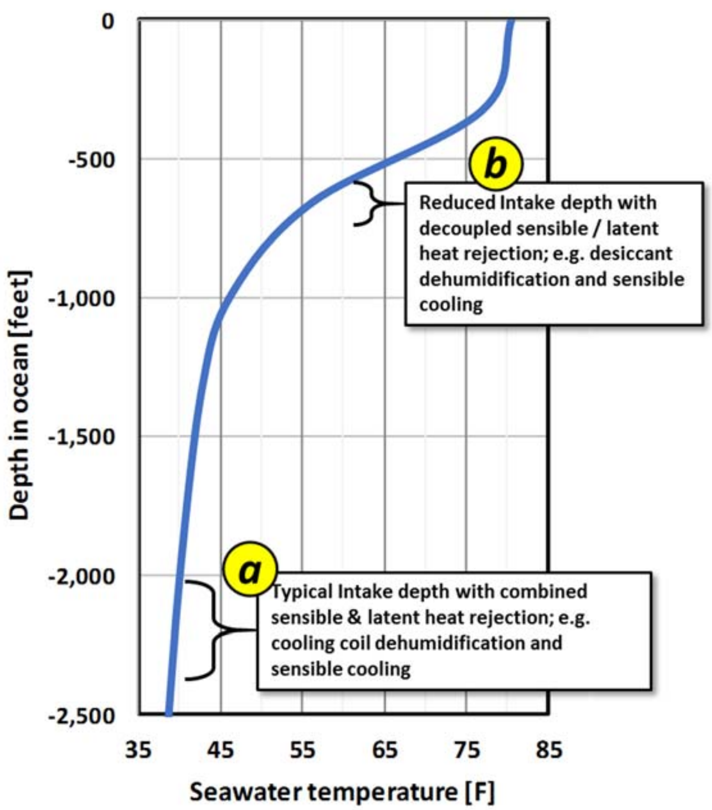


Figure 1.4.3: Seawater AC (SWAC) - Relationship between ocean depth and water temperature in the tropical ocean

(data source: SOEST, University of Hawaii website)

The figure shows two scenarios of seawater AC:

(A) SWAC with conventional cooling coil dehumidification; intake of SW about 2,000 to 2,500 feet depth
(B) Separate dehumidification and sensible cooling; intake of SW about 600 to 800 to 2,500 feet depth

Locating the deep SW intake at a shallower depth saves significant costs for the SW system

(C) Sensible heat rejection can be provided through conventional vapor compression refrigeration cycles. The vapor compression cycle works more efficiently when operating at higher chilled water temperatures than those temperatures used for cooling based dehumidification. Energy savings of up to about 20% - 30% can be realized by the higher evaporator or chilled water temperatures.

Table 1.4.1 shows a comparison of basic energy performance metrics between of the proposed HVAC system with desiccant dehumidification and a conventional AC system with cooling based dehumidification. The table suggests that the proposed HVAC system based on desiccant dehumidification has significant benefits to Hawaii as it can cover most of its energy through renewable energy, leaving only a small portion of the energy demand as electricity to power fans and pumps. The conventional HVAC system based on cooling coil dehumidification, on the other hand, relies entirely on electricity to drive the system's chiller, pumps and ventilation fans.

Table 1.4.1: Comparison of basic energy performance metrics between of the proposed HVAC system with desiccant dehumidification and a conventional AC system with cooling col dehumidification

<table>
<tr><th colspan="3">Liquid Desiccant based dehumidification with decoupled sensible cooling</th><th colspan="3">Conventional AC with cooling coil dehumidification</th></tr>
<tr><td>Description</td><td>preferred type of energy used</td><td>Electrcity or Renewables</td><td>Description</td><td>preferred type of energy used</td><td>Electrcity or Renewables</td></tr>
<tr><td colspan="3">Sensible cooling</td><td colspan="3">Sensible cooling</td></tr>
<tr><td>(A) thermally operated adsorption chillers</td><td>Heat energy (either solat or waste process heat) is converted to cooling capacity</td><td>Renewables</td><td rowspan="3">Sensible cooling by supply air passing over cooling coils</td><td rowspan="3">Mechanical compression of working fluid and subsequent expansion created heat sink</td><td rowspan="3">Electricity</td></tr>
<tr><td>(B) Seawater AC (SWAC) system</td><td>Cooling capacity of deep cold seawater is used to reject heat</td><td>Renewables</td></tr>
<tr><td>(C) Conventional vapor compression</td><td>Elevated evaporator temperature is more energy efficient</td><td>Electricity</td></tr>
<tr><td>Ancillary process</td><td>Pumps and fans</td><td>Electricity</td><td>Ancillary process</td><td>Pumps and fans</td><td>Electricity</td></tr>
<tr><td colspan="3">Dehumidification (latent heat rejection)</td><td colspan="3">Dehumidification (latent heat rejection)</td></tr>
<tr><td>Desiccant absorber (removal of humidity)</td><td>Evaporative cooling tower</td><td>Renewables</td><td rowspan="2">Latent heat removal by supply air passing over below dewpoint cooling coils</td><td rowspan="2">Mechanical compression of working fluid and subsequent expansion created heat sink prdices</td><td rowspan="2">Electricity</td></tr>
<tr><td>Desiccant regenerator</td><td>Solar heat or process waste heat</td><td>Renewables / recycling waste energy</td></tr>
<tr><td>Ancillary process</td><td>Pumps and fans</td><td>Electricity</td><td>Ancillary process</td><td>Pumps and fans</td><td>Electricity</td></tr>
</table>

Given that Hawaii's future energy goals promote a transition to more and more renewable energy, using thermal instead of electric energy increases the energy mix and thus creates value. It also avoids the wasteful conversion of electric energy to heat energy for thermal processes.

SUMMARY

A HVAC system using liquid desiccant (LD) dehumidification and decoupled sensible and latent heat removal is an attractive alternative to conventional HVAC systems with cooling based dehumidification. This is especially true when large latent cooling loads must be removed for increased ventilation, such as in Hawaii's hot and humid climate. The state's energy goals to drastically increasing renewable energy forms and reduce carbon footprint makes the LDAC very suitable for larger utilization in Hawaii since the system can mostly powered thermally through renewables. In addition to the energy advantages, the liquid desiccant systems have the following advantages:

- **Can provide high indoor air quality and thermal comfort through a favorable combination of humidity levels and dry bulb temperatures**
- **Effective in removing large amounts of moisture, even at high ventilation rates, and avoids humidity related problems in buildings**
- **Lowers operations cost by significantly reducing O&W costs**
- **The evaporative medium through which the outside air is conveyed before it is introduced to the conditioned space acts as an effective biocide and particle filter, thus avoiding external pathogens from being introduced through the outside air.**
- **The activated liquid desiccant solution is an effective energy storage to provide latent cooling capacity during hours outside the operating envelope of the renewable energy sources.**

SECTION 2 - DESCRIPTION OF THE LDAC TECHNOLOGY USED FOR PILOT TESTS

This section describes the liquid desiccant technology that will be used for the present project. While liquid desiccants have been used for specific drying and dehumidification applications for several decades, the use of liquid desiccants in generic HVAC systems is a rather recent and innovative technology.

2.1 Liquid Desiccant Technology by AILR Research

The liquid desiccant (LD) dehumidification technology used for this project was developed and is being produced by AIL Research Inc. This relatively recently developed LD technology uses low-flow desiccants processes and internally cooled and heated absorbers and desorbers. The new LD technology represents a significant advantage of conventional LD dehumidification technology.

Conventional LD dehumidifiers have been used in selected industries for high demand air drying applications for many decades. The use of LD technology in general HVAC applications is relatively new. The fundamental difference between LD dehumidification and conventional cooling based dehumidification is that supply air does not have to be chilled to below dew-point temperatures to remove humidity. This reduces energy demand and avoids overcooling of the conditioned spaces, or required reheating.

Figure 2.1.1 is a psychrometric chart that illustrates possible energy savings by using the LD technology instead of conventional cooling based dehumidification. Figure 2.1.1 (obtained from the AILR website) shows typical supply air conditions at 65F and 50% relative humidity (RH), which represents a dew point of 45F. With outside conditions of 86F and 70% relative humidity, the LDAC system would require about 35% less heat energy removal and addition, when accounting for the reheating necessary to compensate from overcooling.

The ability to precisely control humidity independently of the required sensible cooling load is the main advantage of the desiccant technology. This makes it possible to use advanced HVAC technologies, such as displacement ventilation, chilled beams, and radiant panels, as part of an energy efficient HVAC system that separates latent from sensible load.

The AILR LD system uses strong solutions of the ionic salts lithium chloride and calcium chloride. These ionic salts have the attractive characteristic that the salts themselves have essential zero vapor pressure, and therefore evaporation of the desiccant solution will not occur, and vapor will not appear in the air supplied by the LDAC. The liquid desiccant solution forms a strong bond with water molecules, which is a stronger bond than that between molecules in pure liquid water. Therefore, heat released when water condenses on the desiccant will be approximately 7.5% larger than heat of fusion in water. The released absorption must be rejected from the process to maintain good humidity absorption performance, as higher desiccant temperatures increase the water vapor pressure at the desiccant surfaces, thus reducing the ability to absorb water vapor.

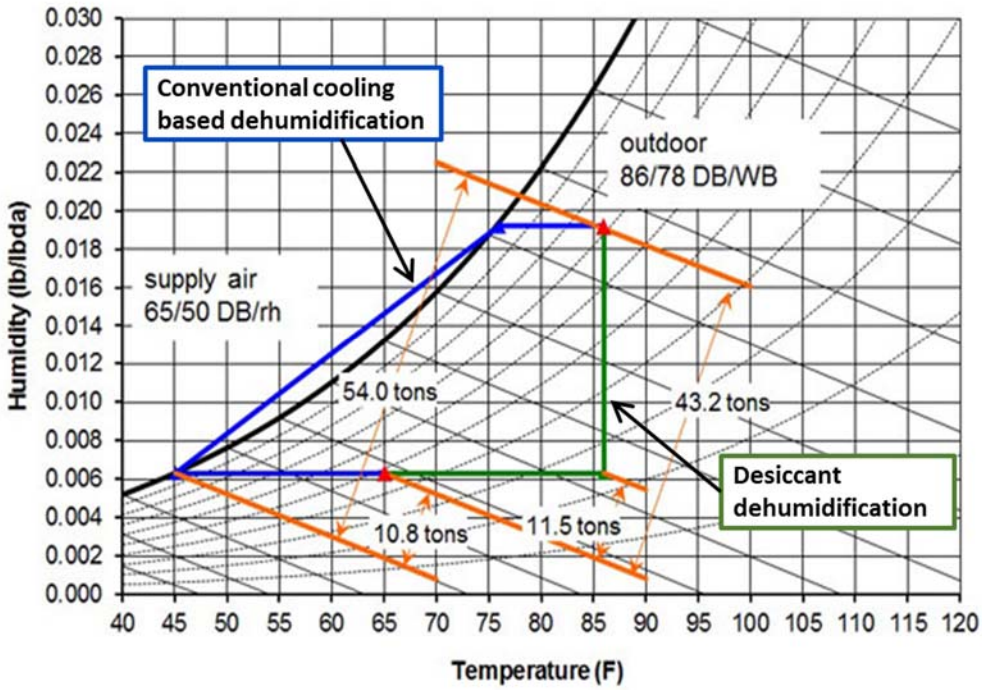


Figure 2.1.1: Comparison of psychrometric performance for cooling and dehumidification between a conventional DX AC system and a LDAC system; (source: Kozubal,et al ,2014, modified)

Conventional LD technology uses high desiccant flow rates and external cooling. The AILR LD technology, on the other hand, uses internally cooled absorbers and much lower desiccant flow rates. This mitigates potential “carry-over” of desiccant to the air supply and other problems that have historically been barriers to the wider adoption of the LD-technology in general HVAC applications.

Figure 2.1.2 shows the basic process diagram of the AILR LD-technology. The "low flow" LD system has three main components: Conditioner, Regenerator and Interchange heat exchanger (IHX). The conditioner is a parallel-plate liquid-to-air heat exchanger. The conditioner is cooled internally with cooling water, typically from a cooling tower water or other heat sink. Desiccant solution flows down the outer surfaces of the plates. The supply air flows horizontally through the gaps between the plates. As the humid air comes into contact with the desiccant, water vapor is absorbed by the desiccant. The heat released by this absorption process is transferred to the coolant inside cooling coils. The air leaves the conditioner much drier, although its temperature might higher than the supply air due to the heat of absorption.

As the desiccant solution absorbs humidity, it becomes diluted and its capacity to absorb humidity needs to be restored through heat application in the desiccant regenerator. The regenerator is configured similarly to the conditioner, which is a parallel-plate liquid-to-air heat exchanger where hot heat transfer fluid, typically hot water, flows within the plates. The heating water can be supplied by a gas-fired boiler, solar thermal collectors, recovered heat from an engine or fuel cell, or other suitable energy source. As the temperature of the desiccant increases, it releases water into a "scavenging" air stream that is discharged outdoors.

The hot, concentrated desiccant that leaves the regenerator and the cool, dilute desiccant that flows to the regenerator exchange thermal energy in the interchange heat exchanger. This exchange increases the efficiency of the regenerator and decreases the cooling load on the conditioner.

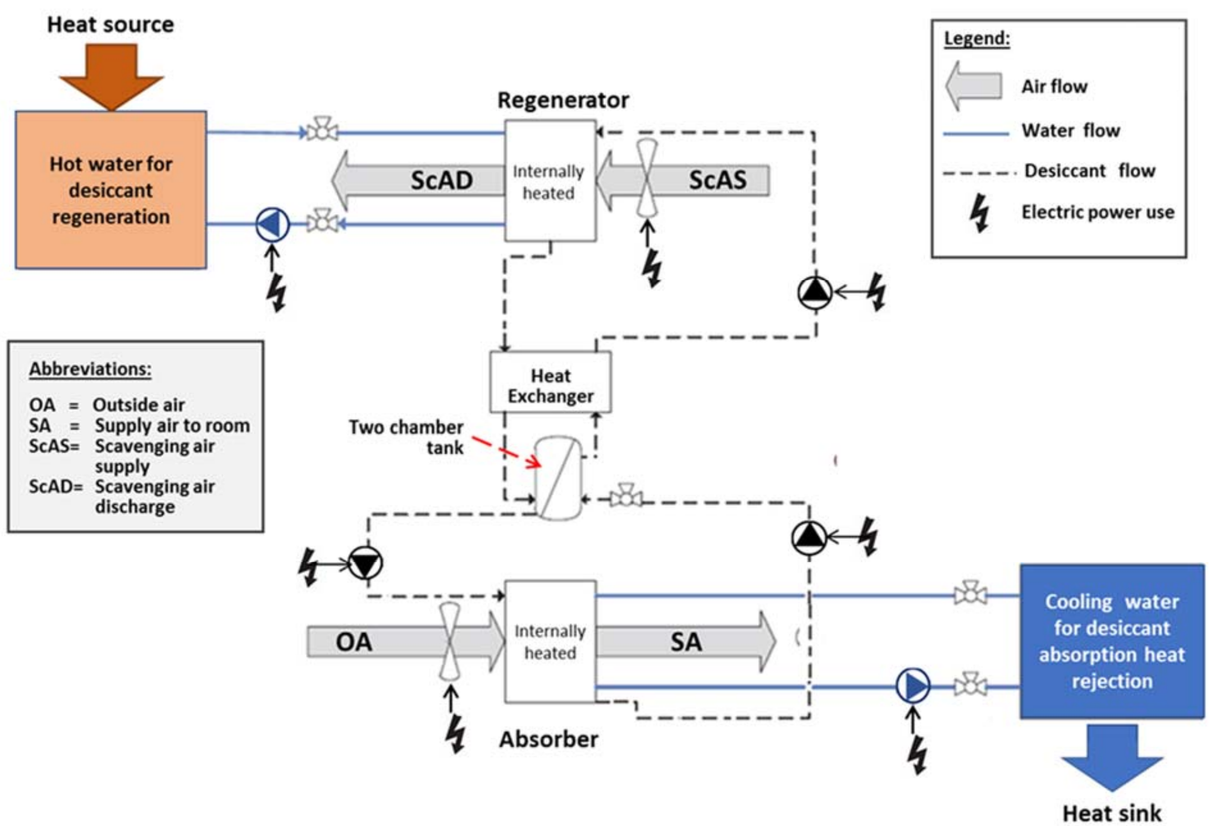
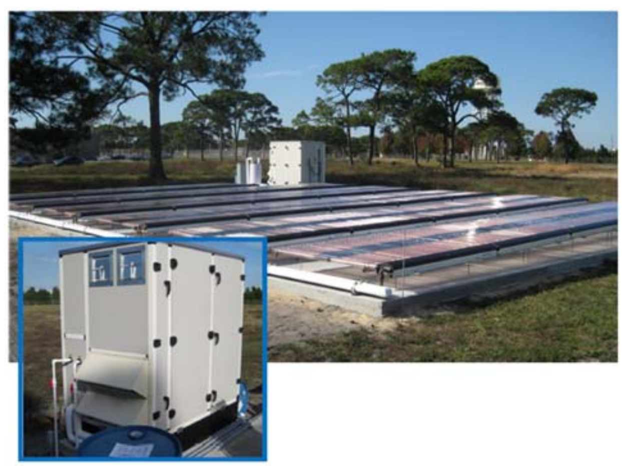


Figure 2.1.2: Basic process illustration of the AILR LD system

2.2 Previous Installations of AIRL Thermally-Driven LDACs

This section presents several previous installations and demonstration tests of thermally-driven LDAC systems. The systems presented hereafter represent the same type of thermally-driven LDAC that will be used for the proposed HNEI tests.

Year / Location / Description	Photo of AILR thermally-driven LDACs installation
2006 - University of Maryland: The first field test of AILR's low-flow liquid desiccant (LD) Technology. Under a contract with the National Renewable Energy Laboratory (NREL), AILR built, installed and operated a 2,400 cfm liquid desiccant conditioner that received concentrated desiccant from a Kathabar regenerator. Heat recovered from an engine-generator set was used to regenerate the desiccant	
2006 - Wrightsville, PA: Under funding from NREL, AILR built a 6,000 cfm gas-fired LDAC that provided ventilation air to a machine shop located at the Donsco Cast Iron Foundry (Wrightsville, PA). The LDAC operated during the summers of 2007 and 2008.	

Year / Location / Description	Photo of AILR thermally-driven LDACs installation
2007 - Kingston, Ontario: AILR delivered a 3,000 cfm, thermally-driven LDAC to Queen's University (Kingston, Ontario). An array of solar collectors was installed in 2011 to provide a portion of the thermal energy required for desiccant regeneration.	
2009 to 2010 – Seal Beach and Tustin, CA: From January 2009 through April 2010, AILR worked with a venture-backed start-up (Pax Streamline) to commercialize thermally driven LDACs applied to supermarkets. Two LDACs were built at AILR and installed at supermarkets in California (one in Seal Beach and the other in Tustin). Despite the successful operation of the LDACs, Pax failed to raise a second round of financing and they ceased operation in April 2010.	
2011 - Tyndall Air Force Base: As a subcontractor to NREL under DOD's ESTCP program, AILR delivered a 3,000 cfm LDAC to the Tyndall AFB. The LDAC operated during the summers of 2010 and 2011on hot water provided by a 1,350-square foot array of evacuated-tube solar collectors.	

Year / Location / Description	Photo of AILR thermally-driven LDACs installation
2010 - Maui, HI and Huntington Beach, CA: AILR supplied core LD components to an angel-backed startup (J&J Mechanical) for two 6,000 cfm LDACs. Both LDACs were installed in supermarkets, one in Maui, HI and the other in Huntington Beach, CA.	
2011 through 2013 - Hoboken, NJ, Encinitas, CA and Kailua, HI: AILR worked under a contract to NREL to build, install and test four LDACs. Two LDACs were sited at the Stevens Institute of Technology (Hoboken, NJ), one at a natatorium and the other at an administrative building. AILR was assisted by the Munters Corporation, who supplied the regenerator for the second of these two LDACs The third LDAC operated at a supermarket in Encinitas, CA. This LDAC rejected heat to a chilled water cooling loop. The fourth LDAC operated on a supermarket in Kailua, HI. A significant portion of the thermal energy for this LDAC was provided by a 3,500-square foot solar array.	 Stevens Institute of Technology, Hoboken, NJ  Whole Foods supermarket Kailua, HI

Year / Location / Description	Photo of AILR thermally-driven LDACs installation
2017 – Kassel, Germany: In July 2017 AILR delivered a 1,200 cfm LDAC to the University of Kassel. The LDAC is now being prepared for laboratory operation at the university. Field operation is expected later. **<u>The proposed LDAC unit for the present project will be the same design as the "Kassel" design.</u>**	

2.3 Energy Performance of Previous Installations of AILR Thermally-Driven LDACs

A comparison of energy performance of a conventional packaged rooftop unit (RTU) and the thermally driven LDAC system was performed by NREL (Kozubal, et al, 2014). (The complete report is attached in APPENDIX **C**.).

The report stresses that as sensible loads decrease, latent loads remain relatively constant, and thus latent cooling loads become a greater fraction of the overall cooling requirement in highly efficient building designs, particularly in humid climates. The paper stresses that conventional HVAC systems do not perform well in buildings with low sensible heat ratio (SHR). Liquid desiccant applications have several advantages over solid desiccant applications. However, solid desiccant systems have a longer track record in HVAC applications, whereas liquid desiccants have been used primarily for specific industrial and institutional dehumidification applications.

Kozubal, et al (2014) suggest that low-flow liquid desiccant air-conditioning (LDAC) technology, which will be used for the present system, several 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 conventional high-flow LD dehumidification systems.
- Have smaller system sizes and allows more flexible configurations than solid desiccant systems.
- Reduces the desiccant droplet carryover problem, thereby reducing maintenance requirements compared to conventional high-flow LD systems.
- Liquid desiccant systems can shift latent loads to times when energy is cheaper and/or renewable or waste energy is abundant. Storing activated liquid desiccant is a relatively inexpensive form of energy storage; correctly speaking it is not energy that is stored but the capacity to dehumidify, which requires significant energy in conventional vapor compression systems.
- Liquid desiccants remove biological and chemical pollutants from process air streams and perform as highly effective air filtration and purification systems, thus improving indoor air quality.

The report (Kozubal, et al, 2014) carried out an estimation of energy savings that can be achieved by using LDAC systems in lieu of conventional HVAC. The results are illustrated in Figure 2.3.1. The identification of cases is presented in the complete report (see APPENDIX D).

Kozubzl performed energy modeling to provide estimates of the savings available with LDAC supermarket applications across the United States. Supermarkets were identified as having especially high dehumidification requirements. Figure 2.4.1 compares the energy performance of LDAC with baseline models (conventional RTUs) with 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.

The results suggested that for a supermarket in the hot humid climate zones 1A and 2A, which are representative of Hawaii climate, with high latent loads requiring 4,000 cfm of ventilation, energy cost savings ranged from $3,000 to $30,000, which corresponded to 1% and 6% of the different supermarkets whole building energy expenditure. The space conditioning savings were realized because of the avoidance of significant energy demand associated with overcooling and required reheat, which are typical for conventional DX system.

The report identified significant additional achievable savings with the use of a two-stage regenerator, which is still under development. A two-stage regenerator can save 40% of the thermal energy required for single-stage regeneration. With a two-stage regenerator, total HVAC energy savings in hot humid climates can range from 34%-57%. These saving will be greatest in climate zone 1A, which is dominated by cooling loads.

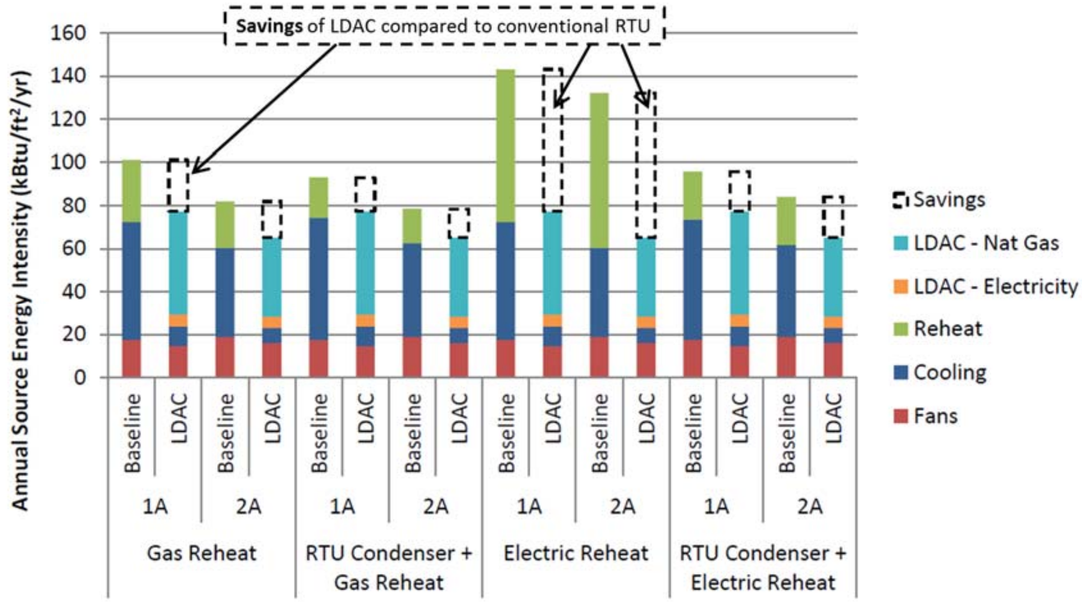


Figure 2.3.1: Annual source end use energy consumption and savings – **Single-stage regenerator** (kBtu/ft2/yr) ; low-flow liquid desiccant air-conditioning (LDAC) and roof top unit (RTU) ; *source (Kozubal, et al, 2014) modified*

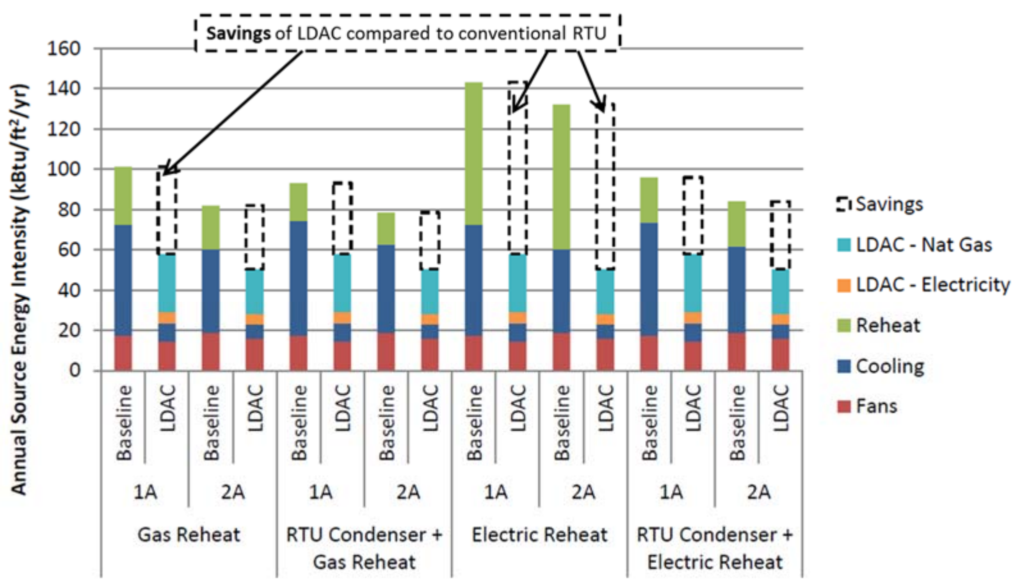


Figure 2.3.2: Annual source end use energy consumption and savings – **Two-stage regenerator** (kBtu/ft2/yr) ; low-flow liquid desiccant air-conditioning (LDAC) and roof top unit (RTU) ; *source (Kozubal, et al, 2014) modified*

The results of a previous NREL system and energy saving analysis suggest that significant energy savings can be achieved by LDAC systems compared to conventional RTUs. The main reason for the energy savings is the avoidance of overcooling and the required reheat for thermal comfort. The emergence of high energy efficient buildings compounds the problems of overcooling, since latent load ratios increase, while sensible loads decrease. Mitigating problems associated with high latent loads are specifically critical for hot humid climates, such as Hawaii.

2.4. Assessing Combined Energy and IEQ Benefits of LDAC System

This section compares the anticipated energy performance of the type of LDAC system, to be used in this project, and a conventional direct expansion (DX) based roof top unit (RTU). The NREL cost comparisons, that were presented in Section 2.3, showing differences between conventional AC and the more advanced LDAC technology, only quantified energy savings. The cost comparison presented in this section goes beyond the energy advantages of LDAC to include tangible improvements of IEQ and IAQ. Such improvements of IEQ and IAQ are based on quantifiable financial benefits that were introduced in Section 1.2 of this report.

An example conditioned space (office) with a total floor area of 8,000 sqft is used for this assessment. The proposed LDAC system is adjusted to the required capacity for 8,000 sqft. A typical packaged RTU unit is used as the benchmark system of a conventional vapor compression HVAC system. The following assumptions were used:

- The assessment uses unit performances and the assessment is approximate and the results reflect anticipated performance conditions.
- An office space is used to allow the use of published estimated increases in revenues from employees working in office spaces with improved indoor environmental quality. The IEQ improvements considered are based on an increase in ventilation rates, increased indoor thermal comfort and avoidance of humidity related problems. The main technology application for the IEQ improvement is the decoupling of sensible cooling and dehumidification with a liquid desiccant system. A conservative 50% of increases in revenues is considered. With the full suggested productivity estimate of $6,500 per employee, the assumed 50% increases of revenues for work performed by office workers is considered as $3,250 per employee.
- The number of employees have been determined based on a “medium” employee density of 110 sqft per employee. The resulting number of employees in the 8,000 sqft office space is 70.

- The ventilation rate for the baseline conventional DX system is the proposed minimum ventilation rate defined in ASHRAE 62.1. The ventilation rate for the proposed LDAC system was chosen **as double** the rate of the baseline ventilation for the DX system.
- According to Section 2.1 of this report, the increased ventilation rate will increase the IEQ and the productivity of office workers.
- The benchmark conventional DX HVAC system uses recycled indoor air for sensible and latent heat removal with a typical ratio of return to outside air of 5.5 to 1.
- The proposed LDAC system employs a dedicated outdoor air supply (DOAS) system, which means that no return air and indoor recycling of air is used.
- Both systems would employ appropriate filters to remove particulate pollutants from the external air stream. The proposed LD system has the significant benefit of achieving high indoor air quality, because the desiccant solution serves as a disinfectant and eliminates virtually all particulate matter from the external ventilation air supply.
- The same internal and external sensible loads are used for the baseline and the proposed LDAC system. The sensible load reflects internal loads from people, lighting and plug loads and external skin loads. The values used are from the current ASHRAE 90.1 and the IECC energy codes. With an energy efficient building envelope, the external heat gain is minimized.
- The same internal latent load has been used for both the baseline and the proposed LDAC system. The internal latent includes latent heat from people as well as penetrations and infiltrations. The example calculation assumed a sufficiently tight envelope.
- The external latent load was calculated from the difference between outside and target indoor humidity levels, and the air mass flow rate.
- The performance evaluation did not consider heat energy recuperation techniques, since both systems would benefit equally from such energy recovery measures.
- The cost assessment uses only predicted energy source costs and not first costs. The reason is that the LDAC technology is in early stages of commercialization, with cost representing small production numbers, whereas the conventional DX HVAC technology is a mature technology where economy of scale favors low equipment costs.

Figure 2.4.1 shows the basic process diagram of the conventional DX system used as the baseline HVAC unit. A roof top unit (RTU) with a conventional DX vapor compression refrigeration cycle provides all the sensible and latent heat removal as well as the supply of fresh outside air flow rate of 850 cfm, which is the minimum ASHRAE recommended outdoor air supply rate. The RTU units uses only electric energy for sensible cooling and latent heat removal.

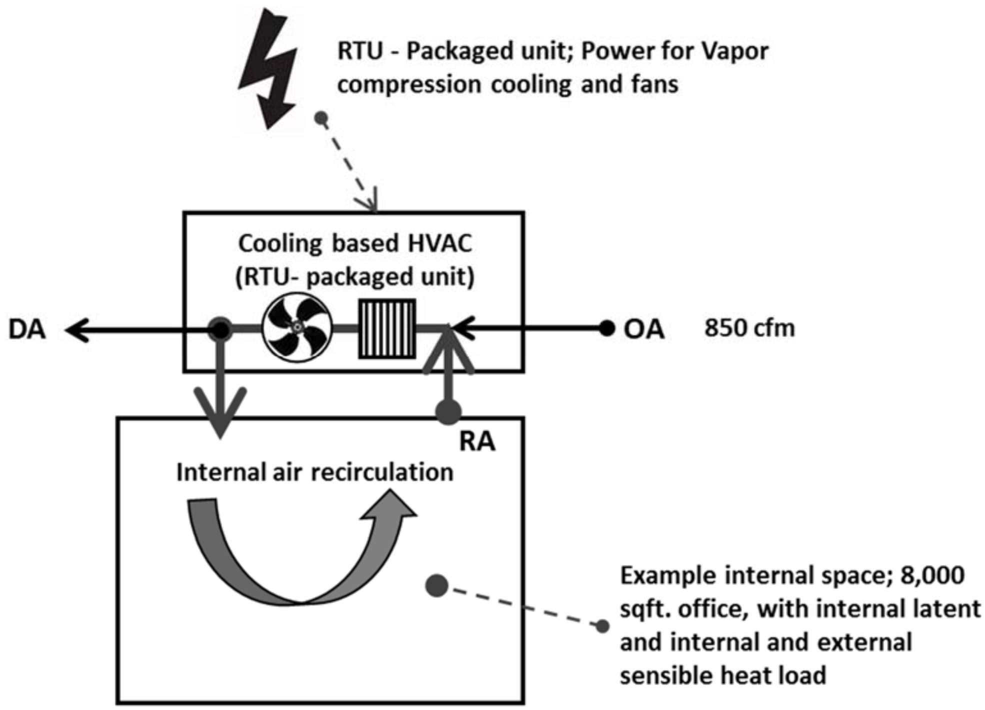


Figure 2.4.1: Conventional DX unit used as the baseline HVAC system for the sizing example; OA=Outside air, RA=Return air; DA = Discharge air

Figure 2.4.2 shows the basic process diagram of the proposed LD system. The system has a DOAS system with 1,700 cfm of fresh outside air supply, which is double the ventilation air supply if the baseline DX RTU. As the outside air flows through the desiccant conditioner (absorber), humidity in the air is removed by means of the liquid desiccant solution which is brought into direct contact with the supply air. While the supply air passes through the desiccant absorber, particulate air pollutants are removed by disinfection property of the liquid desiccant solution. The absorption heat that is generated in the desiccant conditioner is rejected through an evaporative cooling tower. The heat required to regenerate the liquid desiccant is provided mostly through solar heat. Heat from a synthetic natural gas (or propane gas) boiler provides the required heat during periods when solar heat is not sufficient. Concentrated liquid desiccant solution is stored to bridge periods when no desiccant regeneration is occurring.

Upstream of the conditioned space, the dehumidified air flows through a heat exchanger which rejects sensible heat and cools the supply air to the desired supply temperature. The remaining sensible heat is rejected by means of passive chilled beams. The cooling capacity for the sensible heat rejection is provided by a conventional vapor compression chiller. This chiller operates at a higher COP because of the higher temperature chilled water supply temperatures and water cooling of the condenser using the evaporative cooling tower.

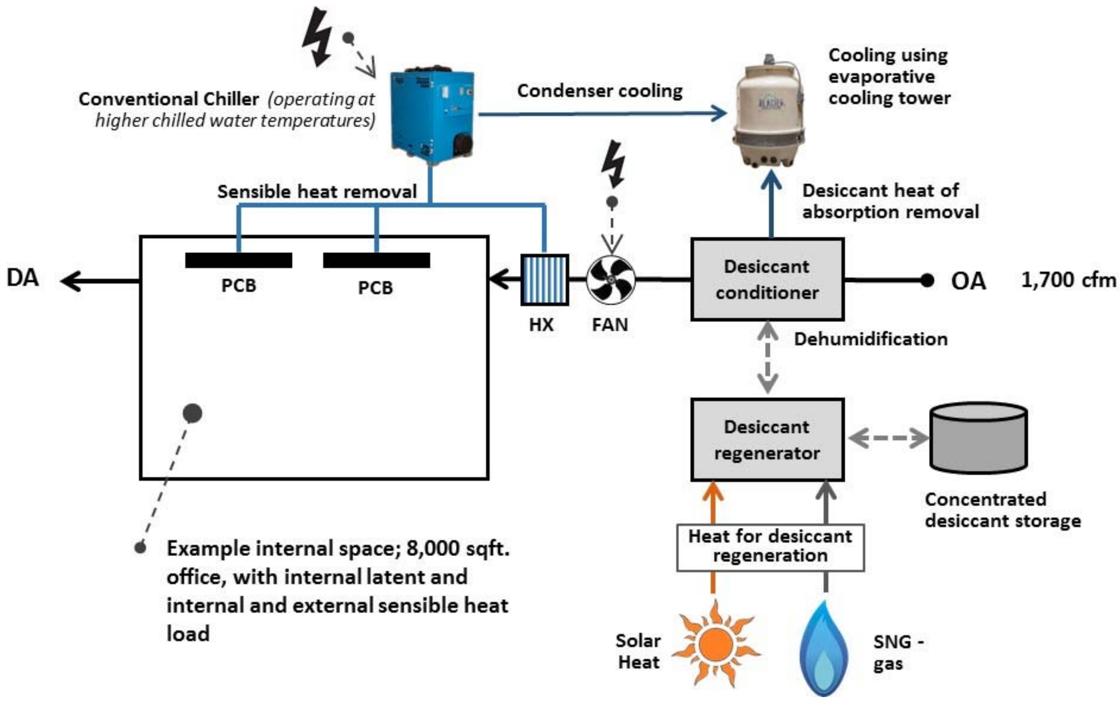


Figure 2.4.2: The proposed liquid desiccant (LD) HVAC system used for the sizing example; OA=Outside air, DA = Discharge air, HX=heat exchanger, PCB=Passive chilled beam, SNG=Synthetic natural gas

Table 2.4.1 depicts the results of this example energy savings and increased revenue analysis. Figure 2.4.3 illustrates a comparison of the <u>potential energy savings</u> of the LDAC system compared to a conventional DX HVAC system. Figure 2.4.4 compares the <u>potential revenue gains of office operation based on the increased IEQ</u> and resulting productivity. The results suggest that: (1) the proposed LDAC system with DOAS and passive chilled beams is more energy efficient than the conventional all air DX HVAC system; (2) that the projected revenue increases from higher productivity inside the conditioned office based on improved IEQ clearly outweigh the energy savings that are projected.

The energy performance properties of the baseline conventional RTU unit and the proposed LD system are shown in Tables 2.4.2 and 2.4.3.

Figures 2.4.3 and 2.4.4 show the psychrometric conditions of the baseline conventional DX system and the proposed AILR LDAC system. The figures depict the differences in dehumidification processes as the conventional DX HVAC system requires the air to flow over below-dewpoint cooling surfaces, while the proposed LDAC system can achieve the desired dehumidification levels without cooling down the supply air.

Description	units	Conventional DX RTU	Proposed LDAC
Example office space:			
Floor area	sqft	8,000	8,000
Number of employees considered		70	70
Ventilation air (total)	cfm	850	1,700
Energy costs			
Electricity costs	$/a	$30,400 (1)	$16,400
SNG costs	$/a	N/A	$4,200
sum energy costs	$/a	$30,400	$20,600
Energy costs savings	$/a	0	**$9,800**
Considerd increased revenues due to double ventilation rates and improvements of Indoor Air Quality (2)	$/a per employees	N/A	$3,250
Number of employees considered			70
sum of increased revenue	$/a		**$227,500**
Improved performance of LDAC unit	$/a		$237,000
energy savings	% of total	$9,800	4%
Increased revenue	% of total	$227,500	96%

Note (1) : Does NOT contain energy costs for reheat of chilled air in conventional DX HVAC
Note (2) : Higher revenues are assumed with improved IEQ (see Section 2.1)

Table 2.4.1: Compariosn of annual energy costs and increased revenue potential of proposed LDAC system

The table summarizes the energy performance and the reuslting energy savings of the proposed LDAC system as compared to the baseline conventinal DX HVAC system.
The increase revenue analysis considers 50% of the potential $6,500 increased revenue per person with higher IEQ.

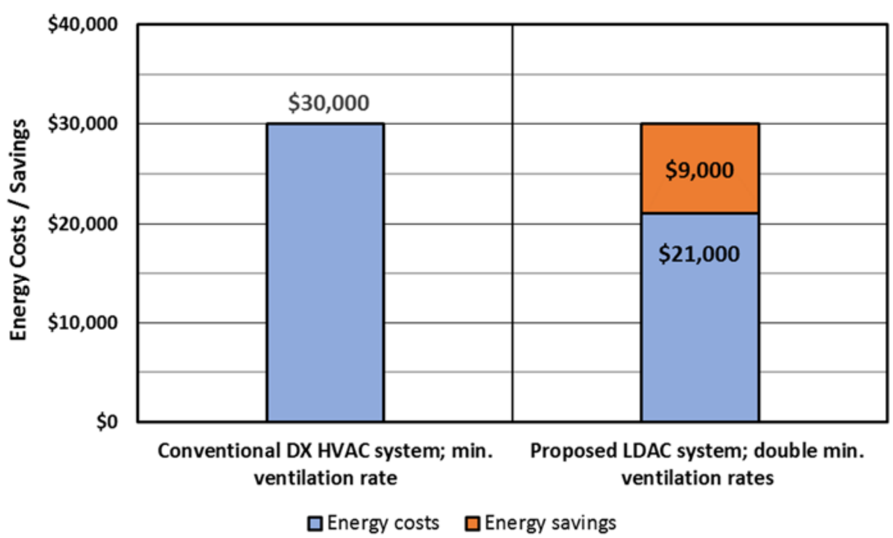


Figure 2.4.3: Comparison of energy perfroamnce

This figure suggests that energy costs savings of about 40% can be anticipated under the assumed conditions.

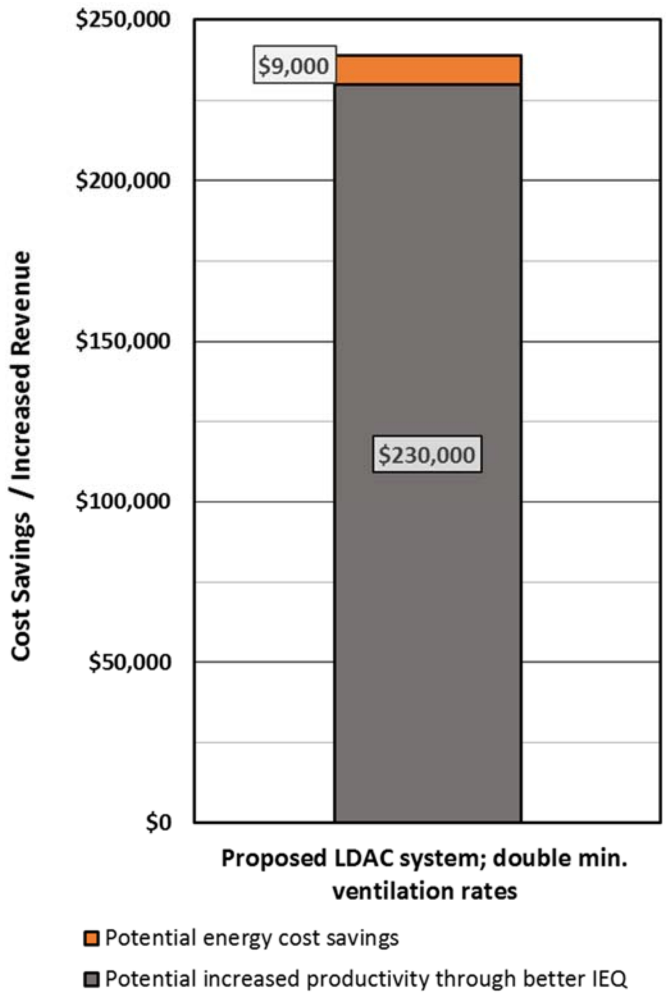
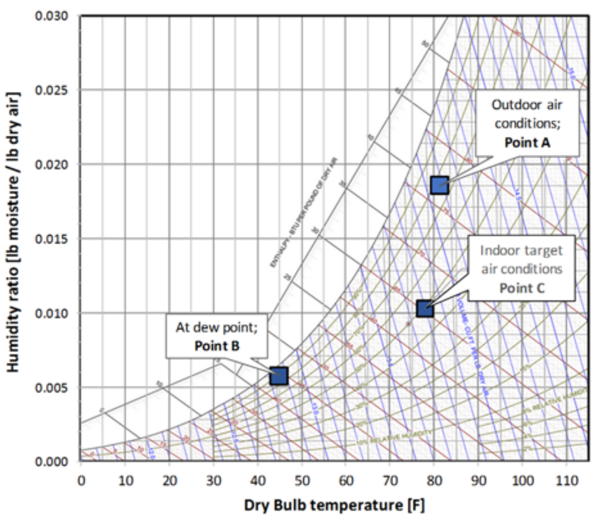


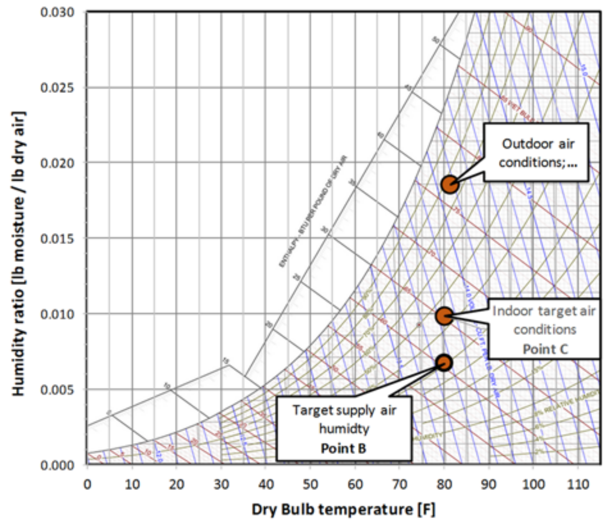
Figure 2.4.4: Comparison of energy savings and increased revenues based on productivity gains through higher IEQ

The image illustrates how high potential revenue gains are compared to the projected energy savings. The potential revenue gains are through increased productivity which is in turn a result of better indoor IEQ.



		F	w	RH
A	Outdoor air conditions	81	0.0186	81%
B	At dew point	45	0.0057	100%
C	Indoor target air conditions	78	0.0103	50%

Figure 2.4.3: Psychrometric conditions of the baseline conventional DX system



		F	w	RH
A	Outdoor air conditions	81	0.0186	81%
B	Target supply air humidty	80	0.0067	100%
C	Indoor target air conditions	80	0.0099	45%

Figure 2.4.4: Psychrometric conditions of the proposed AILR LDAC system.

The example system performance assessment and financial analysis of the proposed LDAC system suggests significant energy savings and, even more importantly, a tangible increase in productivity of office workers in the example office space. A sizable increase in productivity is reported when ventilation is improved, as with the proposed LDAC system compared to the baseline conventional DX HVAC. *The combined energy savings and revenue increases make the proposed LDAC system significantly more cost effective than conventional DX HVAC system.*

Note: The projected energy savings of the proposed LDAC will be even greater when a more efficient 2-stage desiccant regenerator is available in the future. The predicted energy savings in the example office space are consistent with energy savings reported by NREL, see Section 2.3.

Conventional DX RTU

Description	units	Value
Ventilation air	cfm	850
	lb da / h	3,600
Outside air	F	81
	RH	81%
Internal air	F	78
	RH	50%
Dew point	F	45
Heat loads removed		
Sensible loads	BTU/h	165,000
Dehumidification	BTU/h	81,000
sum of heat loads removed	BTU/h	246,000
	tons	20.5
Energy performance:		
Unit energy consumption	kW/ton	1.65
Design load	kW	33.8
Operational envelope	h per day	8
	day / year	313
	hours per year	2,504
	kWh / year	84,000
Unit electrcity costs	per kWh	$0.36
Approximate annual energy costs:		
Electrcity costs	per year	$30,400

Table 2.4.2: Energy performance properties of the baseline conventional roof top system

Proposed LDAC System

Description	units	Value
Ventilation air	cfm	1,700
	lb / h	7,300
Outside air	F	81
	RH	81%
Internal air	F	80
	RH	45%
Internal Dew point	F	57
Heat loads removed / added		
Sensible loads (removed)	BTU/h	165,000
Dehumidification (removed)	BTU/h	84,000
Desiccant regeneration (added)	BTU/h	159,000
Design energy performance:		
Electrcity chiller	kW	13.9
Electrcity fans	kW	2.2
Electrcity pumps	kW	1.2
Electrcity LDAC unit	kW	0.8
Sum of design electrcity load	kW	18.2
SNG	Therms / h	0.56
Operational envelope	h per day	8
	day / year	313
	hours per year	2,504
	kWh / year	45,500
	Therms per year	1,400
Unit electrcity costs	per kWh	$0.36
Unit SNG costs	per therm	$3.00
Approximate annual energy costs:		
Electric energy	$ / a	$16,400
SNG	$ / a	$4,200

Table 2.4.3: Energy performance properties of the proposed LD system

2.5 Comparison of Carbon Footprint

The environmental impact in terms of carbon emissions of the proposed LDAC unit with vapor compression (VP) chiller for sensible heat removal was compared to the baseline conventional all-electric HVAC system. In addition, a LDAC system with a thermally driven chiller was compared against the baseline conventional all-electric AC system.

Table 2.5.1 describes the three HVAC systems used in the carbon footprint comparison.

Table 2.5.1: Three HVAC systems used in the carbon footprint comparison

No.	Description of HVAC system	Sensible heat sink	Latent heat sink
1	LDAC with electrically driven VC chiller (**proposed LDAC system**)	Vapor compression chiller	Liquid desiccant (LD) dehumidification
2	Conventional (**baseline**) HVAC (all electric energy)	Vapor compression chiller	Vapor compression chiller
3	LDAC with thermally driven Adsorption chiller (most efficient system)	Thermally driven adsorption chiller	Liquide desiccant (LD) dehumidification

The following calculation procedure was used to determine the "carbon footprint":

- The site energy rate (electricity and gas) was determined for all three systems.
- The source energy was calculated using source energy conversion factors by Energy Star Portfolio Manager
- Greenhouse Gas Equivalencies published by the EPA for the source energy were used to determine the carbon related emissions.

The results of the carbon equivalency assessment are shown in Table 2.5.2. A percentage comparison with the baseline carbon equivalent emission of the conventional HVAC system is presented in Figure 2.5.1.

Table 2.5.2: Results of carbon footprint comparison between three HVAC systems 1 through 3

Type of system		Site energy		Source energy *Note (4) and (3)*				Carbon Dioxide Equivalent Emissions *Note (5)*	
		Electrcity	Gas	Electrcity	Gas	Electrcity	Gas		
No.	Description	kW	Therms/h	kW	Therms/h	kWh/a	Therms/a	pounds (equiv.)	%
1	LDAC with electrically driven VC chiller **(proposed LDAC system)** ***Note: (1)***	18.2	0.56	57	0.56	143,000	1,400	251,000	58%
2	Conventional (**baseline**) AC (all electric energy)	33.7	N/A	106	N/A	265,000	N/A	435,000	100%
3	LDAC with thermally driven Adsorption chiller **(most efficient system)** ***Note: (2)***	4.2	0.73	13	0.73	33,000	1,800	76,000	17%

Assumptions:

Note (1): thermal source 30% gas an 70% solar
Note (2): thermal source 30% gas an 70% solar
Note (3): 2,500 operating hours per year
Note (4): Source energy conversion factors from Energy Star Portfolio Manager
Factor for electrcity = 3.14; Factor for gas = 1.01
Note (5): Greenhouse Gases Equivalencies factors from EPA

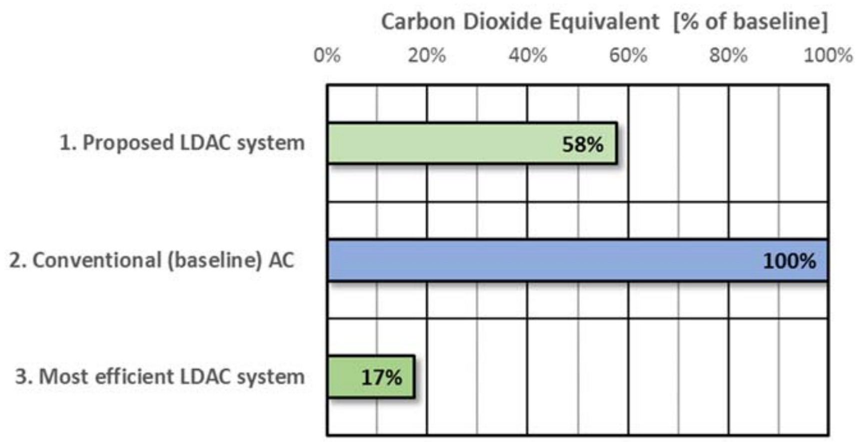


Figure 2.5.1: percentage comparison with the baseline carbon equivalent emission of the conventional HVAC system

A comparison of the carbon equivalence emissions of the proposed LDAC with the baseline conventional all-electric HVAC systems indicates that the proposed LDAC system has 42% lower carbon emissions than the baseline conventional HVAC system. If the electrically driven vapor compression chiller of the proposed LDAC system is replaced with a thermal driven adsorption chiller the carbon equivalent emissions fall to below 20% of the baseline value. In this estimate, it was assumed that for 30% of operating time, the thermally driven desiccant dehumidification and chiller the heat source would be provided by gas and not by solar.

This high reduction of electric energy and resulting carbon emission for a HVAC system which at the same time provides outstanding IEQ conditions would be a significant contribution to Hawaii's future energy and carbon reduction goals.

SECTION 3 - TESTING OF THE LDAC SYSTEM IN PROJECT PHASE II

The proposed LDAC system will be tested in Project Phase II, which has two parts:

Project Phase II/A:	Initial testing of the AILR LDAC system in a lab controlled space; "LAB 123" of the UHM Marine Building
Project Phase II/B:	Field test installation of the AILR LDAC system at a pilot test site (TBD)

The AILR LDAC system capacity has been selected to provide dehumidification for a range of possible pilot installations. The capacity is large enough to serve as a meaningful test platform for applications in a medium size offices or educational institutions. On the other hand, the size of the AILR LDAC system is kept small to avoid high investment for this first installation of this technology in Hawaii. Figure 3.1 delineates the main objectives of the Project Phases II/ A and B.

Project Phase II/A	Project Phase II/B
Shake down testing of the AILR LDAC system in LAB 123 of the UHM marine center	**Field test installation** of the AILR LDAC system at a pilot test site (TBD)
Main objectives of Project Phase II/A	**Main objectives of Project Phase II/B**
Gain operational process experience with the LDAC system; **Provide system design parameters** for the supporting infrastructure for Phase II/B; **Showcase the basic benefits of LDAC** technology and system integration to stakeholders for the Phase II/B field test installation	**Validate the energy performance benefits** of the LDAc system in real world applications; **Familiarize Hawaii HVAC industry** with the new HVAC technology of decoupling sensible and latent cooling load removal; **Validate the advantages of the improved HVAC technology** to provide improved IEQ.
Projected schedule of Project Phase II/A	**Projected schedule of Project Phase II/A**
Start: March 2018 Completion: December 2018	Start: March 2019 Completion: March 2020

Figure 3.1: Overview of Project Phase II

SECTION 4 - TESTS OF PHASE II/A INITIAL “SHAKE DOWN” TESTS IN UHM MARINE CENTER

This section presents the Preliminary Design of the system installation for the initial system test in the lab environment of the UHM Marine Center at Pier 35.

4.1 Main Objectives of Project Phase II/A – Testing in Lab Environment

The main objectives of Project Phase II/A are as follows:

<u>Gain operational process experience with the LDAC system</u>: During the test series in a lab controlled environment, the LD system and the ancillary thermal systems will be operated under a series of process settings. The LDAC systems parts will be tested regarding uninterrupted operation during longer periods with constant loads and intakes as well as during shorter periods of transient processes with changes in internal loading.

<u>Provide system design parameters for the supporting infrastructure for Phase II/B:</u> The design of the pilot installation under Project Phase II/B will be based on design parameters which are in turn verified by operational experiences gained in the initial test series during Project Phase II/A. The AILR LDAC technology is new to the HNEI research team. Adding chilled beam cooling technology and the cooling effect of ceiling fans to the space conditioning system adds complexity. The success of the overall system of LDAC technology and supporting energy saving sensible cooling technologies depends on a successful integration of the decoupled sensible and latent heat removal system. The completed initial tests in Phase II/A will lay the foundation for a suitable design and successful system integration of the cooling systems to be deployed in the demonstration in Phase II/B.

<u>Showcase the basic benefits of LDAC technology and system integration to stakeholders for the Phase II/B field test installation:</u> The most preferable location for the pilot installation under Project Phase II/B will be a test site that has suitable physical characteristic as well as the support of stakeholders of the pilot installation. The initial test under Project Phase II/A will give stakeholders opportunities to obtain first hand experiences of a working test setup of the LDAC technology that will be used of the pilot Installation.

4.2 Test facility LAB 123 in the UHM Marine Center at Pier 35

The lab space for the initial tests is an 800 sqft laboratory space, referred to as “LAB 123”, is in the University of Hawaii Marine Center (see Figure 4.2.1).

Figures 4.2.2 through 4.2.6 show images of the present conditions of the space

Figure 4.2.1: University of Hawaii Marine Center at Pier 35

Source: www.soest.hawaii.edu

View from the North

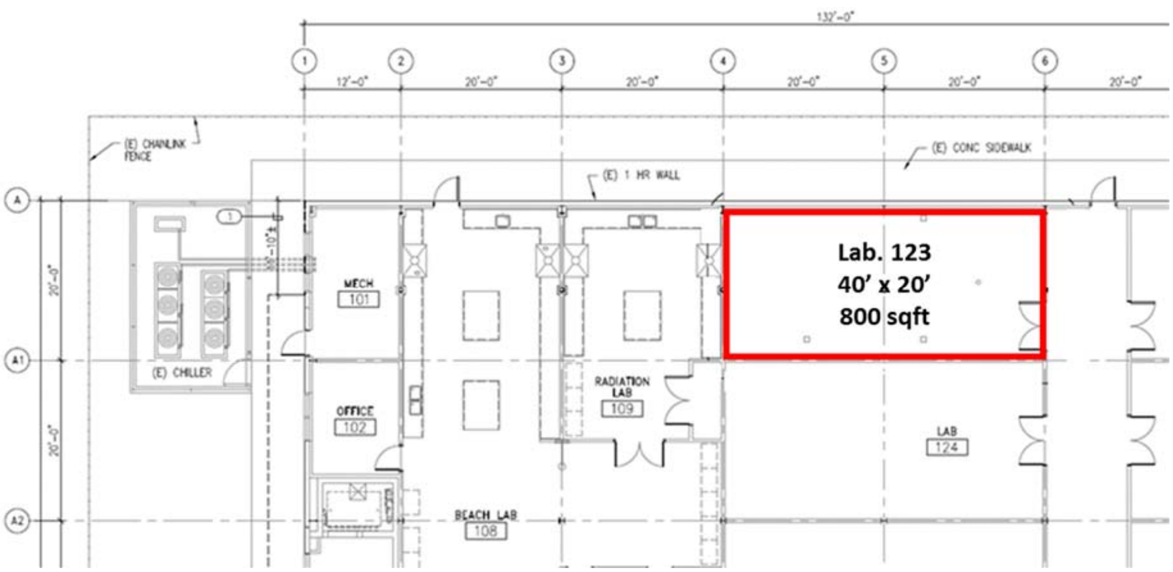


Figure 4.2.2: Location of test space “LAB 123” on the first floor of UHM Marine Center (source: UHM Marine Center, modified)

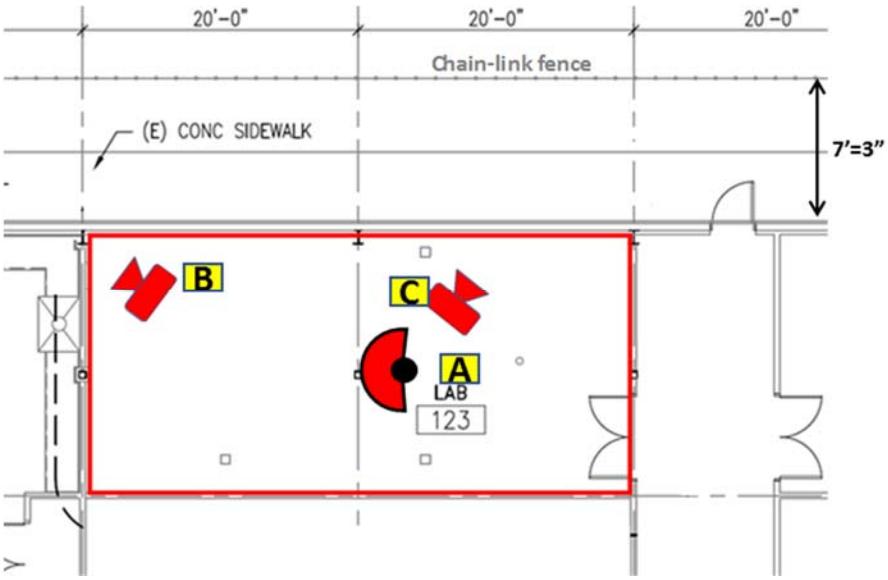


Figure 4.2.3: Definition of interior photos inside space LAB 123

(source: UHM Marine Center, modified)

Figure 4.2.4: View A

The photo shows the present configuration of the space "LAB 123"

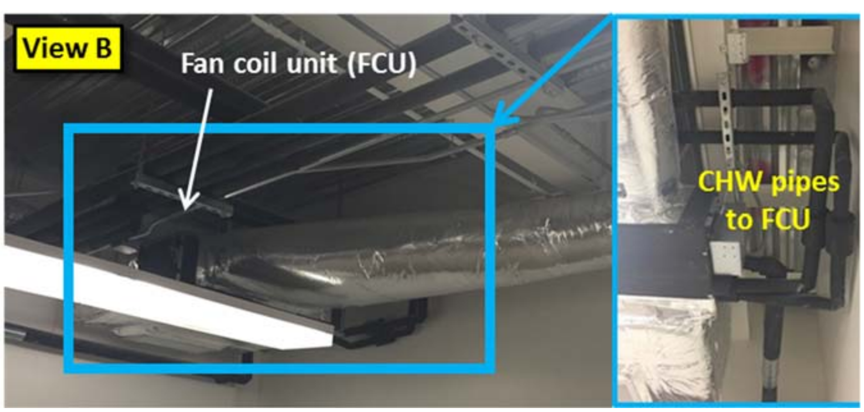


Figure 4.2.5: View B

The photo shows the existing FCU which provides sensible cooling to LAB 123. The insert shows the existing chilled water pipes that serve the FCU. For the tests, chilled water will be taken off the existing chilled water pipes

Figure 4.2.6: View C

The photo shows the existing duct work which provides chilled air to the space

4.3 Basic Energy and Mass Flow Diagram and Analysis

Figure 4.3.1 shows the basic energy and mass flow diagram. Table 4.3.1 provides the energy and mass flow properties at the locations indicated in Figure 4.3.1.

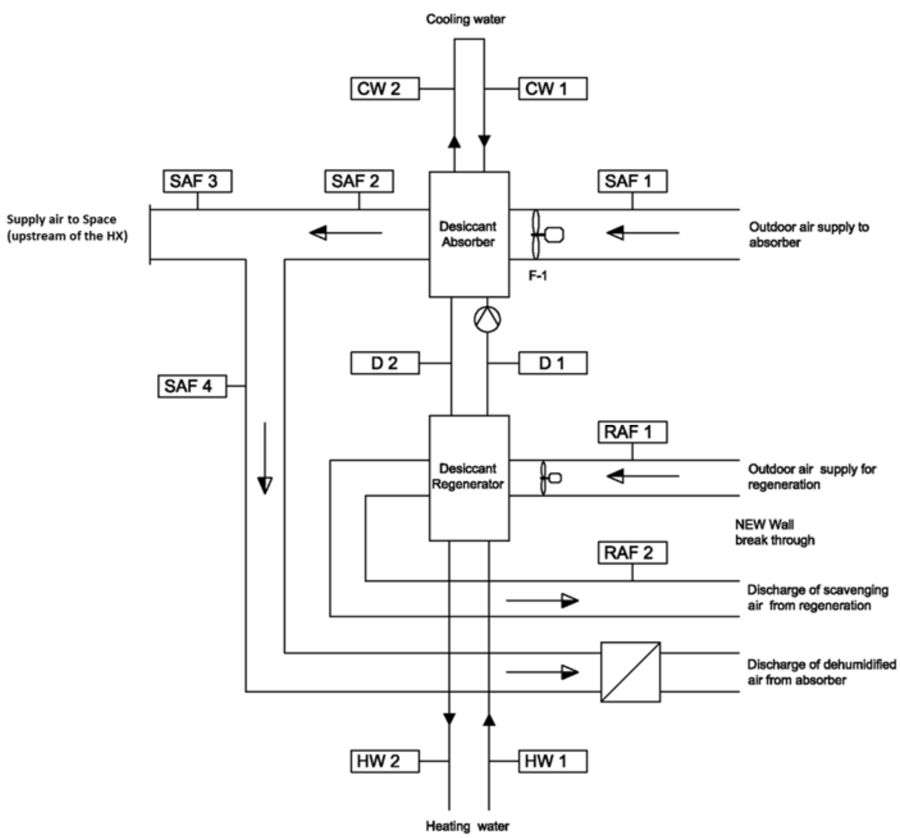


Figure 4.3.1: Basic Energy and Mass Flow Diagram

(see Table 4.3.1 for the referenced parameters of energy and mass)

Table 4.3.1: Data for the Basic Energy and Mass Flow Diagram

Flow ID	Description	Parameter	Unit	quantity
CW 1	Cooling water supply to absorber	Temp.	F	71.4
		Spec. heat	Btu/(F*lbs.)	1.00
		Flow	gpm	7.813
CW 2	Cooling water return from absorber	Temp.	F	82.2
		Spec. heat	Btu/(F*lbs.)	1.00
		Flow	gpm	7.813
SAF 1	Outside air supply for absorber	Temp.	F	81
		Hum. Ratio	lb/lb	0.02
		Spec. heat	Btu/(F*lbs.)	0.25
		Enthalpy	Btu/lbs.	39.42
		Flow	scfm	800
SAF 2	Dehumidified air; absorber downstream	Temp.	F	89.8
		Hum. Ratio	lb/lb	0.01
		Dew Point	F	44.96
		Spec. heat	Btu/(F*lbs.)	0.25
		Enthalpy	Btu/lbs.	28.47
		Flow	scfm	800
SAF 3	Dehumidified air; supply air to space	Temp.	F	89.8
		Hum. Ratio	lb/lb	0.01
		Spec. heat	Btu/(F*lbs.)	0.25
		Enthalpy	Btu/lbs.	28.47
		Flow	scfm	**varies**
SAF 4	Dehumidified air; return to outside	Temp.	F	89.8
		Hum. Ratio	lb/lb	0.01
		Spec. heat	Btu/(F*lbs.)	0.25
		Enthalpy	Btu/lbs.	28.47
		Flow	scfm	**varies**
RAF 1	Outside air supply for regenerator	Temp.	F	81
		Hum. Ratio	lb/lb	0.02
		Spec. heat	Btu/(F*lbs.)	0.25
		Enthalpy	Btu/lbs.	39.42
		Flow	scfm	300
RAF 2	Return air from regenerator	Temp.	F	146.4
		Spec. heat	Btu/(F*lbs.)	0.25
		Enthalpy	Btu/lbs.	91.56
		Flow	scfm	300
HW 1	Heating water supply to regenerator	Temp.	F	176.5
		Spec. heat	Btu/(F*lbs.)	1.00
		Flow	gpm	24.95
HW 2	Heating water return from regenerator	Temp.	F	170.5
		Spec. heat	Btu/(F*lbs.)	1.00
		Flow	gpm	24.95
D 1	Desiccant supply flow to absorber	Temp.	F	105.8
		Spec. heat	Btu/(F*lbs.)	0.61
		Flow	gpm	0.65
D 2	Desiccant return flow from absorber	Temp.	F	100.8
		Spec. heat	Btu/(F*lbs.)	0.61
		Flow	gpm	0.65
D 3	Desiccant supply flow to regenerator (inlet to IHX)	Temp.	F	100.8
		Spec. heat	Btu/(F*lbs.)	0.64
		Flow	gpm	0.74

4.4 Process and Instrumentation Diagram

Figure 4.4.1 shows the process and instrumentation diagram (P&ID) of the proposed LDAC test set-up in the space LAB 123. The full-size sheet is presented in Appendix A - Sheet D-001

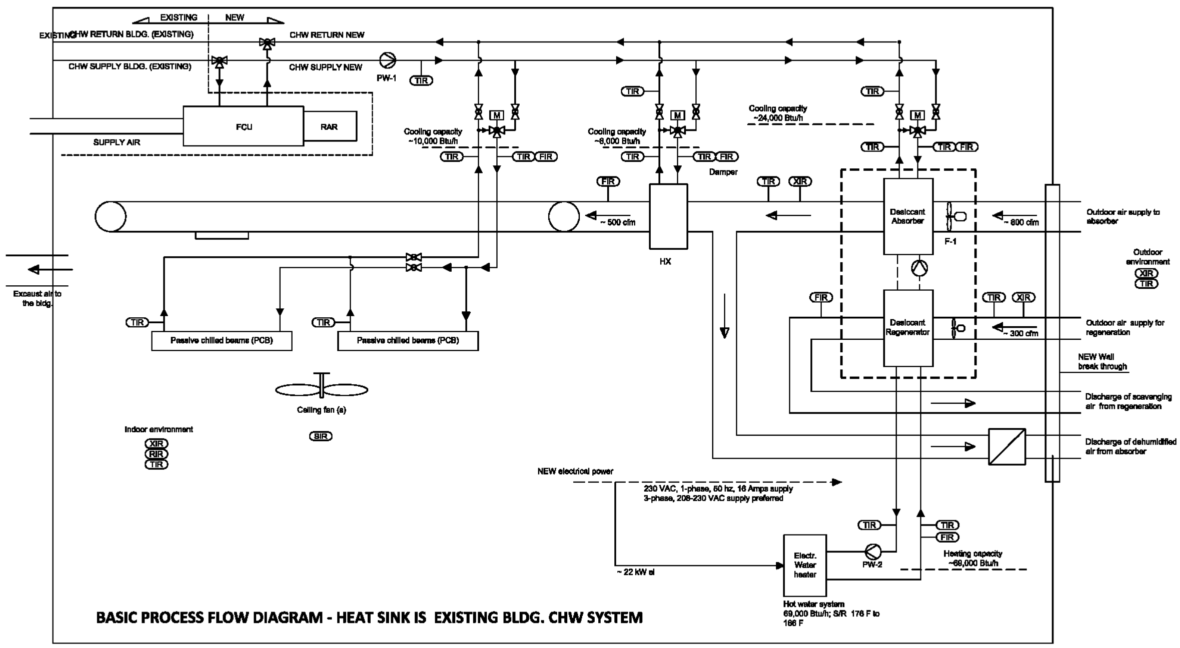


Figure 4.4.1: Process and instrumentation diagram of the proposed LDAC test set-up in the space LAB 123. The full-size sheet is presented in Appendix A - Sheet D-001

4.5 Main System Components and Instrumentation

Table 4.5.1 describes the main components of the test set-up. The identifiers of the main component referred to in Table 4.5.1 are explained in Figure 4.5.1.

Table 4.5.2 shows the generic instrumentation to be used as defined in the P&I D, Figure 4.5.1.

Table 4.5.3 describes the pipe and duct runs on the test set-up. The identifiers of the pipe and ducts referred to in Table 4.5.3 are explained in Figure 4.5.2.

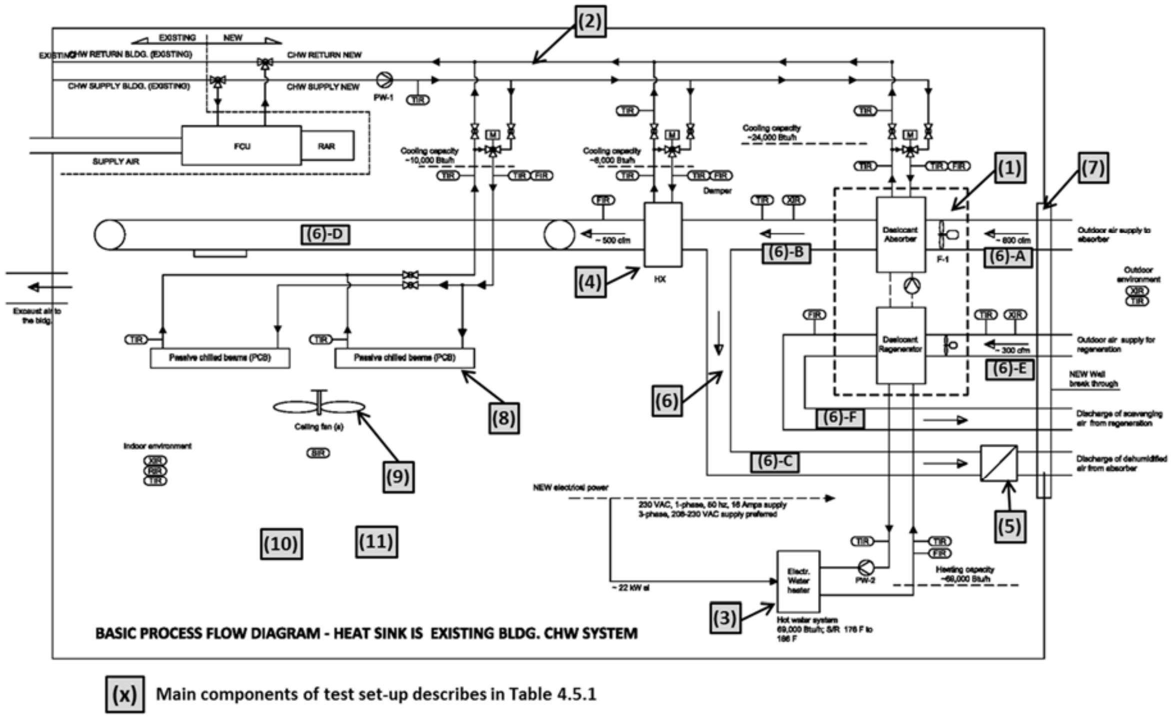


Figure 4.5.1: Identification of the main component of the cooling system in LAB 123, as referred to in Table 4.5.1.

Table 4.5.1: Main component of the cooling system in LAB 123, as identified in Figure 4.5.1

Id IN P&ID (*)	Type pf equipment	Description and working function
(1)	LDAC unit	The packaged LDAC unit by AILR includes all process equipment for liquid desiccant dehumidification within one self-contained housing. This process equipment includes: • Desiccant conditioner (absorber unit) • Desiccant regenerator (desorber unit) • Air fans • Pumps for desiccant • Heat exchanger for desiccant • Instrumentation for system control

Table 4.5.1: Main component of the cooling system in LAB 123, as identified in Figure 4.5.1

Id IN P&ID (*)	Type pf equipment	Description and working function
(2)	Chilled water system	The new chilled water supply system is connected via a new 3-way valve to the existing UHM marine Center building chilled water system. The new chilled water supply system provides chilled water to the following three main components: (A) The LDAC unit for desiccant conditioner (internal) cooling (B) The air-water heat exchanger (HX) (C) The passive chilled beams. Mixing valves with individual temperature regulation will be installed upstream of the components (B) and (C)
(3)	Electric water heater	The electric water heater provides heat to the desiccant regenerator. In the final pilot installation, the process heat for regeneration will be provided by solar and/or gas boiler. The selection of electricity for the heat source is made of convenience and necessity of simplification.
(4)	Air-water heat exchanger (HX)	The air-water heat exchanger (HX) cools the air supply to the space and is located downstream of the LDAC unit. The HX allows the individual control of the supply air to the spaces, independently of the air temperature of the air downstream of the individual sensible cooing of the dried supply air. The HX is installed s=downstream of the flexible tube junction of the desiccant conditioner bypass.
(5)	Damper	The damper provides pressure losses with which the bypass air volume is regulated. The bypass is manually operated.

Table 4.5.1: Main component of the cooling system in LAB 123, as identified in Figure 4.5.1

Id IN P&ID (*)	Type pf equipment	Description and working function
(6)	Flexible air ducts (typical)	There are several flexible air ducts installed which connect the LDAC unit with the exterior air and the internal cooled air Distribution. The following air conduits have flexible air hoses: (6)-A Outside air supply to the desiccant conditioner (6)-B Dehumidified air from the desiccant conditioner (6)-C Bypass - dehumidified air from the desiccant conditioner (6)-D Dehumidified air supply to space (6)-E Savaging air supply to the desiccant regenerator (6)-F Savaging air discharge from the desiccant regenerator
(7)	Wall breakthrough cover plate	The exterior wall must be broken through to install a cover plate which provides connection of internal air supply and discharge hoses to the exterior air.
(8)	Passive chilled beams (PCB)	The passive chilled beams will provide sensible cooling capacity through combined convection and radiation. There will be multiple RCBs installed. They are supplied with chilled water through the existing building chilled water supply. A mixing valve (manually operated) will provide the correct supply temperature for the PCBs.
(9)	Ceiling fan	A ceiling fan of sufficient size and air flow capacity will be installed in LAB 123 to provide additional sensible cooling capacity. The ceiling fan will be installed either below or above the PCBs.
(10)	Humidifier	The humidifier will be used to provide latent load to the room. The space LAB 123 itself does not have any sizable latent loads, therefore latent load must be added to achieve realistic humidity levels. The humidifier will be adjustable to provide different amounts of humidification.

Table 4.5.1: Main component of the cooling system in LAB 123, as identified in Figure 4.5.1

Id IN P&ID (*)	Type pf equipment	Description and working function
(11)	Heater (electric)	An electric heater will be used to provide quantities of sensible heat load, since LAB 123 does not have any sizable sensible loads.

Table 4.5.2: Instrumentation List for test set-up depicted in Figure 4.5.1

Main property: *T = Temperature; F = Flow; X = Humidity; S= Air speed*

Type of data acquisition: *R = Recording; I = Indicating*

ID	Medium	Location	Type	Function
TR-A1	Room air in space LAB123	Inside room LAB123	Air temperature measurement inside the room LAB123	The room temperature is measured. One measurement will be used as representative for the entire space LAB123. The room temperature will be used to determine the heat balance of the space LAB123 and to determine the expected PMV (thermal comfort)
TR-A2	Radiant temperatures inside the space LAB123	Inside room LAB123	MRT measurement inside the room LAB123	The MRT is measured. One measurement will be used as representative for the entire space LAB123 to determine the to determine the expected PMV (thermal comfort)) near the PCBs and the ceiling fan.
XR-A	Room air in space LAB123	Inside room LAB123	Humidity measurement inside air duct	The MRT is measured. One measurement will be used as representative for the entire space LAB123 to determine the to determine the humidity balance of room LAB123.
TR-B	Outside air	Outside the building	Air temperature measurement	The outside air temperature is measured. The outside temperature is used as a baseline air temperature

Table 4.5.2: Instrumentation List for test set-up depicted in Figure 4.5.1

Main property: *T = Temperature; F = Flow; X = Humidity; S= Air speed*

Type of data acquisition: *R = Recording; I = Indicating*

ID	Medium	Location	Type	Function
XR-B	Outside air	Outside the building	Humidity measurement	The outside air temperature is measured. The outside humidity is used as a baseline air temperature
TR-1	Chilled water (CHW)	CHW supply pipe to the test set-up	Temperature measurement inside water pipe	The temperature of the CHW supply is measured. The same CHW supply temperature is assumed for the entire test set-up
TR-2	Chilled water (CHW)	CHW return from the passive chilled beams (PCB)	Temperature measurement inside water pipe	The temperature of the CHW return from the PCB is measured. The temperature is used to determine the heat balance for the PCB array.
TR-3	Chilled water (CHW)	CHW supply to the PCB array downstream of the mixing valve	Temperature measurement inside water pipe	The temperature of the CHW supply to the PCB array is measured. The temperature is used to regulate the mixing valve. The temperature is used to determine the heat balance for the PCB array and the individual PCB unit.
TR-4	Chilled water (CHW)	CHW return from the water-to-air heat exchanger (HX) inside the 12-inch internal air supply duct	Temperature measurement inside water pipe	The temperature of the CHW return from the HX is measured. The temperature is used to determine the heat balance for the HX array.
TR-5	Chilled water (CHW)	CHW supply from the HX	Temperature measurement inside water pipe	The temperature of the CHW supply to the HX is measured. The temperature is used to regulate the mixing valve. The temperature is used to determine the heat balance for the HX.

Table 4.5.2: Instrumentation List for test set-up depicted in Figure 4.5.1

Main property: *T = Temperature; F = Flow; X = Humidity; S= Air speed*

Type of data acquisition: *R = Recording; I = Indicating*

ID	Medium	Location	Type	Function
TR-6	Chilled water (CHW)	CHW return from the LDAC unit	Temperature measurement inside water pipe	The temperature of the CHW return from the LDAC unit is measured. The temperature is used to determine the heat balance for the internally cooled desiccant conditioner.
TR-7	Chilled water (CHW)	CHW return from the LDAC unit	Temperature measurement inside water pipe	The temperature of the CHW supply to the LDAC unit is measured. The temperature is used to regulate the mixing valve. The temperature is used to determine the heat balance for the internally cooled desiccant conditioner.
TR-8 through TR-11	Chilled water (CHW)	CHW return from individual PCB units	Temperature measurement inside water pipe	The temperature of the CHW return from the individual PCB unit is measured. The temperature is used to determine the heat balance for each individual PCB unit.
TR-12	Heating water (HW)	Heating water return from the desiccant regenerator	Temperature measurement inside water pipe	The temperature of the heating water return from the desiccant regenerator is measured. The temperature is used to determine the heat balance for the internally heated desiccant regenerator.
TR-12	Heating water (HW)	Heating water supply to the desiccant regenerator	Temperature measurement inside water pipe	The temperature of the heating water supply to the internally heated desiccant regenerator is measured. The temperature is used to determine the heat balance for the internally heated desiccant regenerator.
FR-1	Chilled water (CHW)	CHW supply to the first PCB unit	Flow measurement inside water pipe	The flow rate of the chilled water supply will be measured to calculate the heat balance of the first individual chilled beam unit.

Table 4.5.2: Instrumentation List for test set-up depicted in Figure 4.5.1

Main property: *T = Temperature; F = Flow; X = Humidity; S= Air speed*

Type of data acquisition: *R = Recording; I = Indicating*

ID	Medium	Location	Type	Function
FR-2-4	Chilled water (CHW)	CHW supply to the first PCB units 2 through 4	Flow measurement inside water pipe	The flow rate of the chilled water supply to the individual PCB units will be measured to calculate the heat balance of the individual chilled beam units 2 through 4.
FR-5	Chilled water (CHW)	CHW supply to the first PCB unit	Flow measurement inside water pipe	The flow rate of the chilled water supply to the HX will be measured to calculate the heat balance.
FR-6	Chilled water (CHW)	CHW supply to the first PCB unit	Flow measurement inside water pipe	The flow rate of the chilled water supply to the internally cooled desiccant conditioner will be measured to calculate the heat balance.
FR-6	Heating water (HW)	HW supply to the desiccant regenerator	Flow measurement inside water pipe	The flow of the heating water supply to the internally heated desiccant regenerator will be measured to calculate the heat balance.
TR-21	Supply air (SA)	Supply air duct downstream of the HX	Temperature measurement inside air duct	The temperature of the air supply to the space LAB 123 is measured. The temperature is used to determine the heat balance for the space LAB 123.
TR-22	Supply air (SA)	Supply air duct downstream of the desiccant conditioner and upstream of the HX	Temperature measurement inside air duct	The temperature of the air supply is measured. The temperature is used to determine the heat balance for the internally cooled desiccant conditioner.
TR-23	Scavenger air return (SA)	Discharge air duct down-stream of the internally heated	Temperature measurement inside air duct	The temperature of the scavenger air discharge is measured. The temperature is used to determine the

Table 4.5.2: Instrumentation List for test set-up depicted in Figure 4.5.1

Main property: *T = Temperature; F = Flow; X = Humidity; S= Air speed*

Type of data acquisition: *R = Recording; I = Indicating*

ID	Medium	Location	Type	Function
		desiccant regenerator		heat balance for the internally heated desiccant regenerator.
FR-21	Supply air (SA)	Supply air duct downstream of the HX	Flow measurement inside air duct	The flow rate of the air supply to the space LAB 123 is measured to determine the heat balance for the space LAB 123.
FR-22	Supply air (SA)	Supply air duct downstream of the desiccant conditioner and upstream of the HX	Flow measurement inside air duct	The flow rate of the air supply to the space LAB 123 is measured to determine the heat balance of the internally cooled desiccant conditioner.
TR-23	Scavenger air return (SA)	Discharge air duct down-stream of the internally heated desiccant regenerator	Flow measurement inside air duct	The flow rate of the scavenger air discharge is measured to determine the heat balance for the internally heated desiccant regenerator.
XR-22	Supply air (SA)	Supply air duct downstream of the desiccant conditioner and upstream of the HX	Humidity measurement inside air duct	The humidity level of the air supply is measured. The humidity level is used to determine the humidity balance for the internally cooled desiccant conditioner.
XR-22	Scavenger air return (SA)	Discharge air duct down-stream of the internally heated desiccant regenerator	Humidity measurement inside air duct	The humidity level of the scavenger air return is measured. The humidity level is used to determine the humidity balance for the internally heated desiccant regenerator.

Table 4.5.2: Instrumentation List for test set-up depicted in Figure 4.5.1

<u>*Main property:*</u> *T = Temperature; F = Flow; X = Humidity; S= Air speed*

<u>*Type of data acquisition:*</u> *R = Recording; I = Indicating*

ID	Medium	Location	Type	Function
SIR-A	Room air	Below the ceiling fan (various locations)	Air speed measurements	The air speed in the room will be measured to determine the effect of air on the thermal comfort.

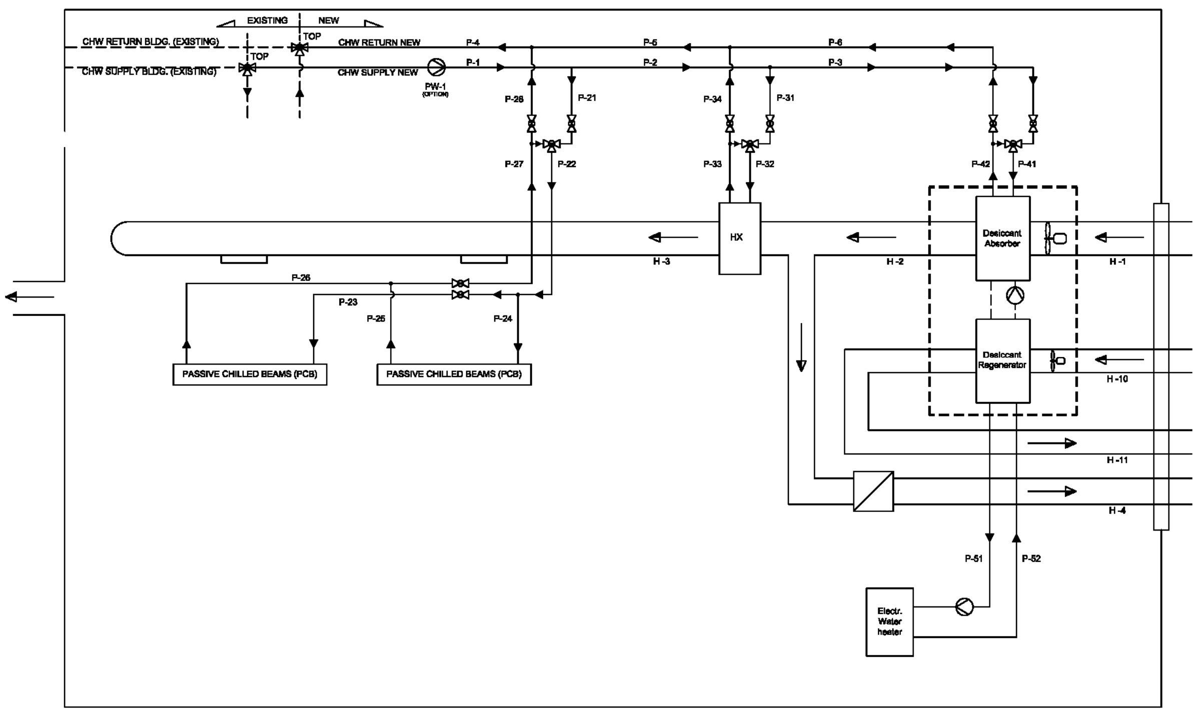


Figure 4.5.2: Identification of cooling and heating water pipes and air duct, as referred to in Table 4.5.3.; The *full-size drawing (Sheet-D-002) is presented in APPENDIX A*

Table 4.5.3: Piping and Ducting

No.	Medium		From	To	Flow rate		Material	Insulated
	type	Temp. F]			gpm	cfm		Y or N
P-1	Water	55	TOP	Junction P-21	15.4	-	PVC	Y
P-2	Water	55	Junction P-21, P-2	Junction P-31	12.1	-	PVC	Y
P-3	Water	55	Junction P-31, P-3	Mixing valve / P-41	7.8	-	PVC	Y
P-4	Water	66	Junction P-28, P-5	TOP	15.4	-	PVC	Y
P-5	Water	66	Junction P-34, P-6	Junction P-28, P-4	12.1	-	PVC	Y
P-6	Water	68	Junction P-42, mixing valve	Junction P-34	7.8	-	PVC	Y
P-21	Water	55	Junction P-1, P-2	Mixing valve / P-22	3.3	-	PVC	Y
P-22	Water	62	Mixing valve / P-21	Junction P-23, P-24	3.3	-	PVC	Y
P-23	Water	62	Junction P-24, P-22	PCB (following PCB in array)	2.5	-	PVC	Y
P-24	Water	62	Junction P-22, P-23	PCB (first PCB in array)	0.8	-	PVC	Y
P-25	Water	68	PCB (first PCB in array)	Junction P-26, P-27	0.8	-	PVC	Y
P-26	Water	68	PCB (following PCB in array)	Junction P-25, P-27	2.5	-	PVC	Y
P-27	Water	68	Junction P-25, P-26	Junction P-28, mixing valve	3.3	-	PVC	Y
P-28	Water	68	Junction P-27, mixing valve	Junction P-4, P-5	3.3	-	PVC	Y
P-31	Water	55	Junction P-2, P-3	Junction P-32, mixing valve	4.3	-	PVC	Y
P-32	Water	55	Junction P-31, mixing valve	HX	4.3	-	PVC	Y
P-33	Water	63	HX	Junction P-34, mixing valve	4.3	-	PVC	Y

Table 4.5.3: Piping and Ducting

No.	Medium		From	To	Flow rate		Material	Insulated
	type	Temp. F]			gpm	cfm		Y or N
P-34	Water	63	Junction P-33, mixing valve	Junction P-5, P-6	4.3	-	PVC	Y
P-41	Water	72	Junction P-3, mixing valve	Desiccant absorber	7.8	-	PVC	N
P-42	Water	83	Desiccant absorber	Junction P-6, mixing valve	7.8	-	PVC	N
P-51	Water	171	Desiccant regenerator	Electric water heater	25	-	Corrugated copper	Y
P-52	Water	177	Electric water heater	Desiccant regenerator	25	-	Corrugated copper	Y
H-1	Air	81	Outside environment	Desiccant absorber	-	800	Flexible hose	N
H-2	Air	89	Desiccant absorber	Junction H-2, HX	-	800	Flexible hose	N
H-3	Air	80	Junction H-2, HX	Inside space, diffuser	-	500	Flexible hose	N
H-4	Air	89	Junction H-2	Outside exhaust	-	300	Flexible hose	N
H-10	Air	81	Outside environment	Desiccant regenerator	-	300	Flexible hose	N
H-11	Air	146	Desiccant regenerator	Outside exhaust	-	300	Rigid ducts	Y

4.6 Description of Test Set up (Drawings)

The test set-up in LAB 123 integrates AILR LDAC system into supporting infrastructure and sensible cooling technologies. This section shows floor plans and section drawings of the test set-up configuration. The individual sheets are presented in full size in APPENDIX A.

Figure 4.6.1 shows the existing conditions in room LAB 123 and the adjacent exterior space. A 7 foot wide walkway runs along the building. The outside space can be accessed through an exit door in the adjacent hallway. The drawing shown in Figure 4.6.1 is Sheet P-001; the full-size drawing is presented in APPENDIX A.

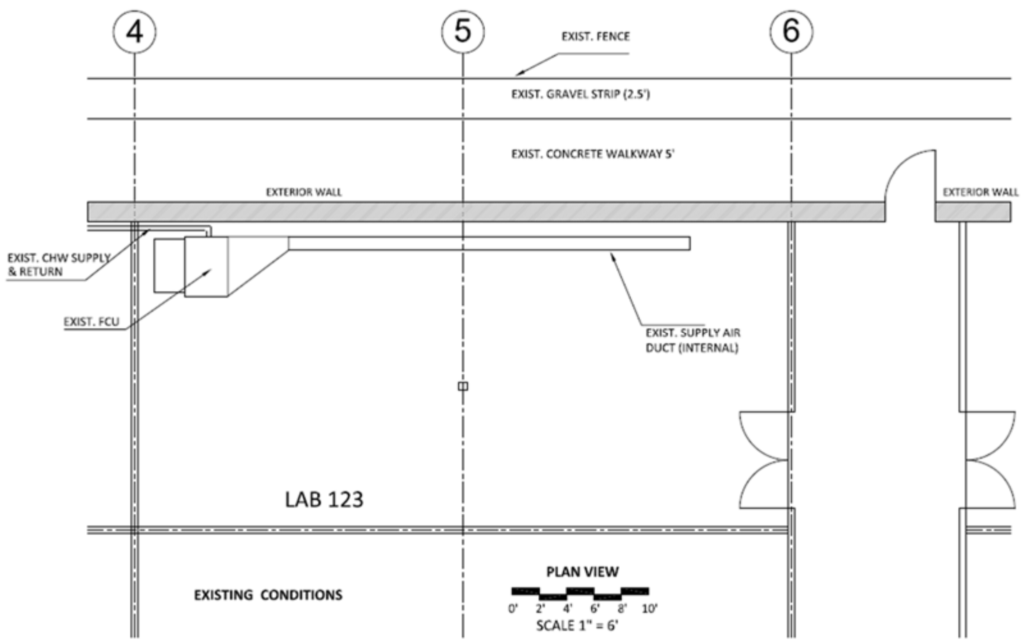


Figure 4.6.1: Existing conditions of space LAB 123 - floor plan; The full-size drawing (Sheet-P-001) is presented in APPENDIX A

Figure 4.6.2 shows the floor plan of LAB 123 with the LDAC installed. Only the supply and discharge air ducts for the desiccant regenerator are shown. The existing fan coil unit (FCU) with the attached air supply duct are shown in light grey color; these existing space components will remain in the space but will be disconnected and not be used while the LDAC system is operating.

Figure 4.6.3 shows the floor plan of LAB 123 with the LDAC installed with all air ducts. Figure 4.6.4 shows the mirrored ceiling plan of LAB 123 with the LDAC system outlined. The passive chilled beans (PCBs) and the ceiling fan are depicted. Figure 4.6.5 shows details of the layout adjacent to the LDAC unit. The figure shows all air ducts which are connected to the LDAC and, in addition, the 12-inch indoor air supply flexible duct. The figure also shows the water-to air heat exchanger which provides cooling to the supply air downstream of the LDAC unit and the branch-off to the air supply bypass.

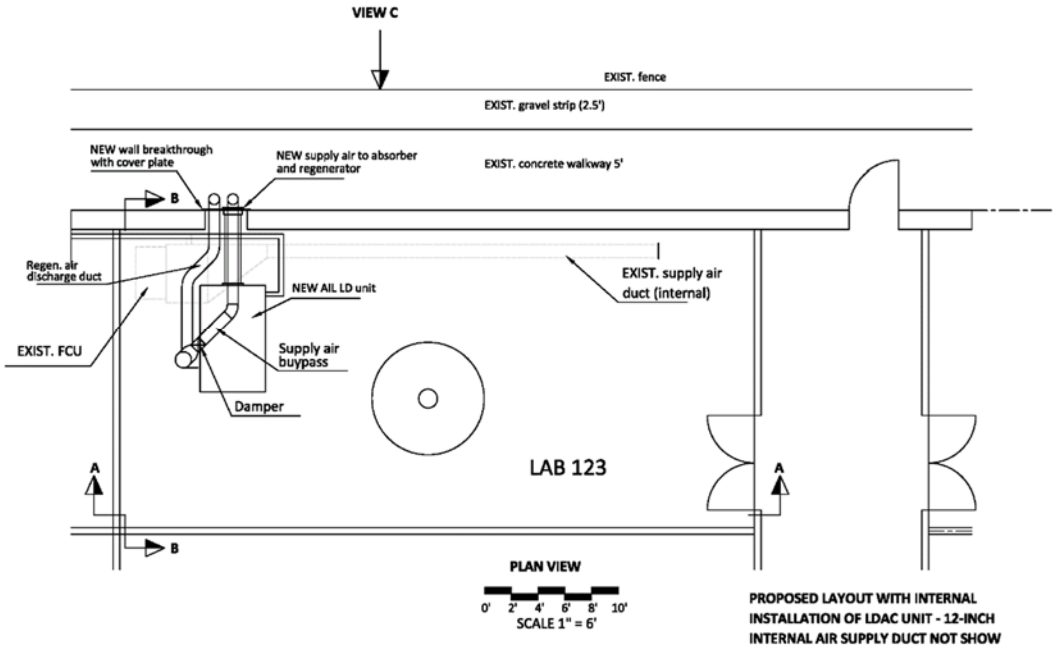


Figure 4.6.2: LAN 123 floor plan with proposed LDAC installed; only the air hoses of the desiccant generator are depicted; The full-size drawing (Sheet-P-002) is presented in APPENDIX A

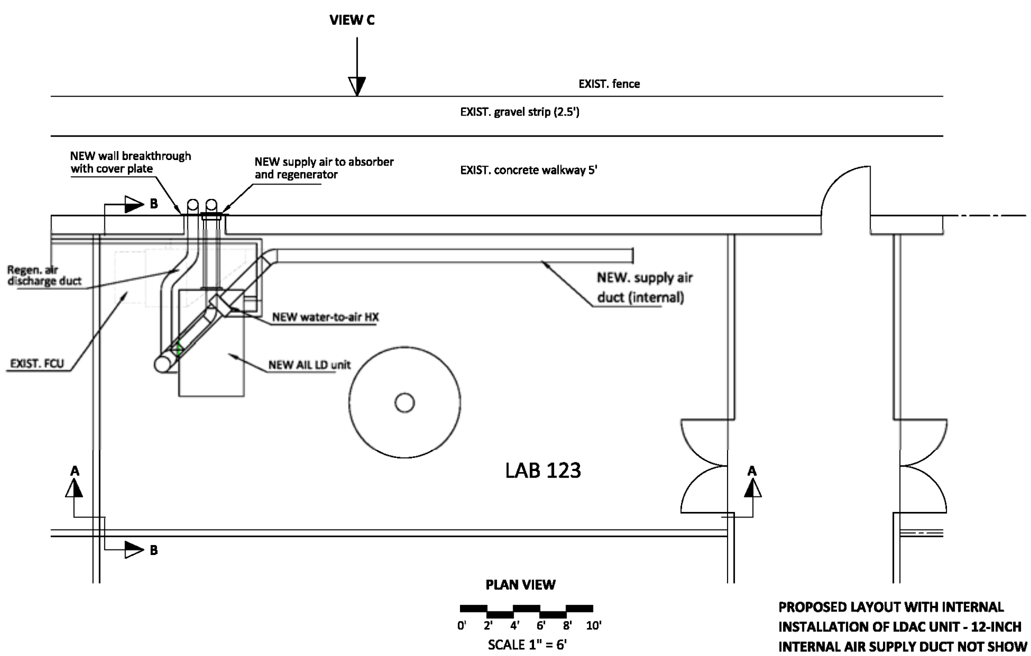


Figure 4.6.3: LAN 123 floor plan with proposed LDAC installed; all air hoses are depicted; The full-size drawing (Sheet-P-003) is presented in APPENDIX A

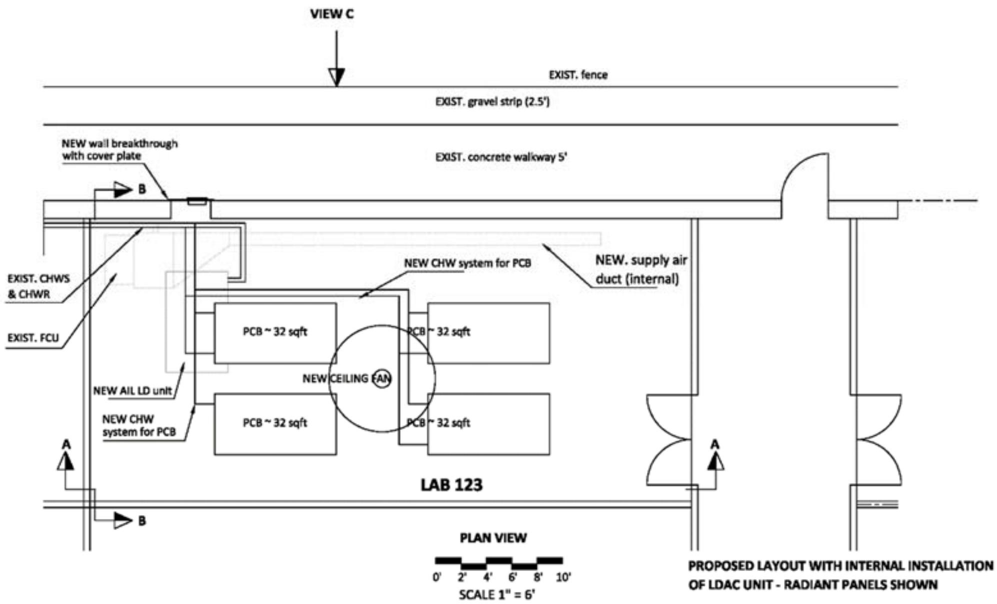


Figure 4.6.4: LAN 123 floor plan with proposed LDAC installed; only the air hoses of the desiccant generator are depicted; The full-size drawing (Sheet-P-004) is presented in APPENDIX A

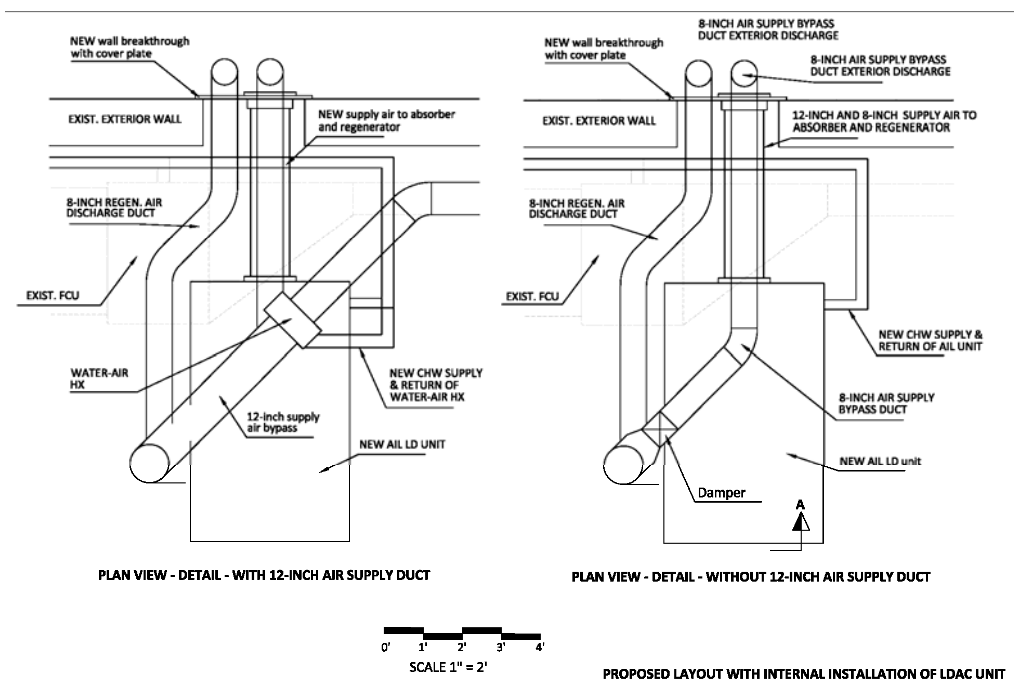


Figure 4.6.5: LAN 123 detail floor plan with proposed LDAC installed; The full-size drawing (Sheet-P-005) is presented in APPENDIX A

Figure 4.6.6 shows Section A-A, with one image showing the entire room LAB 123, and the other showing a detailed section.

Figure 4.6.7 shows Section B-B, with one image showing the entire room LAB 123, and the other showing a detailed section. The section shows the new wall break-through with the installed cover plate.

Figure 4.6.8 shows the new wall break-through with the installed cover plate. The flexible hoses are connected to welded fittings that penetrate the cover plate and serve as attachment points of the interior flexible hoses.

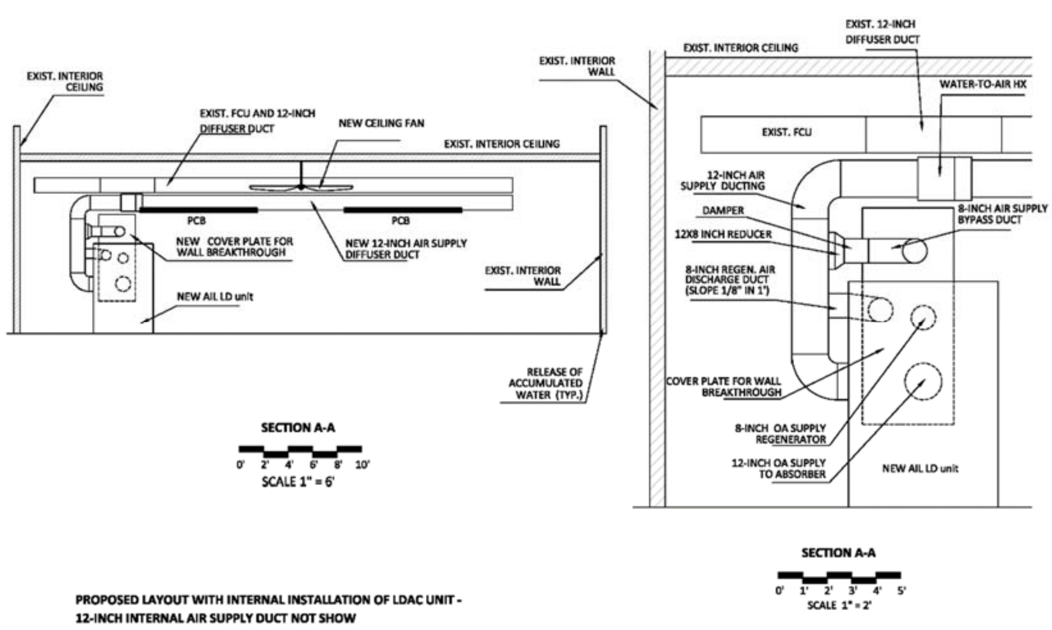


Figure 4.6.6: Section A-A, overview of room LAB 123 and detail around the installed LDAC unit; The full-size drawing (Sheet-P-006) is presented in APPENDIX A

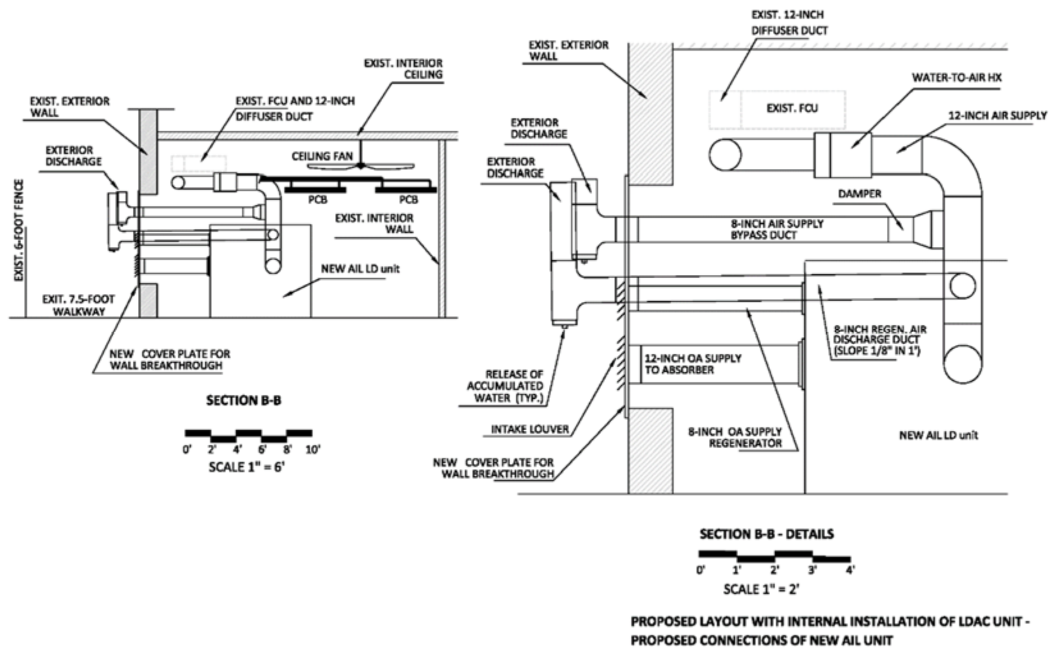


Figure 4.6.7: Section B-B, overview of room LAB 123 and detail around the installed LDAC unit; The full-size drawing (Sheet-P-007) is presented in APPENDIX A

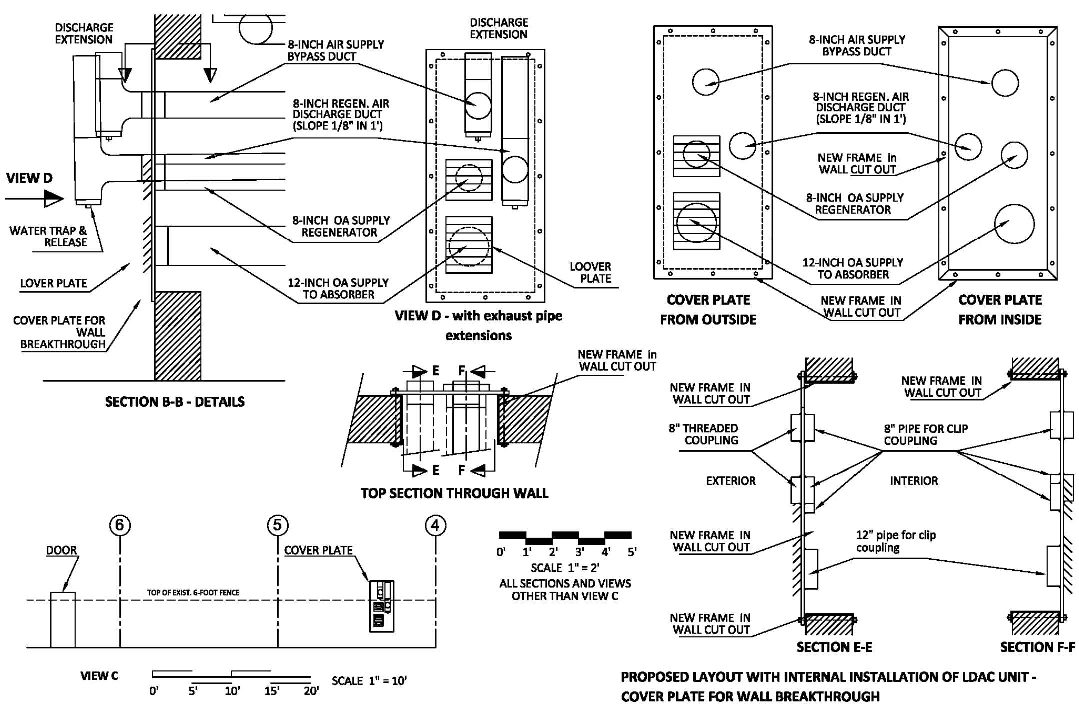
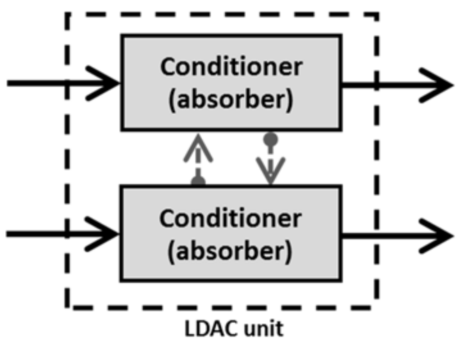
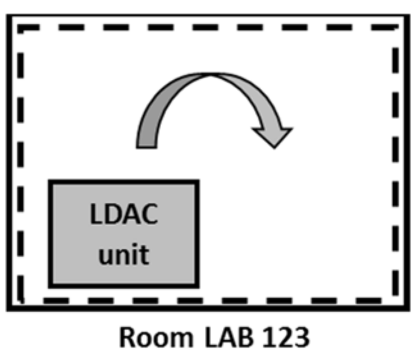
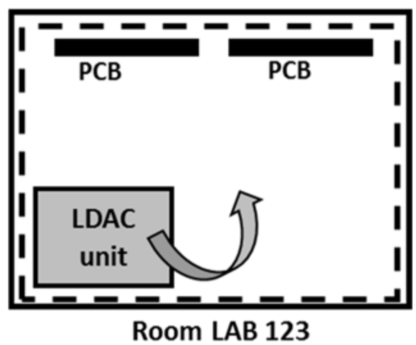
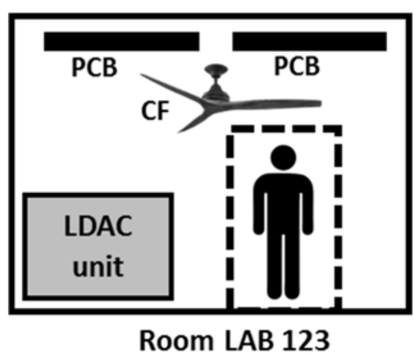


Figure 4.6.8: The new all break through and cover plate; The full-size drawing (Sheet-P-008) is presented in APPENDIX A

4.7 Preliminary Test Approach

The initial test program will structure the types of tests to serve the objectives of Project phase II/A (see Section 2.1). The test program will be comprised of four test task groups, with different objectives and performance metrics. Table 4.7.1 provides an overview of the four test task groups. Table 4.7.1 shows simple process diagrams which illustrate the scope and focus of the test task groups. The dashed lines in the simple process diagrams indicate the systems boundary of the test model, which means those properties and parameters that affect the tests "within" the system boundaries. The four test task groups are later described in more detail.

Test Task Group	Process diagram	Type of testing
1		<u>Evaluate the dehumidification performance of the AILR LDAC unit under static and transient process conditions:</u> Temperature (air and water), mass flow (air and water) and humidity sensors will be placed inside and at the system boundary to record the effects of changing conditions on the performance characteristics of the LDAC unit.
2		<u>Evaluate the response of the room LAB 123 to various air supply conditions of the LDAC unit:</u> Temperature (air and MRT) and humidity sensors will be placed inside room LAB 123 to evaluate the environmental response of the indoor space to changing supply air conditions of the LDAC unit.

Test Task Group	Process diagram	Type of testing
3		Evaluate the performance of the sensible cooling components, passive chilled beams and ceiling fan, under various indoor humidity and temperature levels: Temperature (air and MRT), humidity sensors and air speed metes will be placed inside room LAB 123 to evaluate the sensible heat management performance of the cooling components under changing indoor temperatures and humidity levels.
4		Evaluate thermal comfort in room LAB 123 under various indoor humidity and temperature levels as well as under different cooling levels: Temperature (air and MRT), humidity sensors and air speed metes will be placed inside room LAB 123 to determine the environmental thermal comfort conditions. Test subjects will participate in test runs to determine their thermal comfort sensation.

Test Task Group 1: Evaluate the dehumidification performance of the AILR LDAC unit under static and transient process conditions, as follows:

1.1. The LDAC unit will be operated under different processes parameters which include the following:

1.2. Temperature and flow rate of the desiccant solution through the conditioner and the regenerator will be modulated.

1.3. Air flow rate through the desiccant conditioner and the resulting dew point of the dried air will be modulated.

1.4. Effects of different cooling water temperatures on the performance of the conditioner will be evaluated.

1.5. Effects of different heating temperatures on the performance of the regenerator will be evaluated.

1.6. Effects of different outside air temperatures and humidity levels will be evaluated

1.7. Dynamic response of the conditioner and regenerator to different indoor and outdoor conditions will be evaluated during longer and shorter test series.

1.8. Effects of On-OFF cycles and the time to achieve steady state operation at different indoor and outside conditions will be evaluated.

<u>Test Task Group 2:</u> Evaluate the response of the room LAB 123 to various air supply conditions of the LDAC unit:

Dynamic response of indoor air and radiant temperature as well as humidity level on various LDAC unit's outputs, will be evaluated, including:

2.1. Flow rate of supply air

2.2. Temperature of supply air

2.3. Humidity of supply air

<u>Test Task Group 3:</u> Evaluate the performance of the sensible cooling components, passive chilled beams and ceiling fan, under various indoor humidity and temperature levels:

3.1. Effects of cooling performance of passive chilled beams with different supply and return temperatures will be tested.

3.2. Effects of cooling performance of passive chilled beams under different humidity levels in the room will be tested.

3.3. Observation of condensation on chilled beams operating at different temperatures and at different room humidity and temperature level will be evaluated by modulating the dew point inside the conditioned space.

3.4. Effects of different speeds of and the relative position of the ceiling fan to the passive chilled beams on the heat transfer performance of the chilled beams.

Test Task Group 4: Evaluate thermal comfort in room LAB 123 under various indoor humidity and temperature levels as well as under different cooling levels:

Determine the thermal comfort in the room with and without the cooling provided by the passive chilled beams and ceiling fans under the following conditions:

4.1. Various air temperatures, and resulting MRT

4.2. Various humidity levels

4.3. Various air speeds

4.4. Various clothing insulation and metabolic rates

4.8 Preliminary Budget for Project Phase II/A

This section presents the preliminary budget for the test equipment, labor and consumables to conduct the test program as delineated in Section 4.7. Table 4.8.1 shows the preliminary budget. For the budget items with quotes received, the proposals are furnished in APPENDIX B.

Table 4.8.1: Summary of project cost estimate (preliminary)

Project Cost Summary

Item		Description	Amount	Quote received	Lookup
A		**Main test set-up equipment**			
A	1	AILR LDAC units	$129,214	Quote	
A	2	Electric water heater / Hubbell Electric Heater V624T	$4,645	Quote	
A	3	Radiant System by Barcol-Air Ltd., Product :BRW	$5,921	Quote	
A	4	Shipping for Radiant System by Barcol-Air	$500	Quote	
A	5	Ceiling fan	$500		Estim.
A	6	Humidifier	$900	Quote	
		sum	**$141,680**		
B		**Test set-up infrastrcuture**	$800		Estim.
B	1	CHW and hot water Piping			Estim.
B	2	Hardware	$1,343		Estim.
B	3	Labor (mechanical contractor)	$1,260		Estim.
B	4	Duct hoses & fittings	$1,040		Estim.
B	5	Instruments & transducers	$5,650		Estim.
B	6	Data acquisition	$1,600		Estim.
B	7	Flexible air hoses and appurtenances	$1,040		Estim.
B	8	Electric installation	$1,400		Estim.
B	9	Wall breakthrough	$2,310		Estim.
		Miscellaneous	$2,500		
		sum	**$18,943**		
C		**Staff**			
C	1	Co-principal Investigator (Consultant) and team leader of UHM research team	**x**		
C	2	ERDL staff (~50%)	**x**		
C	3	Two Student helps data	**x**		
C	4	Student helps test operation	**x**		

4.9 Lab Safety Considerations

Work activities related to the installation and the following testing of the integrated LDAC system in room LAB 123 will follow the safety guidelines promulgated by the UHM Environmental Health and Safety Office.

Since the planned test series involves working with electrical and hot equipment as well as chemicals a strict lab safety routine will be maintained. For the present research work exposure to chemicals will be limited to periodically checking the level of the lithium chloride solution the LDAC unit, similarly of checking the refrigerant in a conventional AC system.

The University of Hawaii Environmental Health & Safety Office (EHSO) was contacted and the nature and scope of the planned work in LAB 123 was discussed. The representative of EHSO indicated that a comprehensive Lab safety program is available which consists of online resources and classroom training. (refer to http://www.hawaii.edu/ehso/lab-safety-training/)

Lab Safety Training:

Attending the lab safety training is required on an annual basis following completion of the initial course provided by UH EHSO staff. Refresher sessions should be conducted by the individual PI or designated lab manger using the Lab Personnel Safety Checklist as a guide.

All resources as well as lab safety training documentation should be kept on hand for review during EHSO or other regulatory agency inspections.

Lab Safety https://www.hawaii.edu/ehso/lab-safety-training/

Hazardous Waste Generator Training (if any chemical waste is generated):
https://www.hawaii.edu/ehso/hazardous-waste-generator-training/ .

For the UH EHSO inspections of off-campus facilities, a safety checklist will be used as a guide.

<u>The safety checklist is attached in the Appendix C.</u>

<u>The Material Safety Data Sheet Lithium Chloride is attached in the Appendix C.</u>

Chemical Hygiene Plan

The objective of this CHP is to provide uniform requirements for safe use and disposal of potentially hazardous substances in University laboratories. A variety of hazardous chemicals are used in small quantities in research and teaching laboratories creating a unique environment with a number of risks. These chemicals may cause injury or damage because they are toxic, flammable, corrosive, or reactive with water and other materials. How these substances are handled will determine the degree of risk. General standard operating procedures are outlined, including work with select carcinogens, reproductive toxins, and substance with a high degree of acute toxicity. Specific standard operating procedures must be developed by each lab for operations posing a special hazard, for example, heating phosphoric acid, working with pyrophorics, conducting electrophoresis, distillations, extractions, etc

The Chemical Hygiene Plan is attached in the Appendix C.

Safety Data Sheets (MSDS)

The safety data sheet for the Lithium Chloride, which will be used in the AILR LD system, is presented in the Figure 4.9.1.

Material Safety Data Sheet

Catalog Number: 150134
Revision date: 26-Apr-2006

NFPA

1. IDENTIFICATION OF THE SUBSTANCE/PREPARATION AND COMPANY INFORMATION

Catalog Number: 150134

Product name: LITHIUM CHLORIDE ANHYDROUS

Synonyms: Chlorku litu; Chlorure de lithium; Lithiumchlorid
Supplier:
MP Biomedicals, LLC
29525 Fountain Parkway
Solon, OH 44139
tel: 440-337-1200

Emergency telephone number: CHEMTREC: 1-800-424-9300 (1-703-527-3887)

2. COMPOSITION/INFORMATION ON INGREDIENTS

Components	CAS Number	Weight %	ACGIH Exposure Limits:	OSHA Exposure Limits:
LITHIUM CHLORIDE ANHYDROUS	7447-41-8	90 - 100%	None	None

3. HAZARDS IDENTIFICATION

EMERGENCY OVERVIEW: The product causes burns of eyes, skin and mucous membranes. Harmful if swallowed.

Category of Danger:
Corrosive , Harmful , Lachrymator

Principle routes of exposure: Skin
Inhalation: Vapors or dusts will cause burns of respiratory passages.
May be harmful if inhaled.
Ingestion: Can burn mouth, throat, and stomach
Harmful if swallowed.
Skin contact: Causes skin burns

Figure 4.9.1: Material Safety Data Sheet (MSDS) for Li Cl Anhydrous (partial); complete MSDS is presented in Appendix C

SECTION 5 - TESTS OF PHASE II/B - PILOT INSTALLATION AT LOCATION TBD

This section describes objectives and plans for Project Phase II/B. During Phase II/B the LDAC unit will be installed at a test location, e.g. indoor space, whose size and space conditioning requirements will match the LDAC unit that was used and tested in Phase II/A.

5.1 Objectives and Approach of Project Phase II/B

After successful testing the performance of the AILR LDAC to gain operational experience, the LDAC unit in Project Phase II/A will be installed at a pilot installation location in Project Phase II/B.

The main objective of Project Phase II/B is to showcase and confirm the capabilities of the LDAC technology to provide healthy, comfortable and productive IEQ conditions in a real-world scenario. Candidate pilot installation locations could be offices, classrooms and library spaces.

The objectives of the test installation will be to provide increased ventilation of indoor spaces and provide precise humidity control in the range of 40% to 50% relative humidity (RH). The targeted ventilation rates for the following space functions will be higher than the ASHRAE minimum recommended outside air rates:

Table 5.1.1: Proposed ventilation rates for candidate locations of pilot installation

			Ventilation rate	
Occupancy Category	**Floor area**	**Occupancy**	ASHRAE rates ***(note 1)***	**Selected for pilot installation**
	sqft	person	cfm	cfm
Office space	5,800	55	610	**1,200**
Classroom	1,900	40	610	**1,200**
Library space	3,500	35	595	**1,200**

Note (1): Minimum Ventilation Rates in Breathing Zone (ASHRAE 62.1 - 2013)

5.2 Main Characteristics of Planned Test Set-up for Pilot installation in Project Phase II/B

The integrated LDAC system which will be used to provide good thermal comfort, indoor environmental and air quality and provide the prescribed ventilation rates will have the following main characteristics:

The LDAC unit will remove the latent cooling load and provide the ventilation rates via a dedicated outdoor air system (DOAS). The humidity level in the space will be measured and controlled by regulating the dew point and the air flow rate of the outside air admitted to the conditioned space.

The passive chilled beams (PCBs) will provide sensible cooling which will be entirely independent of the dehumidification process. The PCBs will be controlled by temperature sensors in the spaces and the flow rates of chilled water will be modulated to fit the cooling needs of the conditioned spaces. The installation and operation of the PCBs do not require additional air ducting in the spaces. The sensible heat rejected from the spaces is entirely done by pumping water, which is significantly more energy efficient than pumping cold air through ventilation ducts for sensible cooling. In addition, energy savings are realized by using higher cooling water temperatures than in conventional AC systems.

A water-to-air heat exchanger will be installed in the DOAS downstream of the LD system to provide sensible cooling.

A membrane-based enthalpy recuperation (recovery) device is an option to recover energy spent for dehumidification. The membrane enthalpy recuperation would be installed in a junction of the supply and discharge duct of the DOAS ventilation system, upstream of the LD conditioner. Since there would be only small temperature but significant absolute humidity differentials between the supply and the outside air, the main benefit would be to lower the latent load. Consequently, the square footage, which can be served by the LDAC system would be increased. Implementing the membrane enthalpy recovery unit will, however, require that the return air is collected and discharged at a point such that the discharge air duct is in close proximity to the outside air intake duct.

Air filtration will be provided by outside air flowing through the liquid desiccant conditioner. The liquid-air contact will remove particulate pollutants in the outside air. Additionally, the liquid desiccant acts as a disinfectant to potential pathogens contained in the air supply.

5.3 Basic Process Diagram of Phase II/B Test Set-up

Figure 5.3.1 shows an illustration of the basic process for the integrated LDAC system to be used in Project Phase II/B. Table 5.3.1 describes the main systems parts. Table 5.3.1 further indicates what system components can be reused and which must be purchased new after the completion of the initial test in room LAB 123.

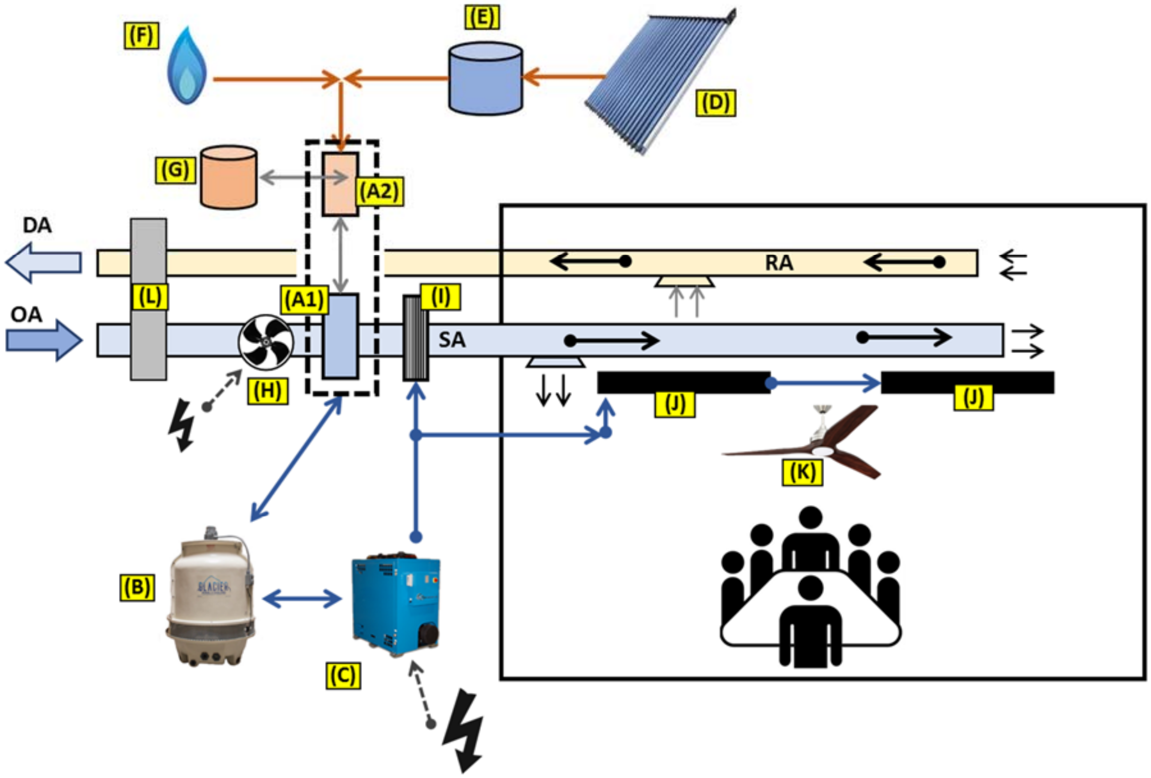
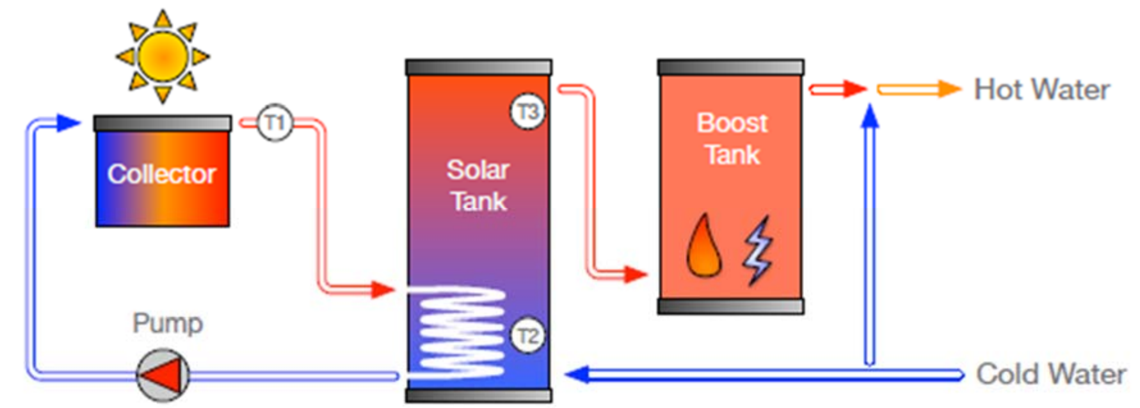


Figure 5.3.1: Illustration of the basic process of the pilot installation in Project Phase II/B

ID	Description and function	New purchase required	Image (if applicable)
OA	Outside air supply	N/A	
SA	Supply air	N/A	
RA	Return air	N/A	
DA	Discharge air	N/A	

ID	Description and function	New purchase required	Image (if applicable)
	A1 and A2 - the desiccant conditioner and regenerator are integrated in one containerized assembly	**NO:** The LDAC system used in the initial tests in marine Center Room LAB 123 can will be used	
A1	Liquid desiccant conditioner (absorber): The liquid desiccant absorber reduces the humidity in the outside air supply		
A2	Liquid desiccant regenerator is removing the humidity in the saturated desiccant solution making it suitable for reuse in the desiccant conditioners		
B	Evaporator cooling tower removes the heat of absorption from the desiccant absorber. The cooling tower will also be used as a heat sink for the vapor compression chiller condenser cooling.	**Yes:** A new cooling tower will be required	

ID	Description and function	New purchase required	Image (if applicable)
C	Vapor compression (VC) chiller will provide sensible cooling load for the Heat exchanger (H) and the passive chiller beams. The size of the VC will depend on the sensible load and is estimated at around 3 to 4 tons.	**Yes:** A new VC chiller will be required	
D	High Efficiency Evacuated Tube Solar Thermal Collector will provide a large portion of the heat required for the desiccant regeneration. The size envisioned size of the solar array is about 1,000 square feet, which includes the solar tunes and space for maintenance. Initial sizing has determined that about 12 panels with 30 evacuated tubes will be sufficient to provide heat more than 70% of the required heat.	**Yes:** A new solar panel array will be required	
E	Solar tank: A solar tank will be used as hot water storage and buffer tank for the solar array.	**Yes**: A new solar tank will be required	The image below illustrates an integrated system of solar collectors (D), solar tank (E) and Boost tank (F). (source of image (www.apricus.com)
F	Gas fired water heater; The gas heater will provide heat for regeneration during periods of insufficient solar heat.	**Yes**: A new boost tank will be required	

ID	Description and function	New purchase required	Image (if applicable)
			
G	Concentrated liquid desiccant solution storage and buffer tank serves as an alternative to the hot water solar tank.	<u>**Yes**</u>: A new storage and buffer tank will be required	
H	Supply air fan in the DOAS system: The supply air fan provides the required pressure to supply the conditioned spaces	<u>**Yes:**</u> A new supply fan will be required	
I	Water-to-air heat exchanger (HX) removes sensible heat from the supply air downstream of the desiccant conditioner (A1). The heat removal capacity of the HX will be modulated to adjust for the desired cooling of the supply air provided to the conditioned spaces	<u>**Yes:**</u> A new HX will be required	
J	Passive chilled beams provide a larger portion of the sensible cooling load. The passive chilled beams receive their	<u>**No and Yes:**</u> The PCBs used in the	

ID	Description and function	New purchase required	Image (if applicable)
	cooling capacity through a chilled water supply. The passive chilled beams provide cooling capacity through a combination of convective and radiative heat rejection.	initial test in LAB123will be reused. Additional PCBs will be required	
K	Ceiling fans provide cooling capacity to occupants through a combination increased sensible and latent heat loss. The elevated wind speed over the occupants increase their convective heat loss and an increase of evaporation from their exposed skin area. The cooling capacity provided by the ceiling fans are very cost effective if advanced and energy saving fans are used. The best results are obtained a high-volume air "shower" while avoiding too high air movements.	**<u>No and Yes:</u>** The ceiling fan used in the initial test in LAB123will be reused. Additional ceiling fans will be required	
L	Membrane-based enthalpy recuperation device: The dense polymer membrane uses selective transfer technology to allow heat and water vapor to permeate through, while blocking contaminant compounds. The transfer is driven by temperature and humidity differentials between the airstreams.	**<u>Yes:</u>** A new system must be acquired	

5.4 Proposed Duration of Initial Testing During Project Phase II/B

Once installed at the site of the pilot project in Project Phase II/B, the LDAC system should operate for an extended time to obtain long term operational experience and prove the LDAC technology is suitable for applications in Hawaii climate. It is proposed that an entire year be chosen for the length of the initial test in Phase II/B to obtain operational data that reflect season changes.

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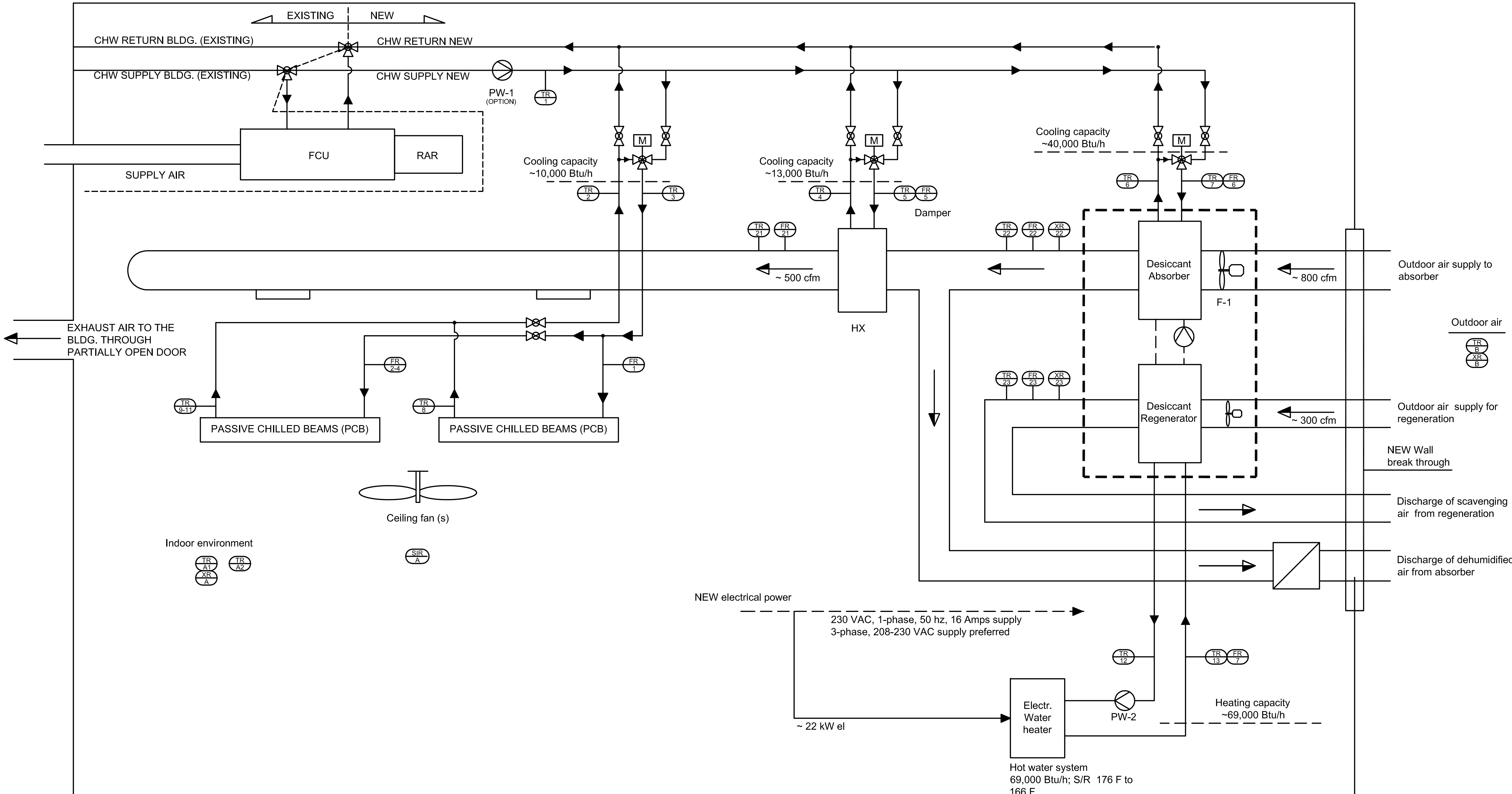
APPENDICES

APPENDIX A.

Technical drawings of test set-up in LAB 123



BASIC PROCESS FLOW DIAGRAM - HEAT SINK IS EXISTING BLDG. CHW SYSTEM

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BASIC PROCESS DIAGRAM OF PROPOSED SYSTEM USING EXIST. BLDG. CHW SYSTEM AS HEAT SINK

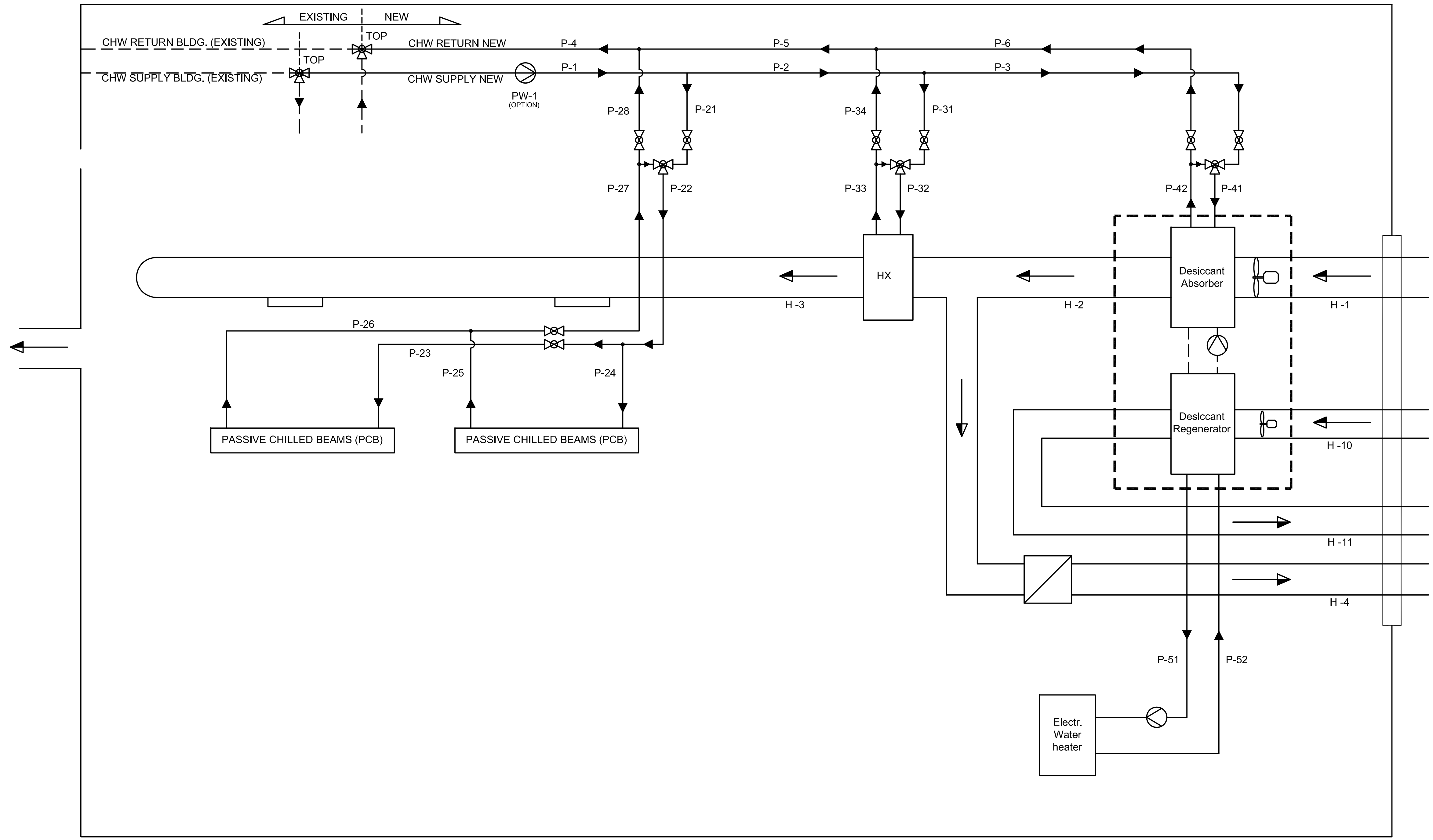
PROJECT
HNEI – LD 2017

DATE
Oct.19, 2017

DRAWN BY
M. ZAPKA

SCALE
NO SCALE

SHEET NO.
D-001



PIPING DIAGRAM AND DUCTING DIAGRAM

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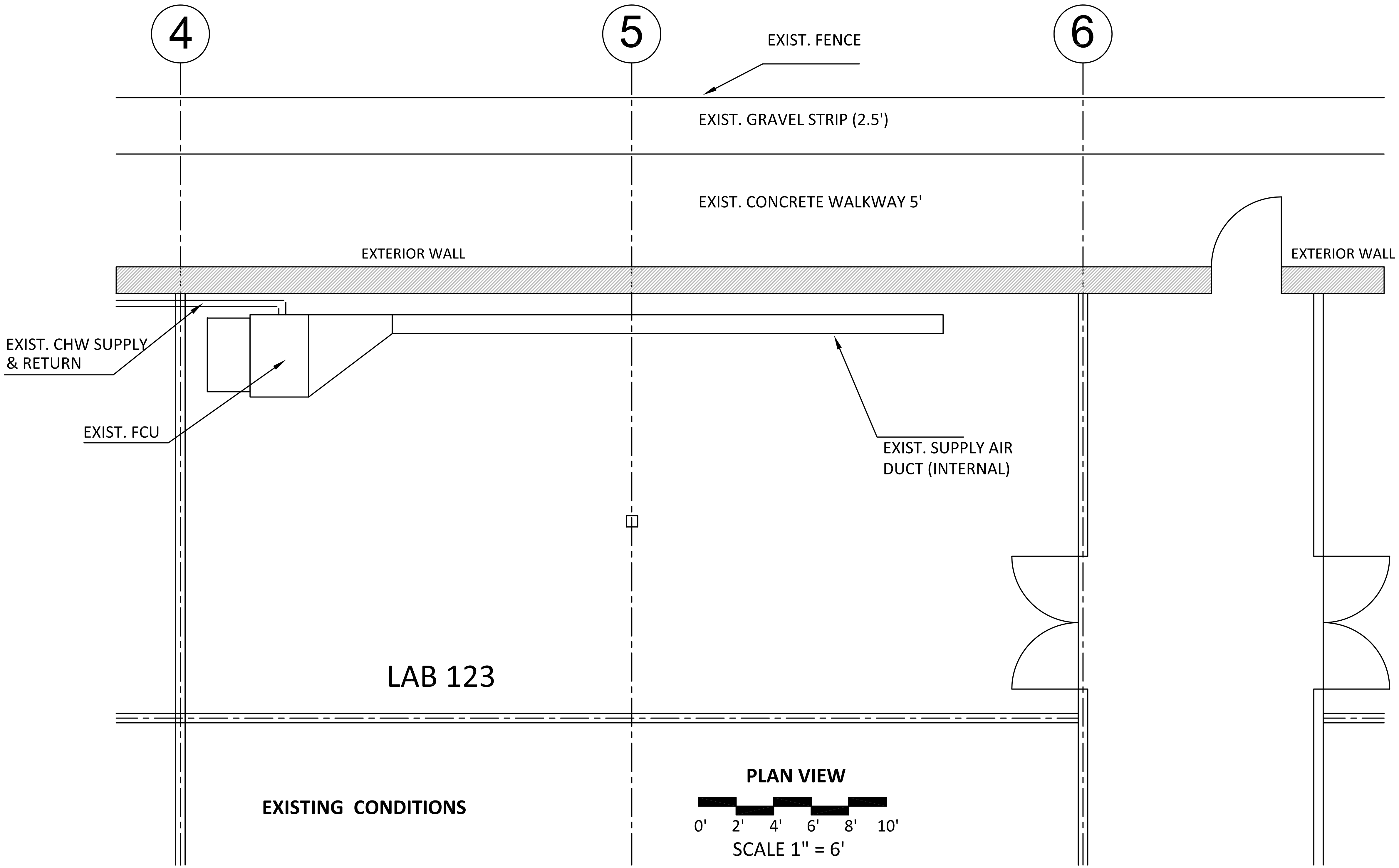
PIPES AND DUCTING DIAGRAM

PROJECT	HNEI – LD 2017
DATE	Oct.19, 2017
DRAWN BY	M. ZAPKA
SCALE	NO SCALE
SHEET NO.	D-002



EXISTING CONDITIONS

PLAN VIEW

SCALE 1" = 6'

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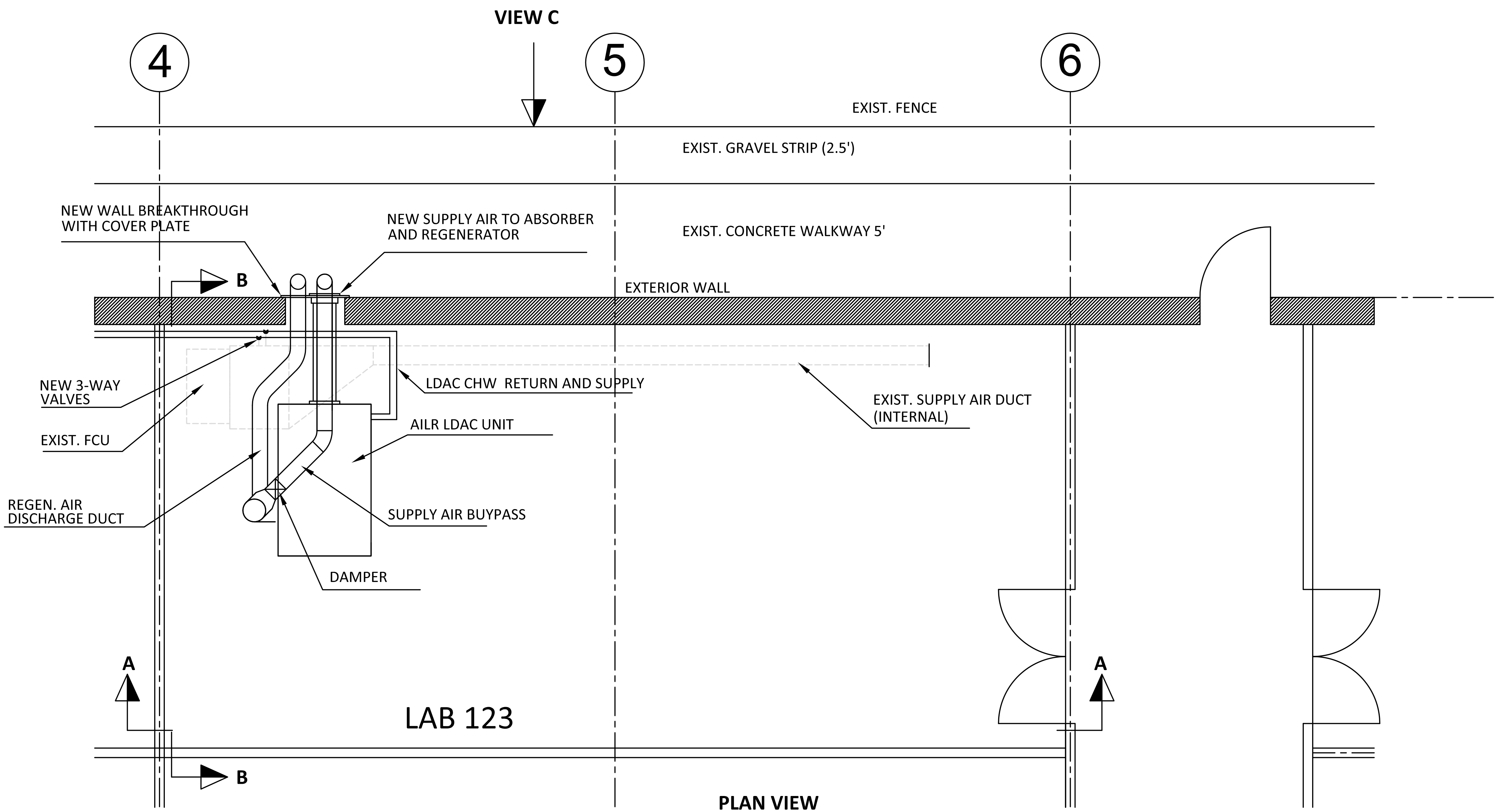
Sustainable Design & Consulting LLC

HNEI - LIQUIDE DESICCANT TEST FACILITY

LAB 123 IN MARINE SCIENCE BLDG.
EXISTING CONDITIONS

PROJECT	HNEI – LD 2017
DATE	Oct.19, 2017
DRAWN BY	M. ZAPKA
SCALE	1" = 6'
SHEET NO.	**P-001**



PLAN VIEW

0' 2' 4' 6' 8' 10'

SCALE 1" = 6'

PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF LDAC UNIT - 12-INCH INTERNAL AIR SUPPLY DUCT NOT SHOWN

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HNEI - LIQUIDE DESICCANT TEST FACILITY
Proposed layout of LDAC system in LAB 123 INTERNAL AIR SUPPLY DUCTS NOT SHOWN

PROJECT
HNEI – LD 2017

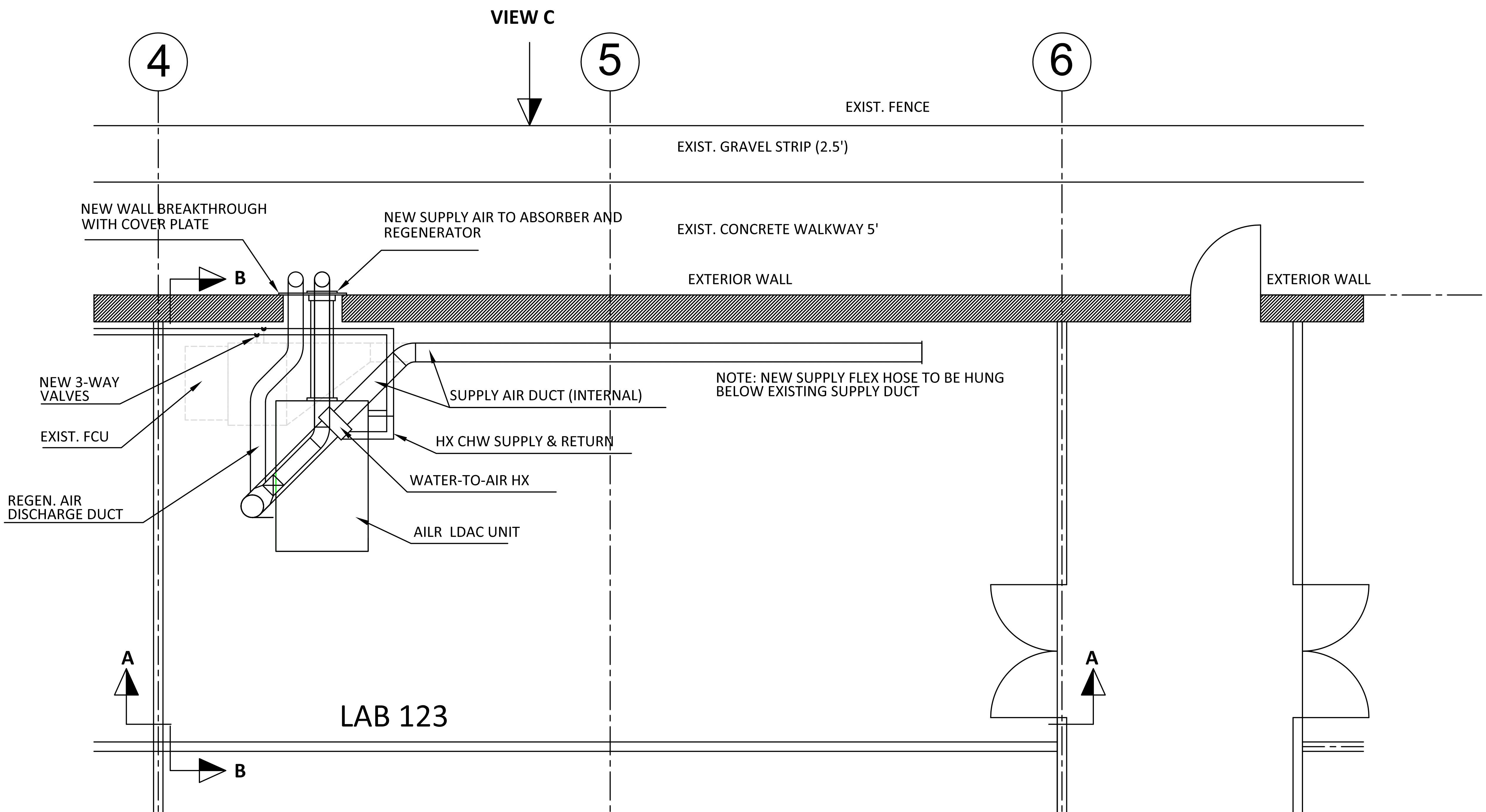
DATE
Oct.19, 2017

DRAWN BY
M. ZAPKA

SCALE
1" = 6'

SHEET NO.

P-002



PLAN VIEW

0' 2' 4' 6' 8' 10'

SCALE 1" = 6'

PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF LDAC UNIT - 12-INCH ALL AIR DUCTS ARE SHOWN

HNEI Hawai'i Natural Energy Institute
University of Hawai'i at Mānoa

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HNEI - LIQUIDE DESICCANT TEST FACILITY

PROPOSED LAYOUT OF LDAC SYSTEM IN LAB 123
ALL AIR DUCTS ARE SHOWN

PROJECT
HNEI – LD 2017

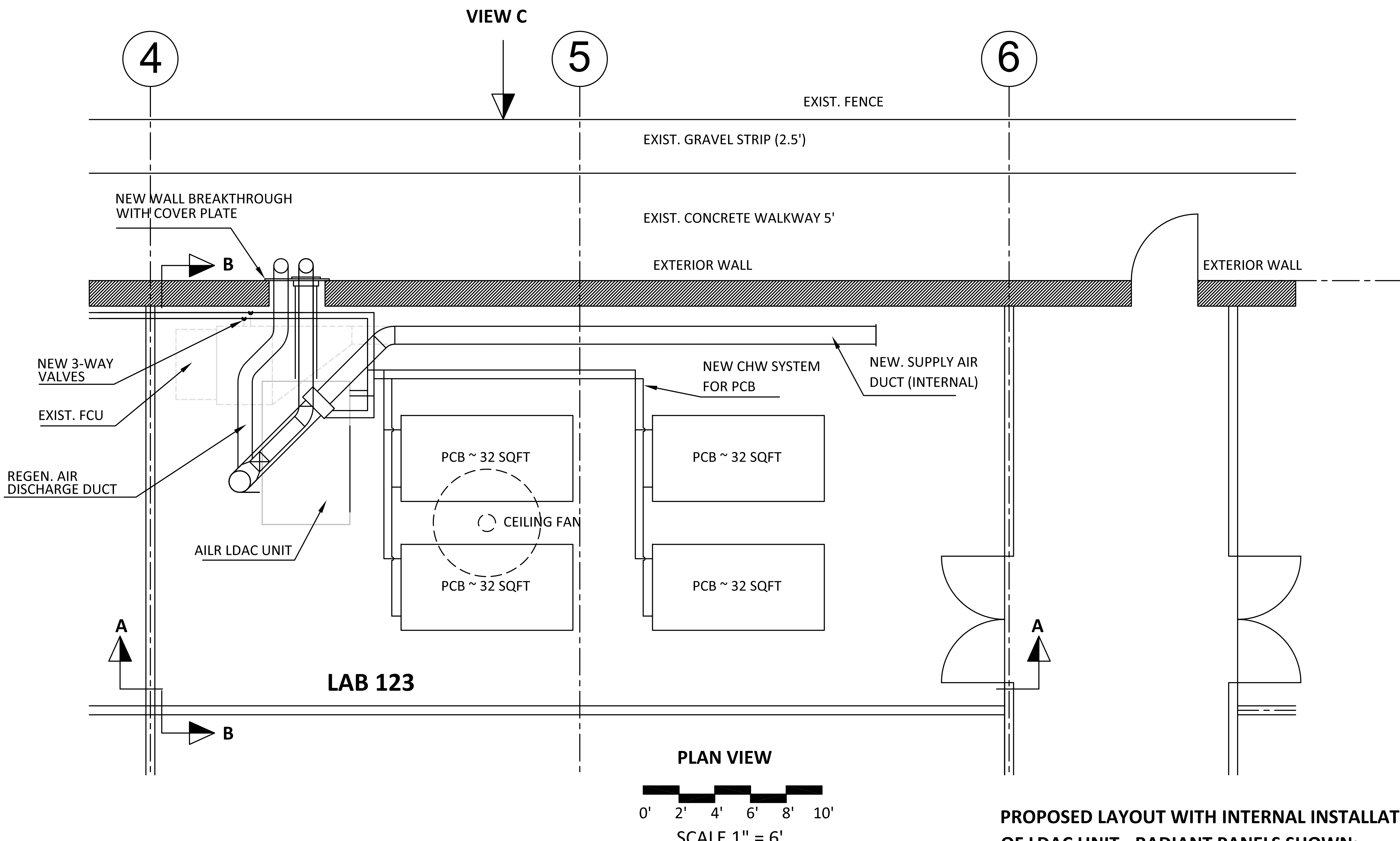
DATE
Oct.19, 2017

DRAWN BY
M. ZAPKA

SCALE
1" = 6'

SHEET NO.

P-003



PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF LDAC UNIT - RADIANT PANELS SHOWN; REFLECTED CEILING PLAN

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University of Hawai'i at Mānoa

Sustainable Design & Consulting LLC

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PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF LDAC UNIT - RADIANT PANELS SHOWN; REFLECTED CEILING PLAN

PROJECT
HNEI – LD 2017

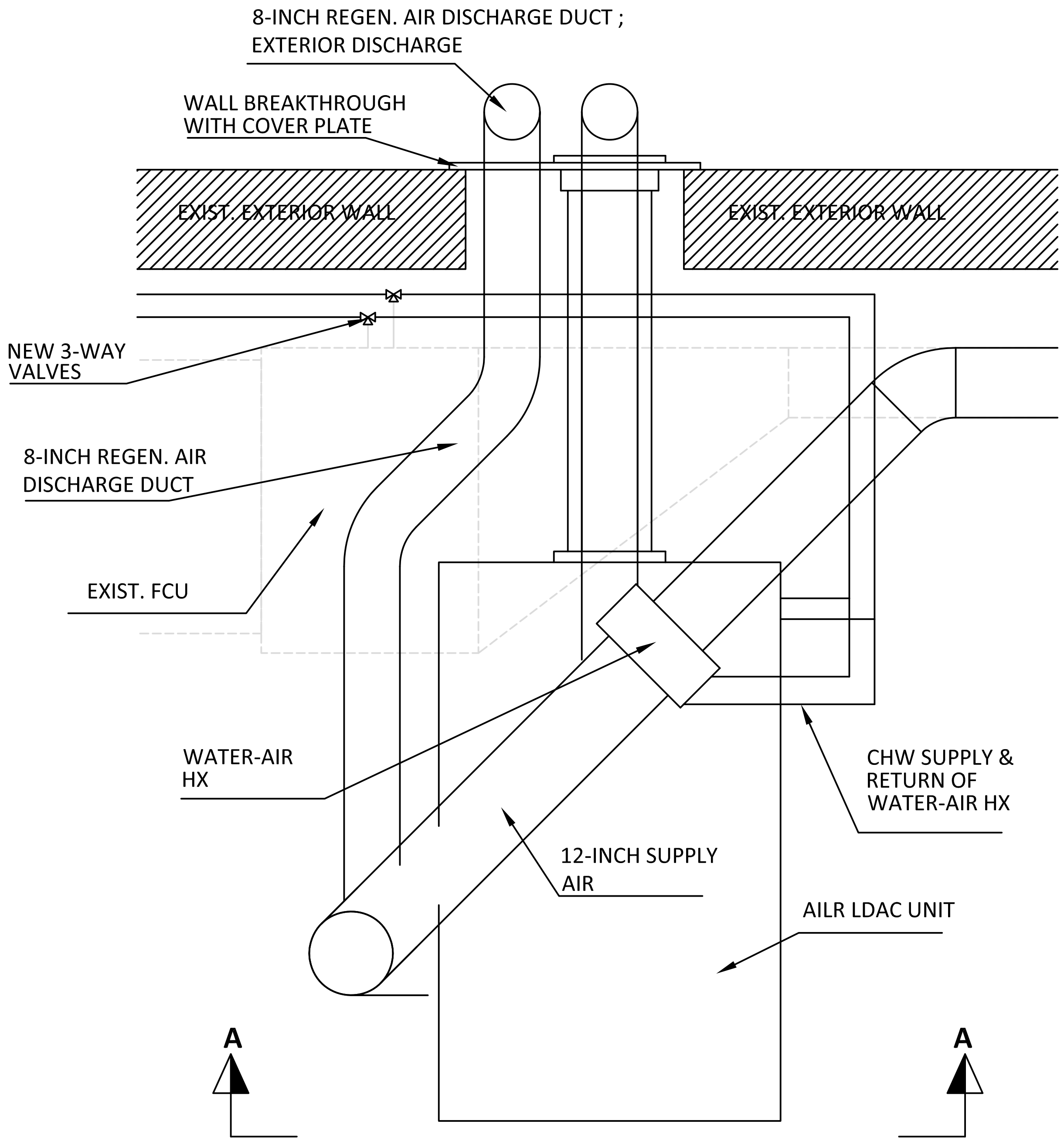
DATE
Oct.19, 2017

DRAWN BY
M. ZAPKA

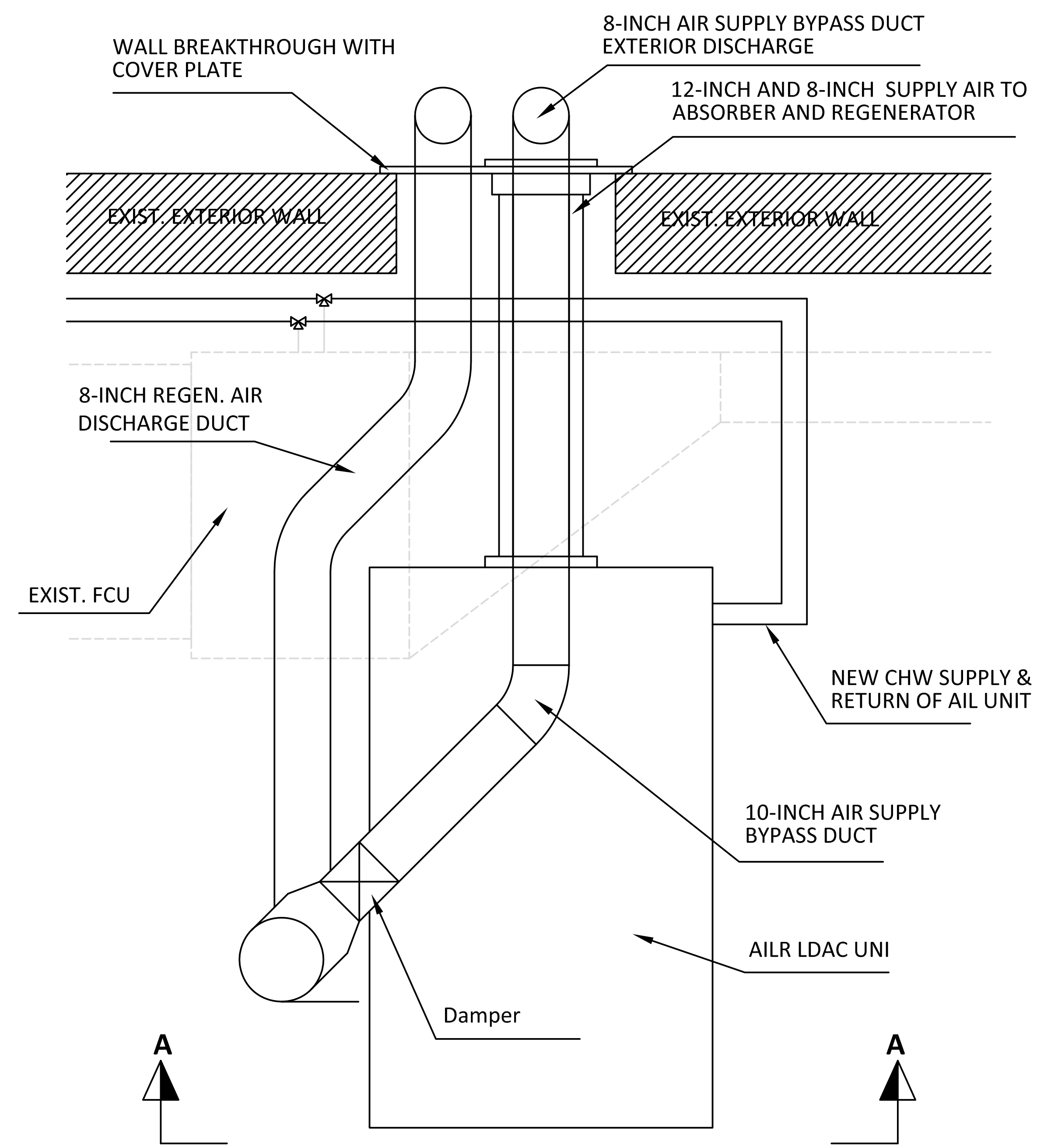
SCALE
1"= 6'

SHEET NO.

P-004



PLAN VIEW - DETAIL - WITH 12-INCH AIR SUPPLY DUCT



PLAN VIEW - DETAIL - WITHOUT 12-INCH AIR SUPPLY DUCT

0' 1' 2' 3' 4'

SCALE 1" = 2'

PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF AILR LDAC UNIT; DETAIL DEPICTION OF LAYOUT

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University of Hawai'i at Mānoa

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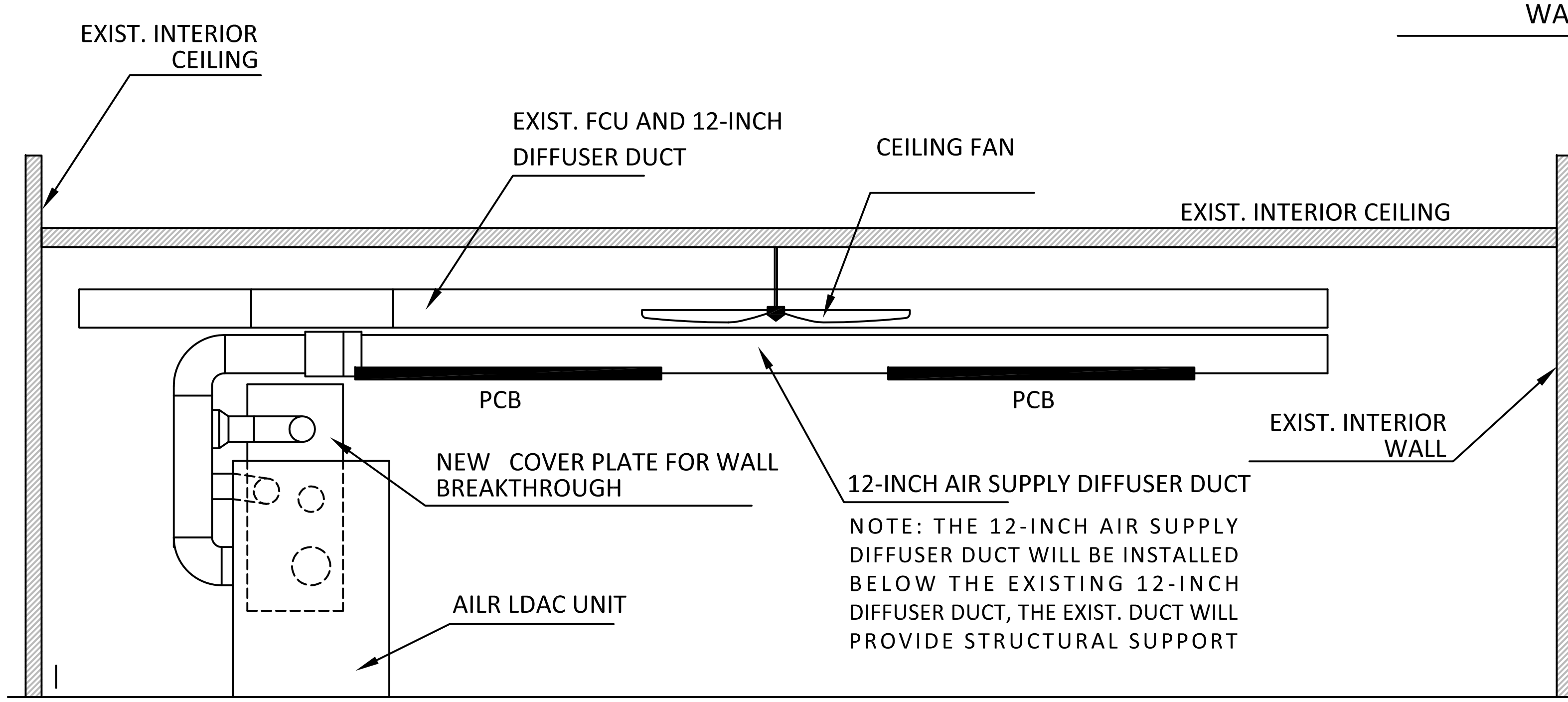
PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF AILR LDAC UNIT; DETAIL DEPICTION OF LAYOUT

PROJECT	HNEI – LD 2017
DATE	Oct.19, 2017
DRAWN BY	M. ZAPKA
SCALE	1"= 2'
SHEET NO.	**P-005**



SECTION A-A

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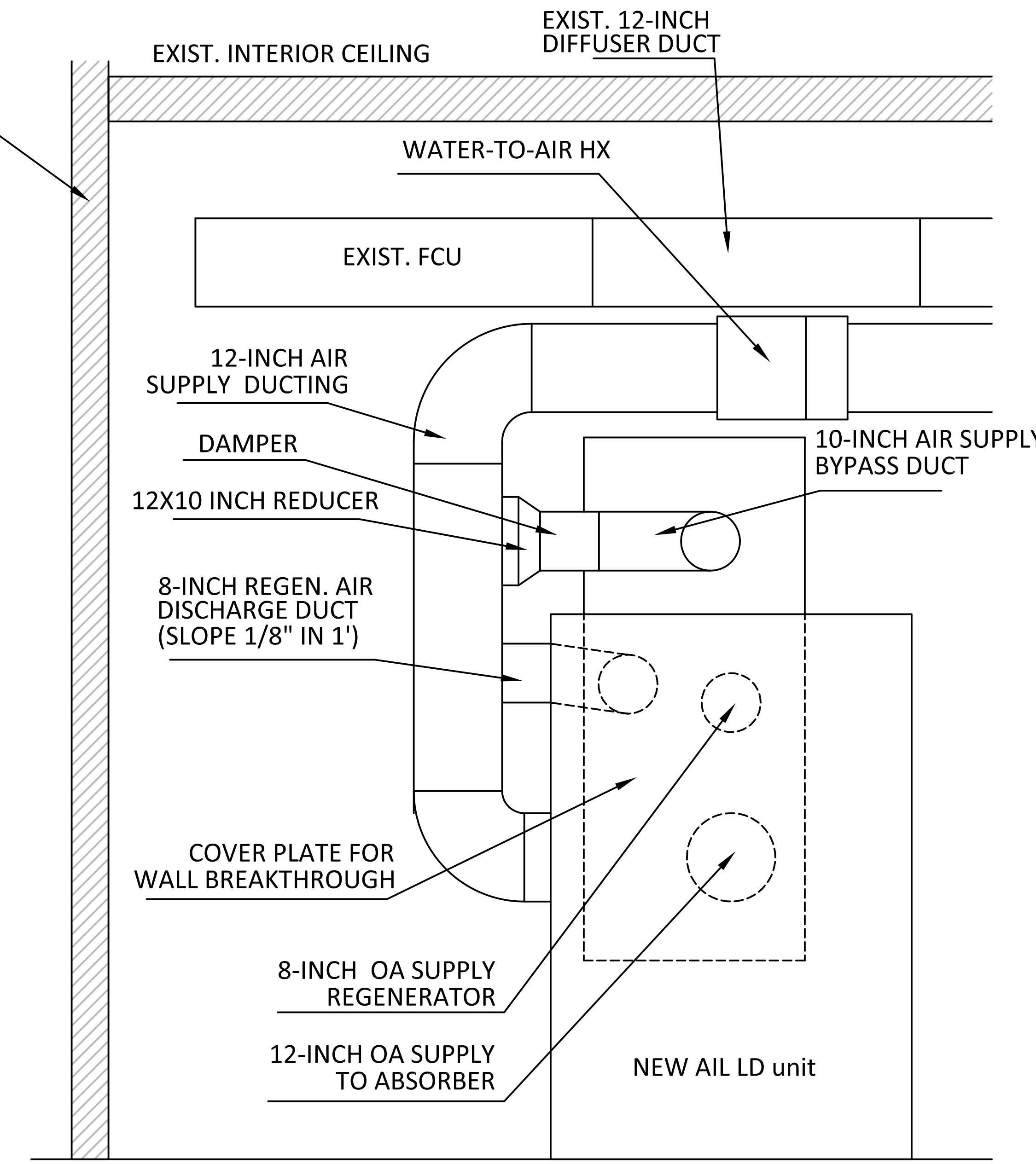
SCALE 1" = 5'



SECTION A-A

0' 1' 2' 3' 4' 5'

SCALE 1" = 1'

PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF AILR LDAC UNIT - SECTION A-A (OVERVIEW AND DETAILS)

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University of Hawai'i at Mānoa

Sustainable Design & Consulting LLC

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PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF AILR LDAC UNIT - SECTION A-A (OVERVIEW AND DETAILS)

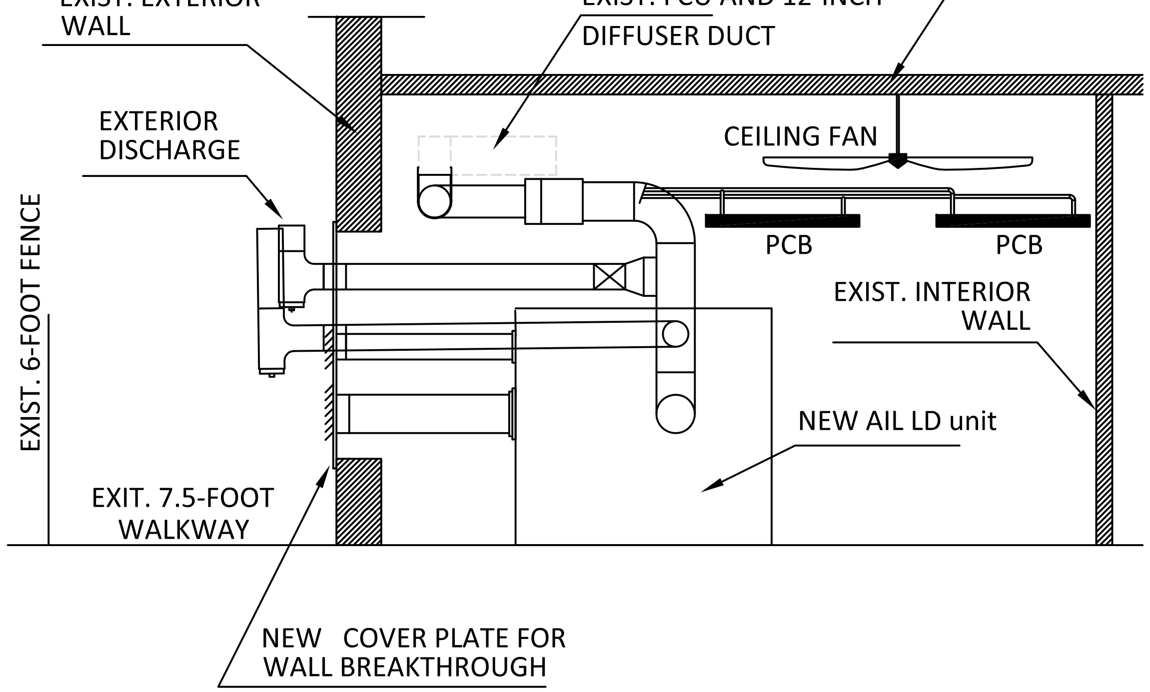
PROJECT
HNEI – LD 2017

DATE
Oct.19, 2017

DRAWN BY
M. ZAPKA

SCALE
VARIES

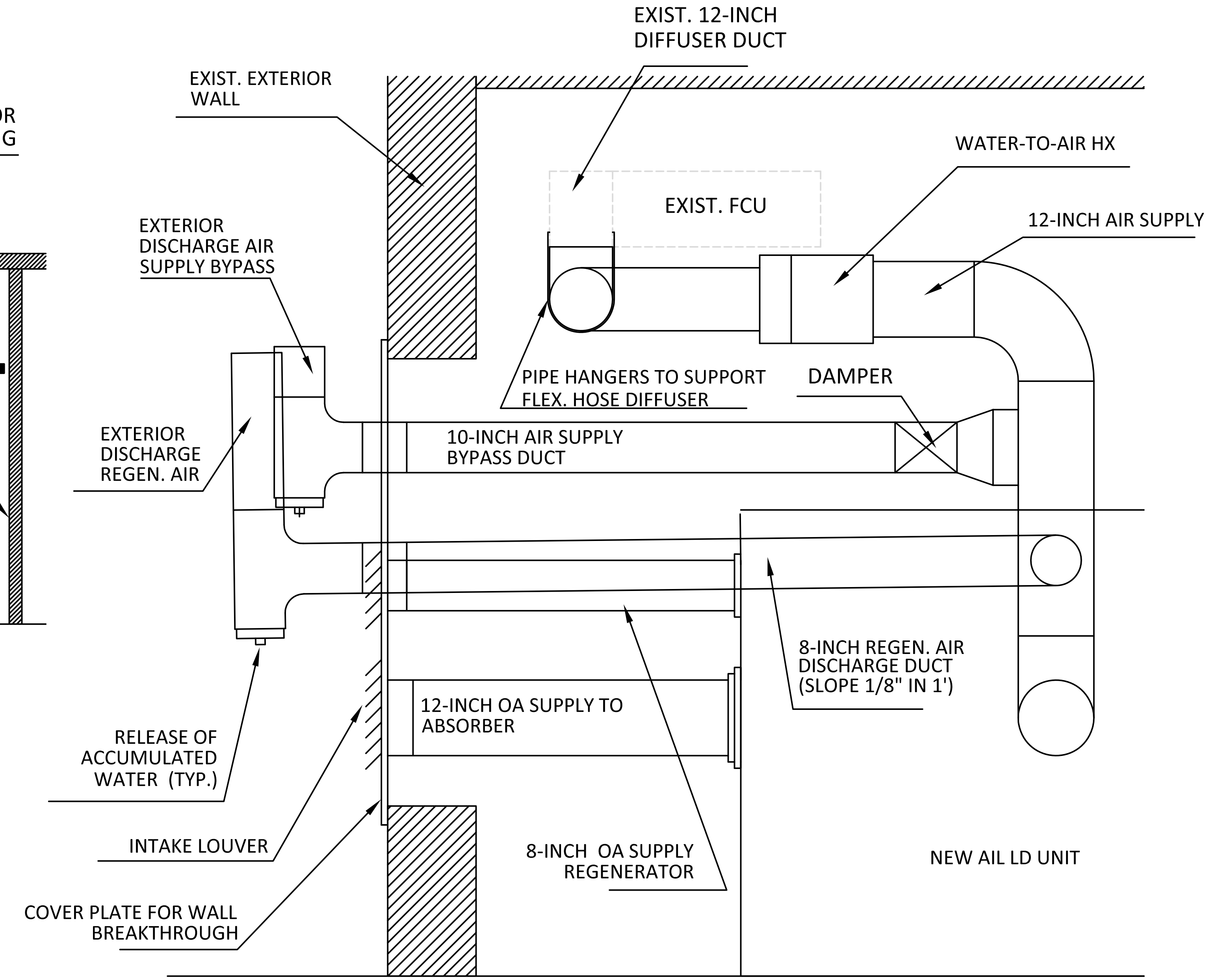
SHEET NO.
P-006



SECTION B-B

0' 2' 4' 6' 8' 10'

SCALE 1" = 6'



SECTION B-B - DETAILS

0' 1' 2' 3' 4'

SCALE 1" = 2'

PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF AILR LDAC UNIT - SECTION B-B (OVERVIEW AND DETAILS)

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University of Hawai'i at Mānoa

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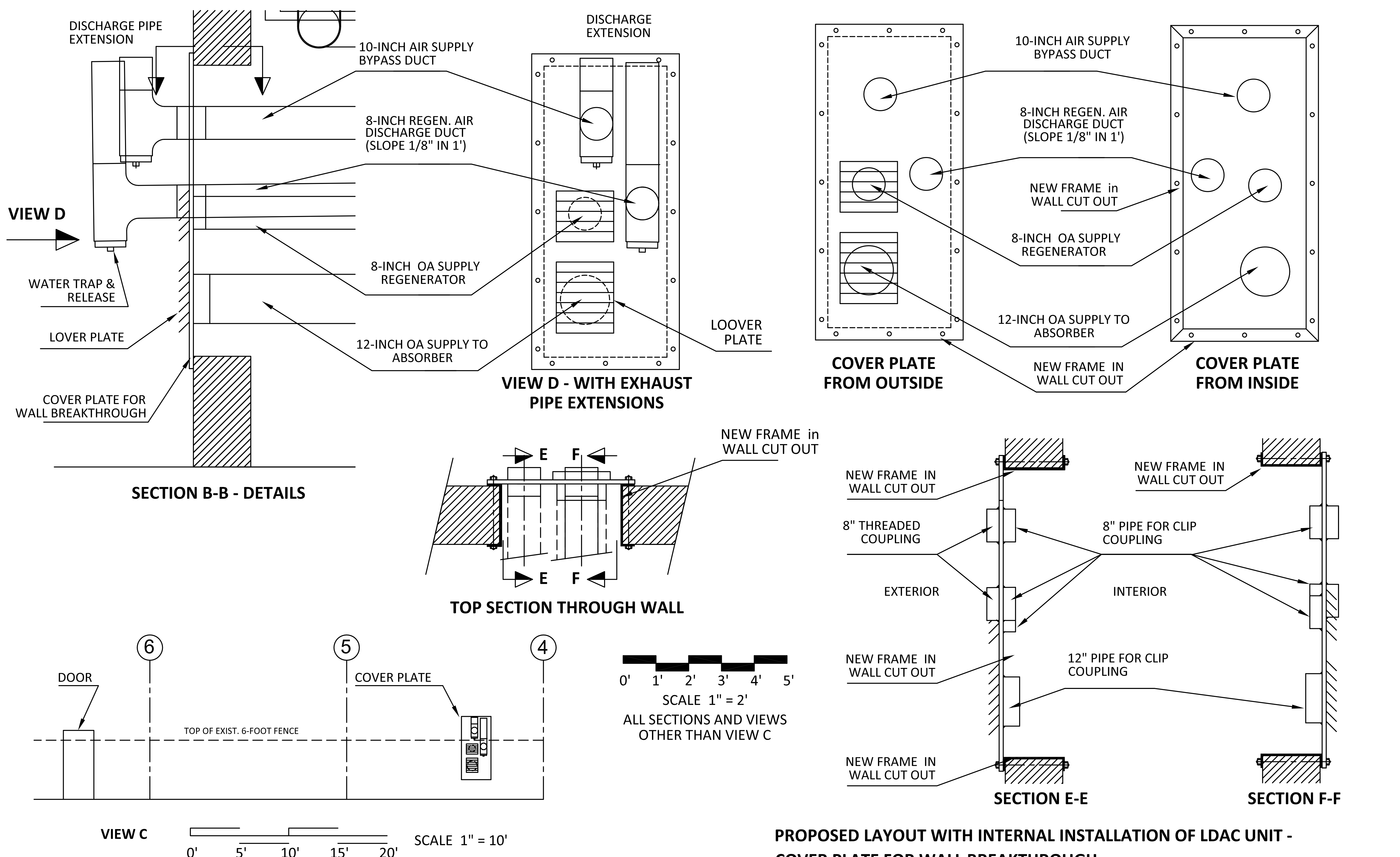
PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF AILR LDAC UNIT - SECTION B-B (OVERVIEW AND DETAILS)

PROJECT	HNEI – LD 2017
DATE	Oct.19, 2017
DRAWN BY	M. ZAPKA
SCALE	NO SCALE
SHEET NO.	**P-007**



PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF LDAC UNIT - COVER PLATE FOR WALL BREAKTHROUGH

HNEI Hawai'i Natural Energy Institute
University of Hawai'i at Mānoa

Sustainable Design & Consulting LLC

HNEI - LIQUIDE DESICCANT TEST FACILITY
PROPOSED LAYOUT WITH INTERNAL INSTALLATION OF LDAC UNIT - COVER PLATE FOR WALL BREAKTHROUGH

PROJECT
HNEI – LD 2017

DATE
Oct.19, 2017

DRAWN BY
M. ZAPKA

SCALE
VARIES

SHEET NO.
P-008

APPENDIX B.

Quotes received for major test equipment

Quotation

800 cfm Liquid Desiccant Air Conditioner

October 16, 2017

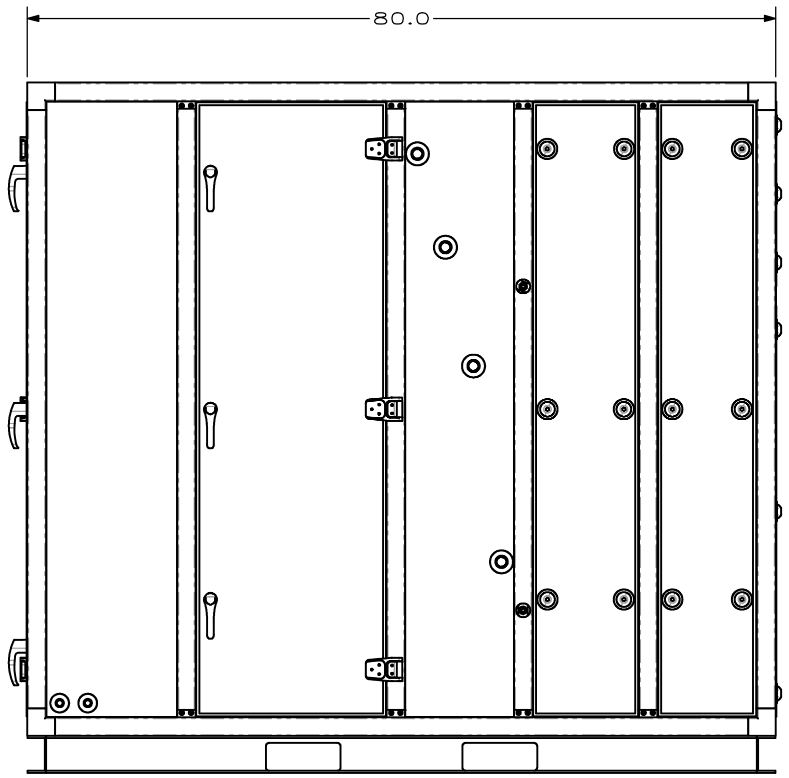
PREPARED BY:
AIL RESEARCH, INC.
57 HAMILTON AVENUE
HOPEWELL, NJ 08525

SUBMITTED TO:
SUSTAINABLE DESIGN & CONSULTING
P.O. BOX 25914
HONOLULU, HI 96825

Item No.	Description	Price
1	Pre-Piped, Skid-Mounted 800 cfm LDAC including:	
	wicking-fin conditioner	
	over capacity to 1,200 cfm	
	1/2" OD, 90/10 cupronickel tubes	
	corrugated fiberglass contact media	
	power-coated SS sidewalls	
	Plasticoated U-bends and solder joints	
	wicking-fin regenerator	
	1/2" OD, 70/30 cupronickel tubes	
	corrugated fiberglass contact media	
	power-coated SS sidewalls	
	PlastiDip U-bends and solder joints	
	modular framed enclosure; laminated panels	
	I-channel aluminum base	
	VFD controlled conditioner and regenerator fans	
	magnetically coupled centrifugal pumps (2)	
	welded polypropylene sumps	
	DirectLogic PLC controller	
	AcquiSuite data monitoring	
	pleated air filters for regeneration and conditioner	
	100 lb anhydrous lithium chloride	
	all electrical components UL listed	
		$98,218
2	Crating and Shipping	
	shipping to destination port in Hawaii	
	(Sustainable Design & Consulting responsible	
	for delivery to site and finally installation)	$6,133
3	Start-up/Field Support Labor	$14,109
4	Travel	
	one engineer and one technician; four days on site	$3,440
	Fee at 6%	$7,314
	Total Selling Price	$129,214

Payment Schedule

1) 30% due upon submission of order
2) 30% due upon completion of wicking-fin regenerator and conditioner
3) 30% due upon completion of LDAC; shop operation of all fans and pumps
4) 10% due upon receipt of LDAC at destination port in Hawaii



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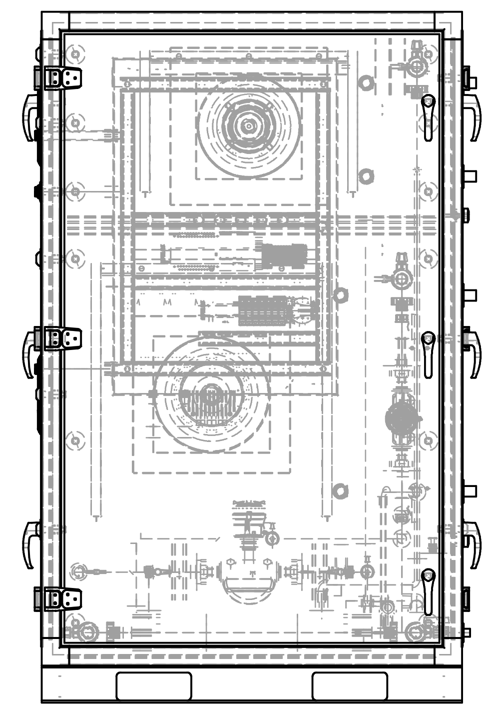

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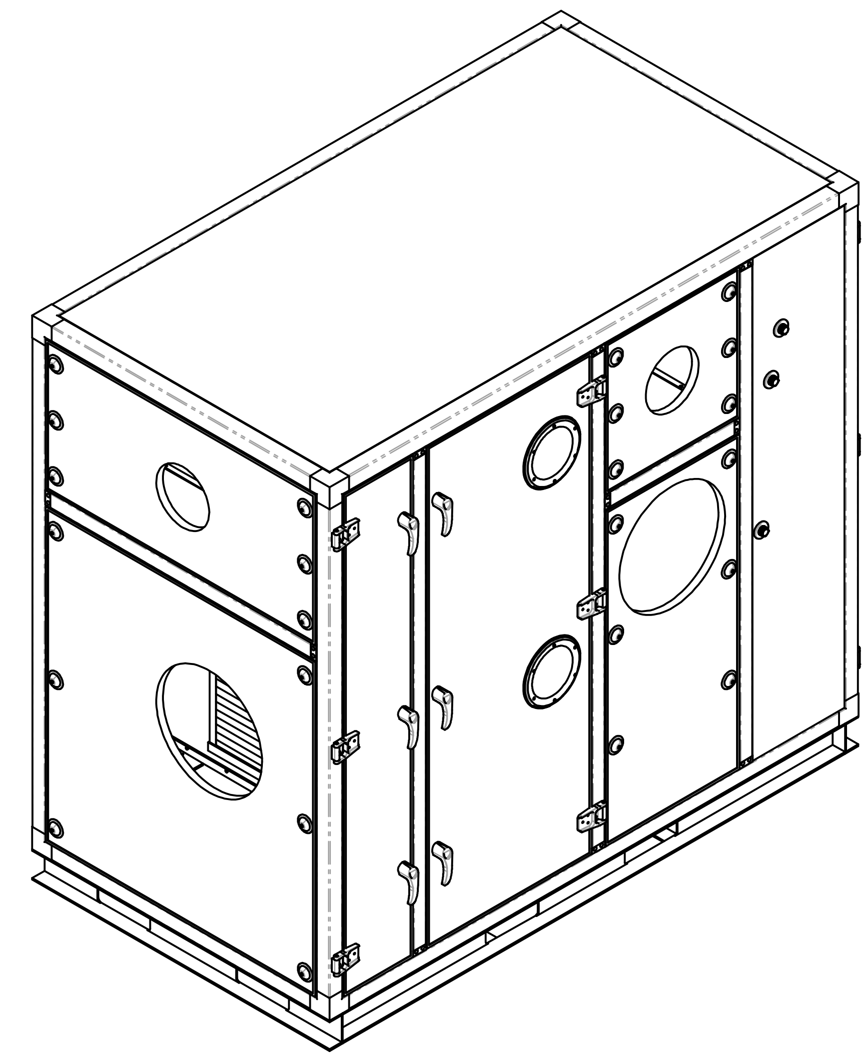

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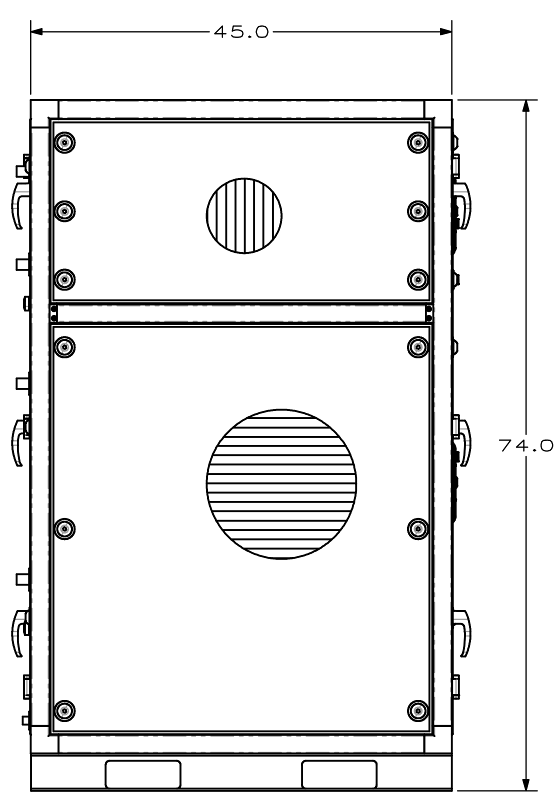

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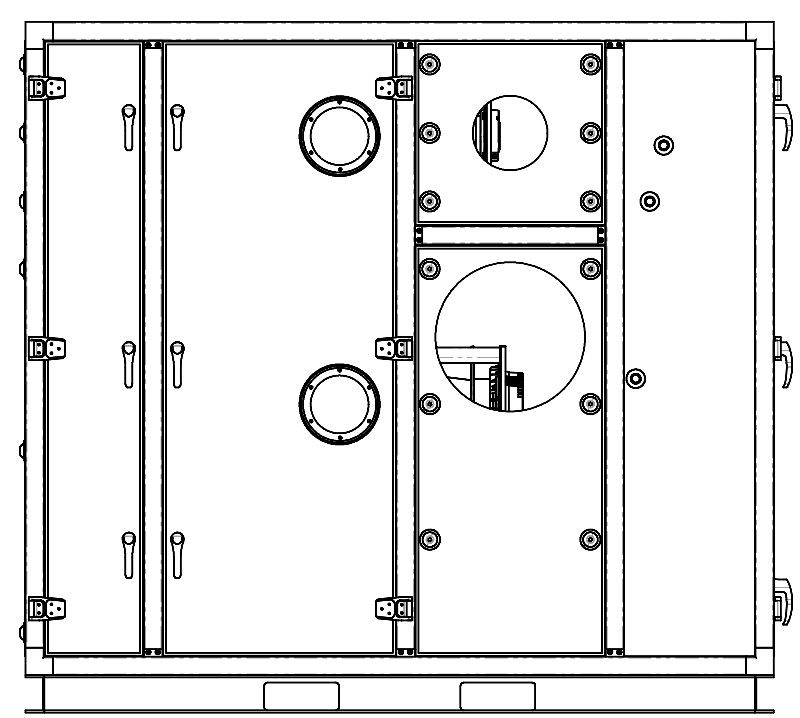

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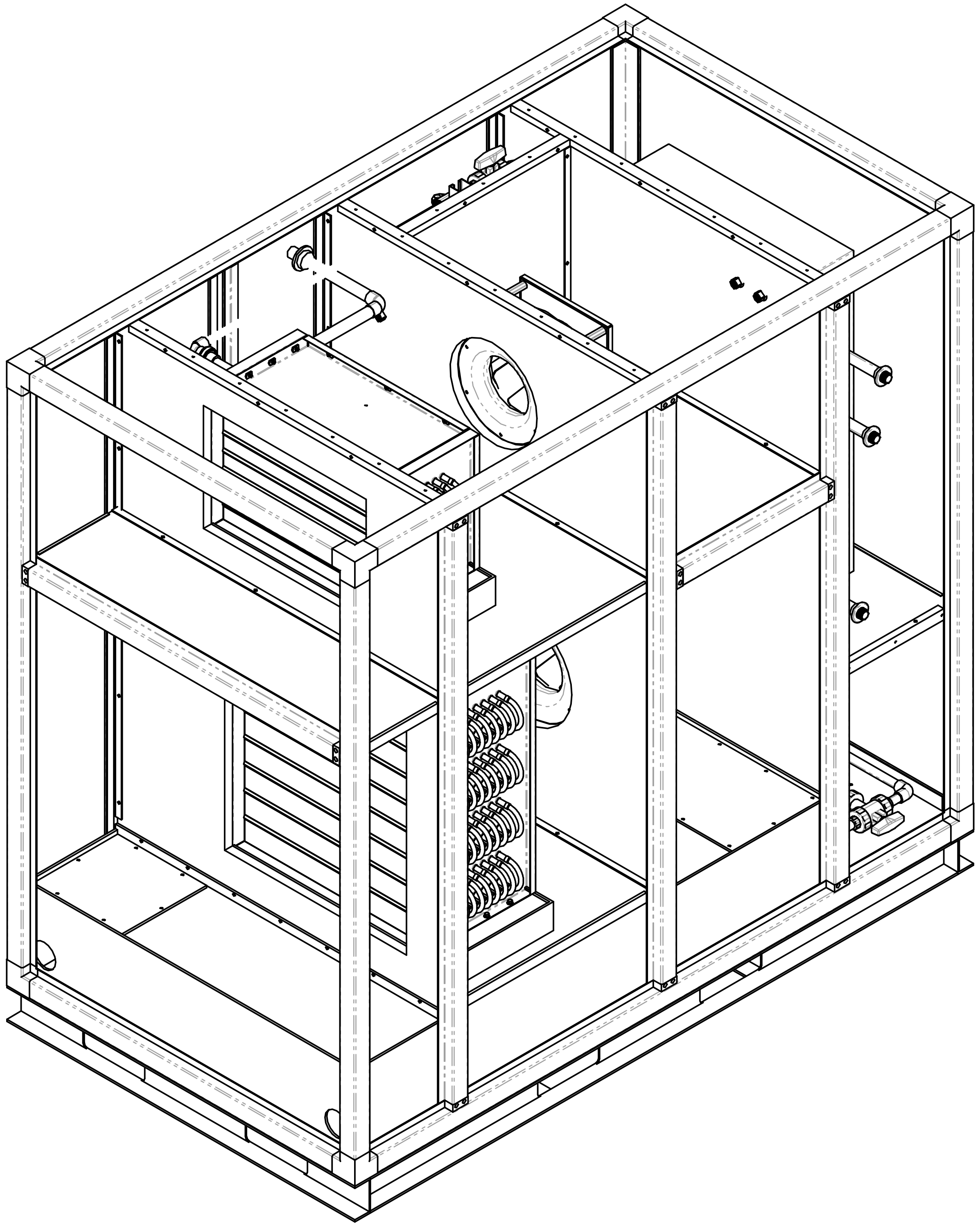
DESIGNED BY MILLER	DATE 10/13/17	AIL Research, Inc. 57 Hamilton Ave, Ste 205 Hopewell, NJ 08525 (609) 925-9002 x102	
DETAILED BY MILLER	DATE 10/13/17		
FILE U of H System.Z3			
UNLESS OTHERWISE NOTED, ALL DIMENSIONS ARE IN INCHES TOLERANCES EXCEPT AS NOTED:		TITLE UH LDAC	
DECIMAL .X ±.010 .XX ±.010 .XXX ±.005	FRACTIONAL ±1/32 ANGULAR ±1°	PROJECT University of Hawaii	
SCALE 1:10	1 OF 1	DWG UNCONTROLLED	REV



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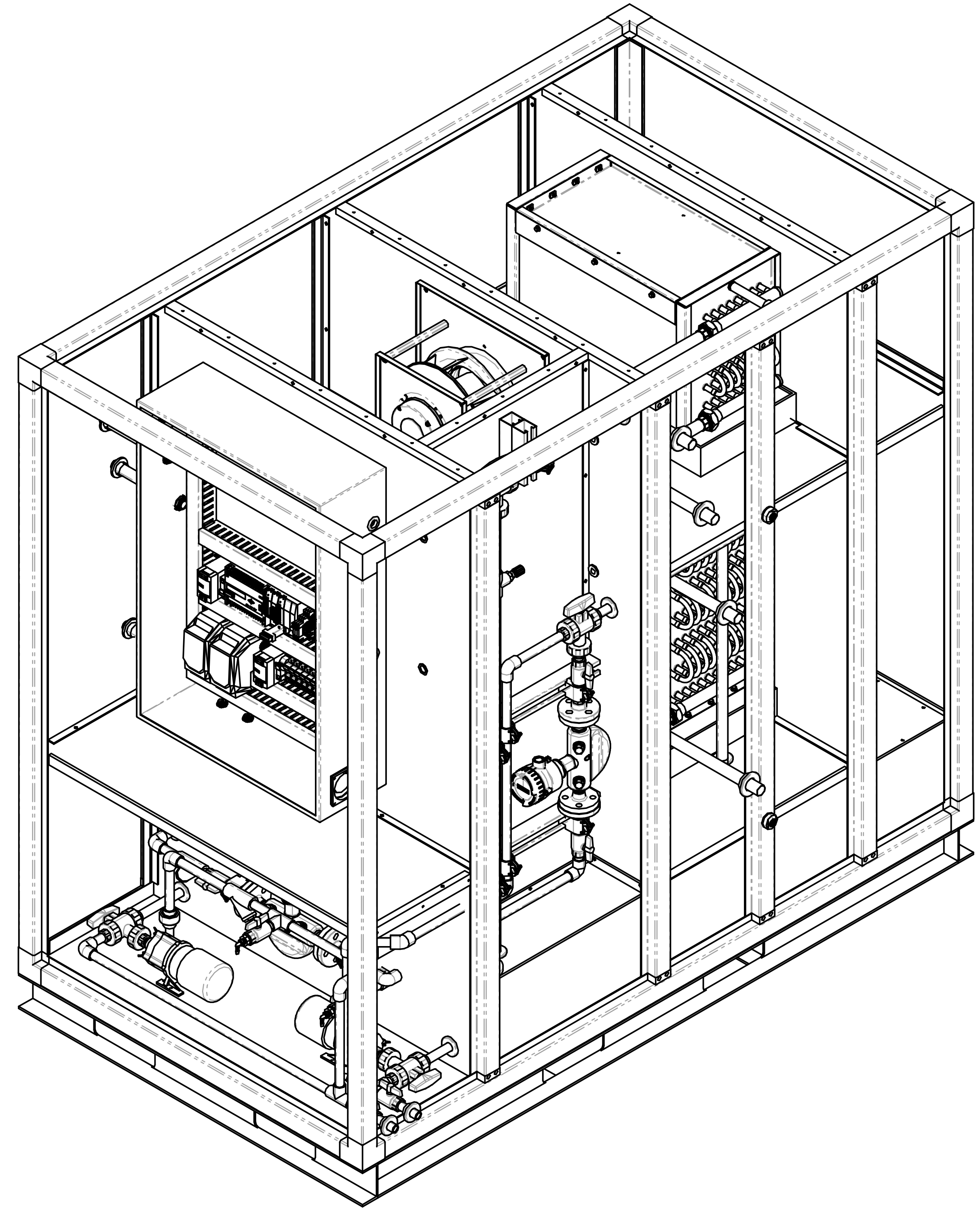

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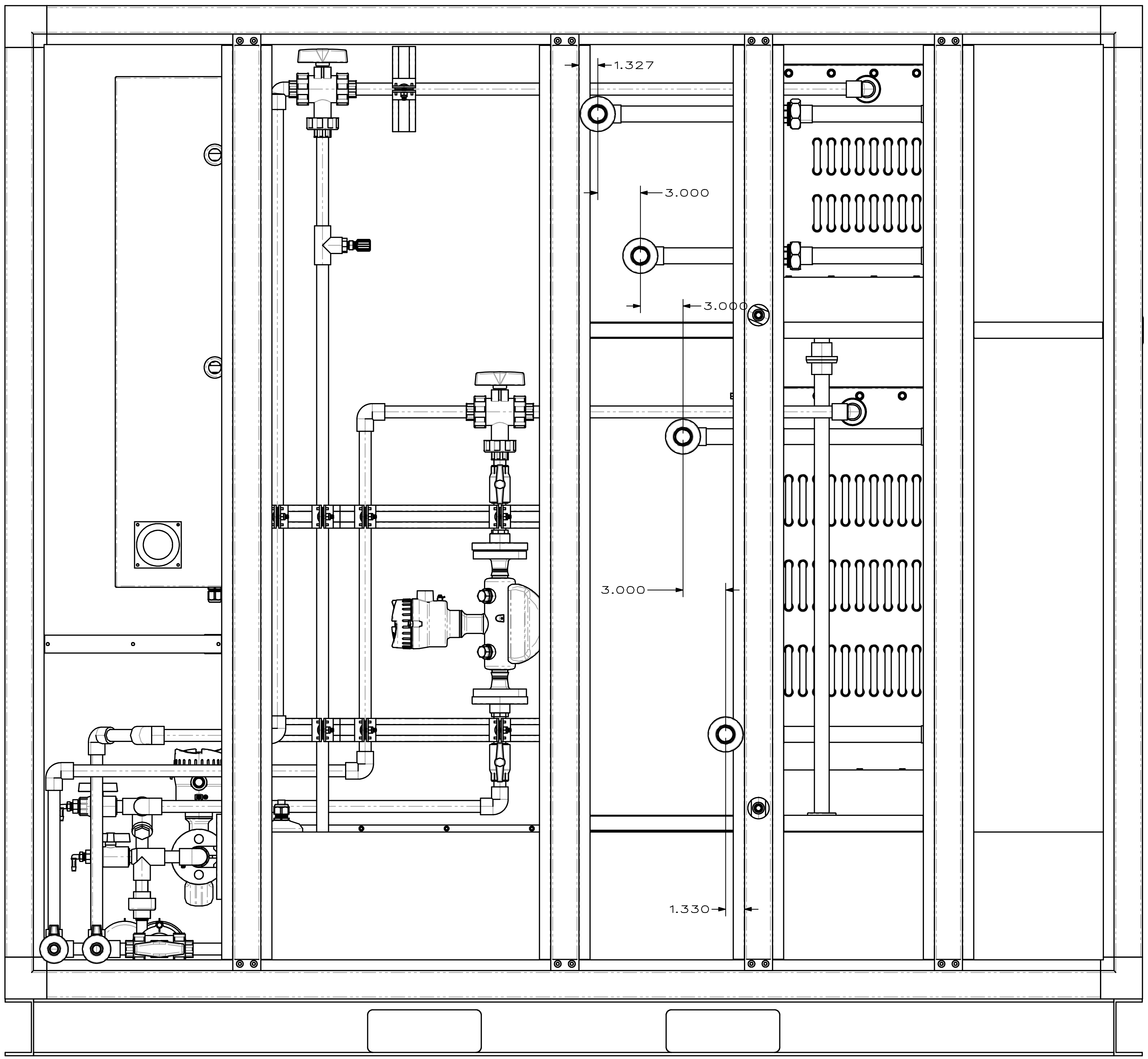
DETAILED BY MILLER DATE 10/13/17	
FILE U of H System.Z3	
UNLESS OTHERWISE NOTED, ALL DIMENSIONS ARE IN INCHES TOLERANCES EXCEPT AS NOTED:	TITLE UH LDAC
DECIMAL .X ±.010 .XX ±.010 .XXX ±.005 / FRACTIONAL ±1/32 / ANGULAR ±1°	PROJECT University of Hawaii
SCALE 1:7 / 1 OF 4	DWG UNCONTROLLED / REV



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DETAILED BY MILLER DATE 10/13/17		
FILE U of H System.Z3		
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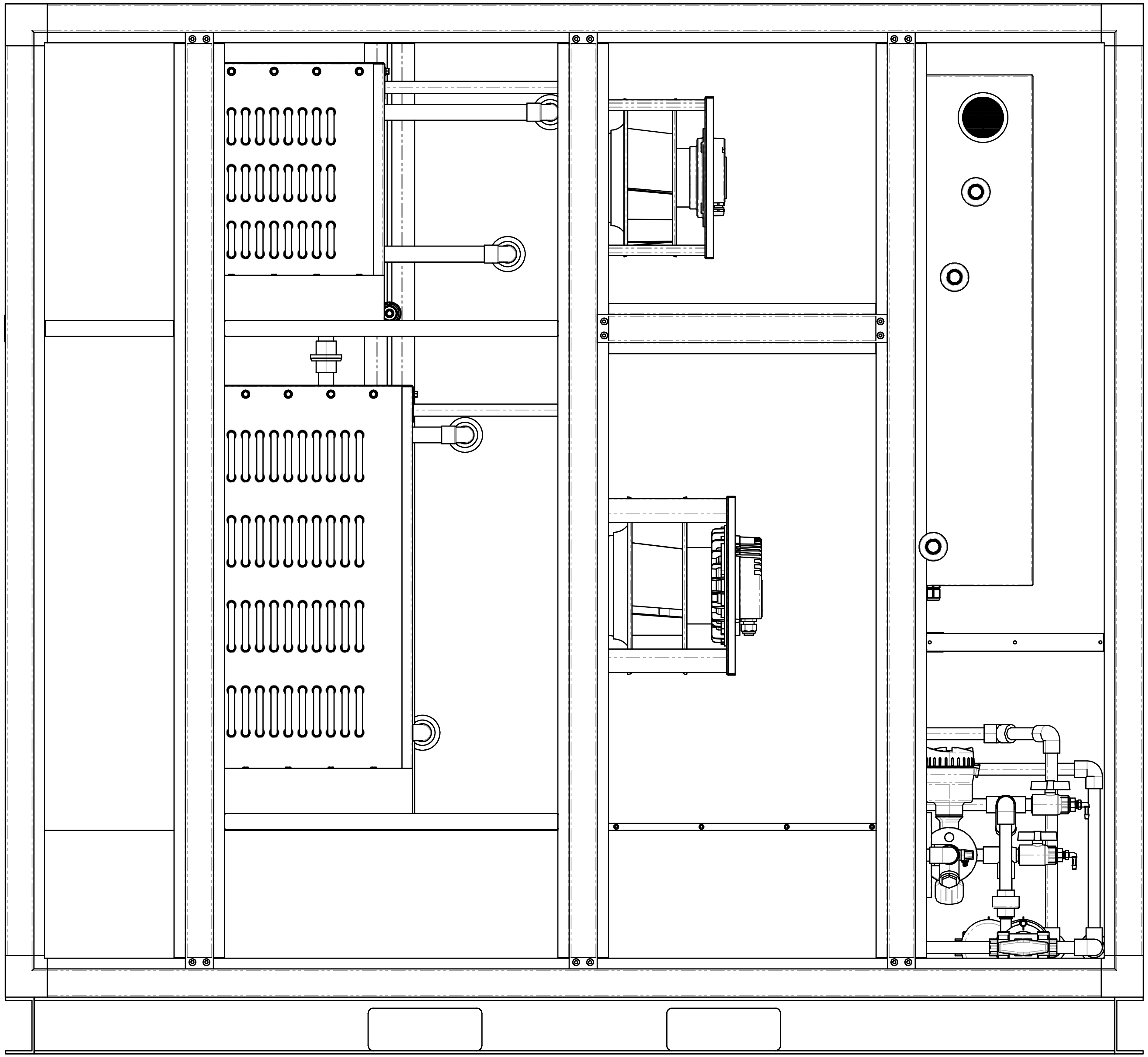
DECIMAL .X ±.010 .XX ±.010 .XXX ±.005	FRACTIONAL ±1/32 ANGULAR ±1°	PROJECT University of Hawaii
SCALE 1:4	2 OF 4	DWG UNCONTROLLED REV



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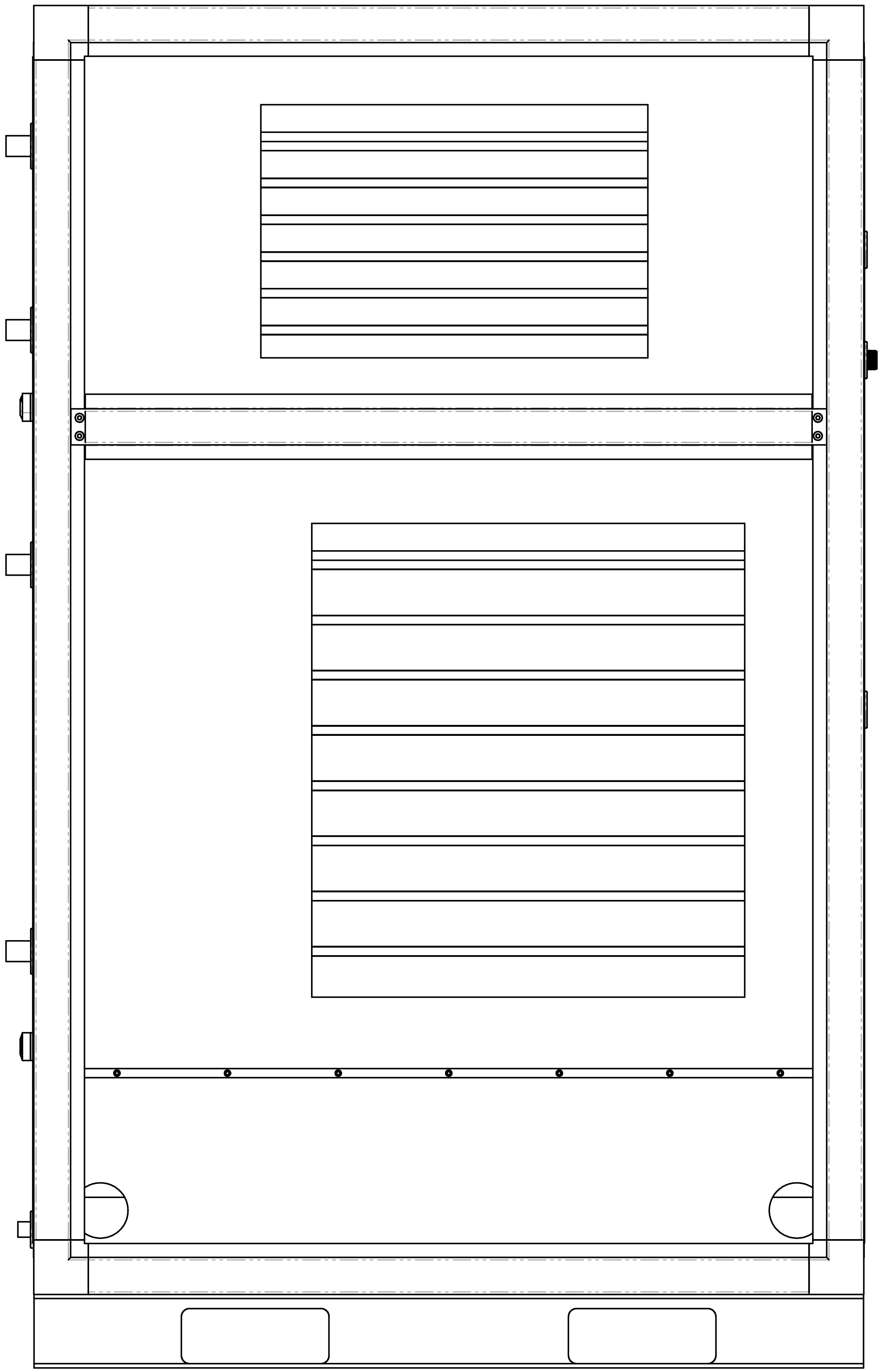
DETAILED BY MILLER	DATE 10/13/17		
FILE U of H System.Z3			
UNLESS OTHERWISE NOTED, ALL DIMENSIONS ARE IN INCHES TOLERANCES EXCEPT AS NOTED:		TITLE UH LDAC	
DECIMAL .X ±.010 .XX ±.010 .XXX ±.005	FRACTIONAL ±1/32 ANGULAR ±1°	PROJECT University of Hawaii	
SCALE 1:4	3 OF 4	DWG UNCONTROLLED	REV



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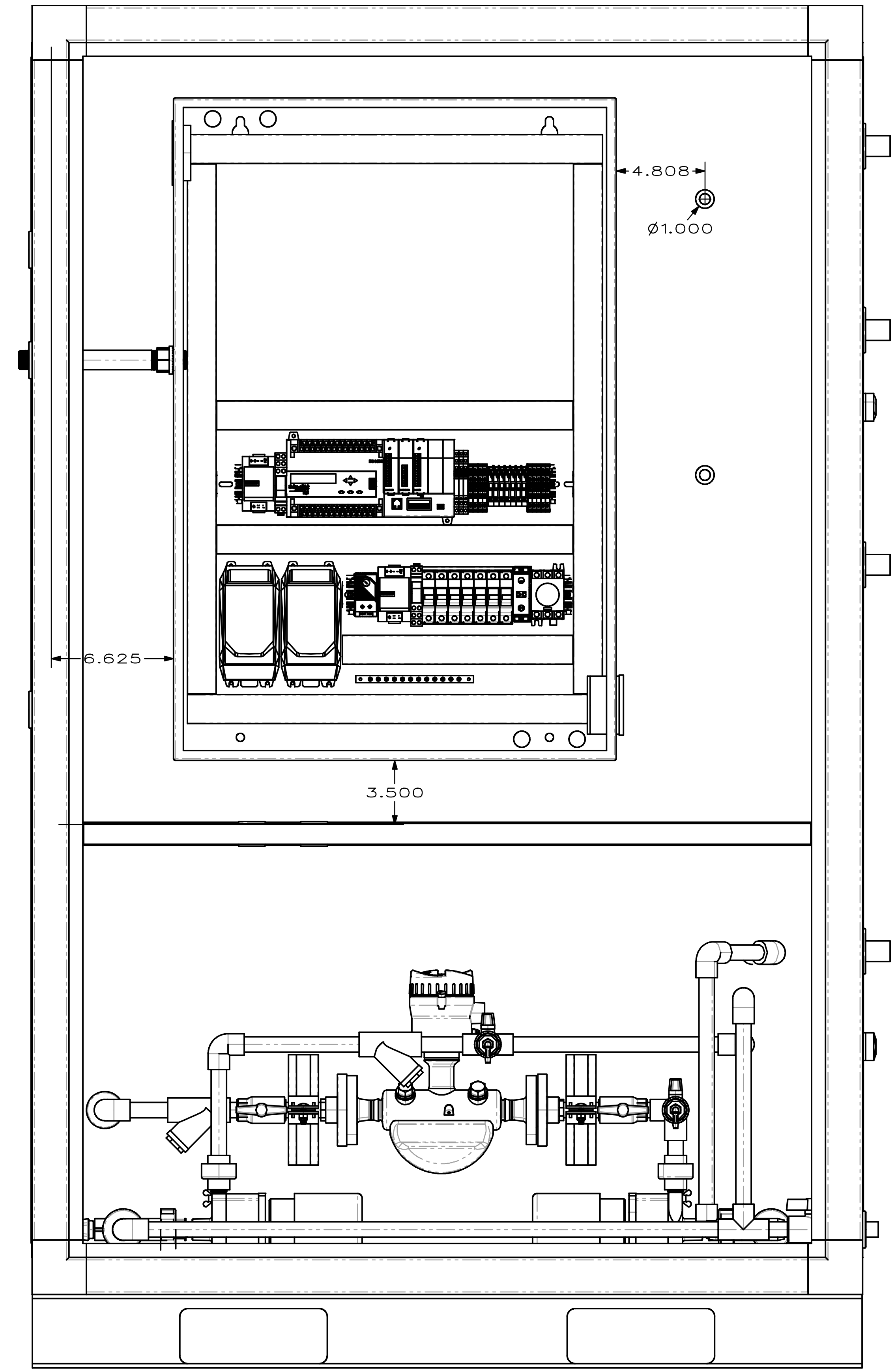

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DECIMAL .X ±.010 .XX ±.010 .XXX ±.005	FRACTIONAL ±1/32 ANGULAR ±1°	PROJECT University of Hawaii	
SCALE 1:4	4 OF 4	DWG UNCONTROLLED	REV

Barcol-Air Ltd.
115 Hurley Road
Oxford, CT 06478
P: (203) 262-9900
F: (203) 262-9906

Quote to: University of Hawaii
Your Contact: Manfred Zapka
Our Contact: Khalid Bu Khamsin

Project Name: Lab 123
Project No.: 00641

October 18, 2017
Quote-1

Radiant System by Barcol-Air Ltd.
System: BRW
Profiles: 14
Type: Full Slot
Tube: 15 mm OD copper tubing
Surface: Standard White

Unit USD	Chilled Ceiling Product	Length	Width	Units	Total USD
$ 1,200.00	BRW-FS-14-8	8'	4'-7/8"	4 pcs	$ 4,800.00
$ 650.00	Setup Activation			1 pcs	$ 650.00
$ 250.00	Setup Change Activation			0 pcs	$ -
SUBTOTAL Activation:				**4 pcs**	**$ 5,450.00**

Waves: -
-

Additional Components for Radiant Wave:

Barcol Stainless Steel Braided Hose, integrated oxygen barrier, push-fit couplings, C-clips

$ 25.00	Length 28 inch - 15 mm couplings on both ends	10 pcs	$ 250.00
$ 22.50	Length 18 inch - 15 mm couplings on both ends	0 pcs	$ -
	Subtotal Std Hose :	**10 pcs**	**$ 250.00**

Brass Nipples

$ 8.60	2 units per loop, ø15mm	10 pcs	$ 86.00
	Subtotal Nipples :		**$ 86.00**

Gripple Hangers for Radiant Wave Installation: 10ft cable. 1/4" stud (HF-SG-NO2-10FT)

16 pcs	4 wires per wave+Extras:	$ 7.14	**$ 114.24**

Clevis Pins with Hitch Pin Clips for Gripple Hanger Installation (1/4"x2")

16 pcs	4 pins per wave+Extras:	$ 1.00	**$ 16.00**

Barcol-Air Ltd.
115 Hurley Road
Oxford, CT 06478
P: (203) 262-9900
F: (203) 262-9906

Quote to:	University of Hawaii	Project Name: Lab 123	October 18, 2017
Your Contact:	Manfred Zapka	Project No.: 00641	Quote-1
Our Contact:	Khalid Bu Khamsin		

Summary Material Costs:	Sale Price	
Waves	$	5,450.00
Hoses	$	250.00
Nipples	$	86.00
Gripples	$	114.24
Suspension Bracket Pins	$	16.00
Subtotal Waves & Components	**$**	**5,916.24**

Additional Costs :		
Customs (approx. 2%)	$	5.00
0 pcs Transport / Handling Fees : Trucks	$	-
0 pcs Transport / Handling Fees : Packaging	$	-
0 pcs Transport / Handling Fees : Accessories	$	-
Subtotal	**$**	**5.00**
Total Project Costs :	**$**	**5,921.24**

Warranty : 1 year after commissioning, extension possible
Pricing valid for 3 months as of original quote.
Price of freight not included.
Counts are preliminary and are the subject of approval by the customer.
Please see also our terms and conditions.

Hubbell Electric Heater Company
45 Seymour Street
P.O. Box 288
Stratford CT 06615-0288

Phone: 203-378-2659
Fax: 203-378-3593
www.hubbellheaters.com

Quote Number: 46690

Attn: Manfred PROSP
To: University of Hawaii

Job Reference: Laboratory
EMail: mzapka@hawaii.edu
Phone: **Fax:**

Quote Date: 10/16/2017
Expires: 01/14/2018
FOB: FCA Stratford, CT
Ship Via: Best Way
Terms: To Be Decided
Sales Rep: House Account

Line	Model No. / Description	Quantity	Unit Price	Ext. Price
1	**V624T**	1	4,645.00	$4,645.00

Process Water Heater Heater
-6 Gallon Stainless Steel ASME Vessel
-24 kW, 240 volt, 3 phase

Packaged Hubbell electric heater with a 304 stainless steel vessel ASME Section VIII stamped Complete with all electrical operating controls including the following: magnetic contactor, immersion copper elements, digital temperature controller with safety hi-temperature cut out, lo water cut out, and an ASME rated combination temperature and pressure relief valve. Unit is factory assembled, insulated, jacketed, wired, and tested. Unit is ready for electrical and plumbing service connections.

Lead Time: 2 Weeks ARAD

All accepted purchase orders related to this quote shall be governed by the Seller's Terms and Conditions of Sale, viewable at www.hubbellheaters.com/termsofsale

Quote Total: **$4,645.00**

Quoted By: *Maggie Cumings*
mcumings@hubbellheaters.com
203-378-2659 ext.124

For financing options, go to:
www.hubbellheaters.com/lease

QUOTE # 1701109-Q

TAC Water & Misting Systems
18650 Collier Ave unit K
Lake Elsinore, CA 92530
(714) 336-8282

Quote Number: 1701109-Q
Quote Date: 11/9/17

ORDER DATE
PO#

Sold to:

HAWAII NATURAL ENERGY INSTITUTE
1680 EAST WEST ROAD POST 109
HONOLULU, HI 96822

Ship to:
SAME

ATTN; MANFRED ZAPKA

Entered By:
System Rep:
Account Cd: TACMISTING
Salesp TAC

Taxable:

Pmt Terms: COD

Order Qty	Item code	item	price	discount	price	extended price
1	ULTRAW100	ULTRA WALL COMPLETE SYSTEM	999each	795		795
1	DIFIL21	DI FILTER WITH FITTINGS	34	23		23
1	SHIP	USPS TBD				

Payments made to;

T.A. CAUDILL
18650 COLLIER AVE UNIT K
LAKE ELSINORE, CA 92530

TOTAL $ 818.00

APPENDIX C.

Laboratory safety information

UH MANOA-BASED OFF CAMPUS FACILITIES/LAB SAFETY INSPECTION CHECKLIST

Principal Investigator/Director:	Department:
Building / Lab Room #:	Date of Inspection:
Inspector(s):	Escorted By:

1	GENERAL SAFETY	Y	N	N/A	Inspector Comments	PI Comments/ Date Corrected
1.1	Is the laboratory locked when not in use?					
1.2	Are emergency eye washes and showers available, unobstructed and inspected monthly and quarterly, respectively? (required if corrosive materials are present).					
1.3	Are disposable containers for broken glass provided and specifically labeled for glass disposal ("Broken Glass")?					
1.4	Is protective clothing, goggles, face shields, gloves, closed-toe shoes and other PPE available and used?					
1.5	Are protective googles or face shields provided and worn where there is any danger of flying particles or corrosive materials?					
1.6	Have all chemical fume hoods passed inspection within the past 12 months?					
1.7	Are chemical fume hoods free from excessive storage?					
1.8	Are chemical fume hood sashes closed when not in use?					
1.9	Is good housekeeping maintained?					
1.10	Are all floors kept clean and dry and in good repair?					
1.11	Are food and beverages prepared and consumed in areas separate from chemicals?					
1.12	Are glass containers not stored on the floor?					
1.13	Are exits free of any trip hazards or obstruction? (minimum 28 inches clearance in any exit access such as hallways and aisles)					
1.14	Do refrigerators, freezers, microwaves, and ice machines designated for laboratory use have proper "No Food/Drink" signage?					
1.15	Are safety guards in place for equipment with moving parts (belts, blades, fans, etc)?					
1.16	Is there a first aid kit in the lab and is it adequately stocked?					
1.17	Employer has a written Respiratory Protection Program?					
1.18	Users are annually trained in the proper use of respirators and their limitations?					
1.19	Employees are fit tested to their respirators annually and are current in their medical clearance? Respirators are clean and maintained?					

2	CHEMICAL SAFETY	Y	N	N/A	Inspector Comments	PI Comments/ Date Corrected
2.1	Are all highly flammable and toxic procedures performed in a fume hood?					
2.2	Are approved spark-proof refrigerators used for cold storage of flammable liquids?					
2.3	Are flammable chemicals stored in a safe manner (more than 10 gallons stored in an approved flammable storage cabinet)?					
2.4	Are incompatible chemicals segregated in storage? (flammables and oxidizers; nitric acid/acids; acids and bases)					
2.5	Are all chemicals properly labeled, including hazard identification, and percentages of mixtures?					

2.6	Are chemical containers kept closed and in good condition?					
2.7	Are air and water reactive chemicals properly stored?					
2.8	Does the laboratory test peroxide-forming chemicals?					
2.9	Are chemical storage areas identified with signs (e.g., flammables, corrosives, carcinogens, poisons, etc.)?					
2.10	Are combustible scrap, debris and waste materials (i.e.oily rags) stored in covered metal receptacles and removed from the worksite promptly?					
2.11	Is a chemical spill kit available (with posted procedures)?					
2.12	Is metallic mercury used in the laboratory? If yes, is a Hg spill kit available?					

3	HAZARDOUS WASTE AUDIT CHECKLIST	Y	N	N/A	Inspector Comments	PI Comments/ Date Corrected
3.1	Is any hazardous waste generated in the facility or laboratory?					
3.2	Is non-hazardous chemical waste disposed of properly from the facility or laboratory?					
3.3	Does the facility generate less than 100 kg/month (220 lbs/27 G) of all hazardous waste and less than 1 kg/month (1 quart/2.2 lbs/0.27 G) of P-coded waste? (must not accumulate more than 1000 kg of hazardous waste at any time)					
3.4	Is the satellite accumulation area in the same laboratory where the waste is generated?					
3.5	Is the satellite accumulation area kept in good housekeeping condition?					
3.6	Are waste containers separated by hazard class to avoid incompatible storage?					
3.7	Are all the waste containers in good condition (e.g.,not corroded or leaking, and properly sealed or closed)?					
3.8	Are all waste containers properly labeled as to their contents (correct chemicals names, readable labels, and percentages of individual components for mixtures)?					
3.9	Are secondary containers used when required?					
3.10	Can the facility document the proper disposal of all hazardous waste?					
3.11	Is there at least one person in the facility who has attended the EHSO training for Hazardous Waste Generators?					

4	COMPRESSED GAS CYLINDERS	Y	N	N/A	Inspector Comments	PI Comments/ Date Corrected
4.1	Are cylinders legibly marked to clearly identify the gas contained?					
4.2	Are incompatible gases properly segregated when not in use? (i.e.oxygen and flammable gases must be separated by minimum 20 feet).					
4.3	Are cylinders secured properly (recommend chains) and protective caps in place when not in use?					
4.4	Are cylinders located or stored in areas where they will not be damaged by passing or falling objects or subject to tampering by unauthorized persons?					
4.5	Are oxygen cylinders stored 20 feet apart from combustible material or acetylene cylinders, or separated by an approved fire wall (at least 5 feet high) having a fire resistant rating of at least ½ hour?					

4.6	Multiple gas cylinders shall be securely stored in a cylinder rack and not by strap.					
4.7	Cylinders of different heights/sizes are not strapped together by one chain/strap.					
4.8	Cylinders have been hydrotested within the last 5 years to determine their integrity for current and further use.					
4.9	Cylinders are in good condition (no rusting, sidewall indentations, bulging, crack and fissures).					
4.10	Gas tubing used for gas cylinders are in good condition, show no leaks and are not pinched.					
4.11	Tygon tubing is not used for flammable gases (i.e. hydrogen) since it can cause static electricity.					

5	**ENVIRONMENTAL COMPLIANCE**	**Y**	**N**	**N/A**	**Inspector Comments**	**PI Comments/ Date Corrected**
5.1	Are best management practices (BMPs) in place to prevent illicit discharge and pollutants from entering the storm drains?					
5.2	Are storm drains regularly inspected and placarded?					
5.3	Oil-storage tanks (ASTs, day tanks, transformers) are regularly inspected for leaks and corrosion? (SPCC Plan required if oil storage exceeds 1320 gallons)					
5.4	Are drain disposal restrictions posted and followed according to the facility's Industrial Waste Water Discharge Permit?					
5.5	If underground storage tanks are in place, has the facility completed monthly monitoring requirements and proper DOH notification?					
5.6	If large capacity cesspools (>20 people per day or multi-unit housing) are in place, does the facility have a plan to shut down the cesspool?					
5.7	If a septic tank system is in place, does the facility maintain the system?					
5.8	Have all spills been cleaned up, documented and reported if necessary? Document below.					
5.9	Drip plans are used to control leaks from vehicles or equipment?					
5.10	Does the facility handle any pesticides meeting WSP requirements?					
5.11	Are employees trained on environmental compliance regulations?					
5.12	Does site have a current NPDES permit?					
5.13	Does site have a Industrial Wastewater Discharge Permit (IWDP) and follow proper drain disposal regulations?					

6	**FIRE SAFETY / ELECTRICAL SAFETY**	**Y**	**N**	**N/A**	**Inspector Comments**	**PI Comments/ Date Corrected**
6.1	Are the cords of all electrical equipment in good condition?					
6.2	Are cords used properly (e.g., no piggy-backing of surge protectors; clear of burners, sinks, aisles; no use of extension cords)					
6.3	Are electrical panel readily accessible and not blocked? (3 foot clearance in front & 30 inch working width clearance)					
6.4	Gasoline portable containers are approved metal safety cans with a spring-closing lid and spout cover?					
6.5	Are fire extinguisher(s), and fire pull stations readily accessible? Is the building fire alarm system tested annually?					
6.6	If you have outside private fire hydrants, are they flushed at least once a year and on a routine preventative maintenance schedule in accordance with the county water requirements?					

6.7	Are fire-rated doors not propped open?					
6.8	Are exits visibly marked and illuminated?					
6.9	Is storage at least 18 inches below the ceiling/sprinkler heads (24 inches for rooms without sprinklers)?					
6.10	Are equipment that draw large amounts of power (e.g. refrigerators, microwaves) plugged directly into an outlet?					
6.11	Equipment with exposed heating elements are unplugged when not in use (hot plates, coffee makers, toasters)?					
6.12	Does each electrical outlet, plug box, junction box, and cabinet have a faceplate, cover or canopy cover and are unused openings in cabinets and boxes effectively closed?					
6.13	When electrical equipment or lines are to be serviced, maintained or adjusted, are necessary switches opened, locked-out and tagged whenever possible?					
6.14	Combustible material is not stored in boiler, mechanical or electrical rooms.					
6.15	Does the facility have a written fire emergency plan? Have fire drills been conducted?					

7	FACILITIES / WORKSHOP SAFETY	Y	N	N/A	Inspector Comments	PI Comments/ Date Corrected
7.1	Are work surfaces elevated more than 4 feet above the floor or ground provided with standard guardrails? (top rail 42" high, w/ intermediate rail halfway between top rail and floor).					
7.2	Are all elevated surfaces (beneath which people or machinery could be exposed to falling objects) provided with standard toeboards?					
7.3	Is every floor hole or opening guarded by a standard railing or hole cover?(including skylights).					
7.4	Light bulbs/fluorescent tubes are protected by guards or enclosures to avoid breakage?					
7.5	Path to ground is permanent and continuous (ground wire and ground pin are in place and in good working order).					
7.6	Are only trained and authorized personnel permitted to operate powered industrial trucks or utility vehicles?					
7.7	Are pre-operation inspections performed on facility-owned powered industrial trucks before use? Are ones that are in need of repair or unsafe, taken out of service until they are restored to a safe operating condition?					
7.8	Is compressed air used for cleaning purposes reduced to less than 30 psi?					
7.9	Are all hand and power tools functioning and in safe condition?					
7.10	Are machine guards provided to protect the operator and other employees in the machine area from hazards such as those created by point of operation, ingoing nip points, rotating parts, flying chips, and sparks?					
7.11	Are portable circular saws equipped with guards above and below the base shoe?					
7.12	Are circular saw guards checked to assure they are not wedged up, thus leaving the lower portion of the blade unguarded?					
7.13	Are machines designed for a fixed location (e.g. drill presses, saws) securely anchored to prevent walking or moving?					
7.14	Are work rests on abrasive wheel machinery (grinders) kept closely adjusted to the wheel with a maximum opening of 1/8 inch? (Tongue guards within ¼ inch?)					
7.15	Are ladders adequately maintained and inspected?					

7.16	Are ladders appropriately used?					
7.17	Is employee exposure to welding fumes controlled by ventilation, use of respirators, exposure time, or other means?					
7.18	Does battery charging area provide general ventilation to control hydrogen gases emitted by lead-acid and nickel cadmium batteries?					

8	**CRANE OPERATION**	**Y**	**N**	**N/A**	**Inspector Comments**	**PI Comments/ Date Corrected**
8.1	Are cranes visually inspected for defective components prior to beginning of any work shift?					
8.2	Are all electrically operated cranes effectively grounded?					
8.3	Is a crane preventive maintenance program established?					
8.4	Is the load chart clearly visible to the operator?					
8.5	Are operating controls clearly identified?					
8.6	Is a fire extinguisher provided at the operator's station?					
8.7	Is the rated capacity visibly marked on each crane?					
8.8	Is an audible warning device mounted on each crane?					
8.9	Are cranes of such design, that the boom could fall over backwards, equipped with boomstops?					
8.10	Are crane inspection and maintenance records on file and available for inspection?					

	DOCUMENTATION	**Y**	**N**	**N/A**	**Inspector Comments**	**PI Comments/ Date Corrected**
	Is a current Chemical Hygiene Plan available in the laboratory?					
	Does the facility (non-laboratory) have a Hazard Communication Program Plan?					
	Does the facility have a current individual hazardous material and hazardous waste management program (HMMP) on file?					
	Are Standard Operating Procedures available for experiments, or equipment, that pose an increased hazard?					
	Does the lab have a written (annually updated) chemical inventory?					
	Are Safety Data Sheets (SDS) available for all chemicals in the lab (hardcopy or accessible online by all lab members)?					
	Have personnel received appropriate safety trainings? (Lab safety, Hazard Communication, Ladder Safety, Fire Safety, Respirator Training, etc). Are the training records on file?					
	Are emergency notification procedures, contacts with current phone numbers, and hazardous warning signs posted at the entry to the lab or facility?					
	Is the "HIOSH Workplace Poster" displayed in your workplace where all employees are likely to see it?					

ADDITIONAL INSPECTOR COMMENTS/ISSUES/PICTURES:

Once each identified problem has been rectified, state the date it was corrected and include any comments if you wish. The PI or Facility manager has 30 days upon receipt of this checklist to respond with corrective action(s). Please electronically sign and email the form to Carolyn Oki-Idouchi at the UH Manoa EHSO (coki@hawaii.edu). A follow-up inspection may be conducted to ensure corrections were made.

I certify that all rectifications required are complete, and to the best of my knowledge, true and accurate. By typing in my name I agree that it is equivalent to my handwritten signature.

Facility Manager/PI:		
	Type or Print Name	*Date*

Safety Data Sheet

Lithium Chloride

CAROLINA®
www.carolina.com

Section 1 — Product Description

Product Name: Lithium Chloride

Recommended Use: Science education applications
Synonyms: Hydrochloric acid, Lithium Salt
Distributor: Carolina Biological Supply Company
2700 York Road, Burlington, NC 27215
1-800-227-1150
Chemical Information: 800-227-1150 (8am-5pm (ET) M-F)
Chemtrec: 800-424-9300 (Transportation Spill Response 24 hours)

Section 2 — Hazard Identification

Classification of the chemical in accordance with paragraph (d) of §1910.1200;

WARNING

Harmful if swallowed. Causes skin irritation. Causes serious eye irritation. May cause respiratory irritation. Harmful to aquatic life.

GHS Classification:
Skin Corrosion/Irritation Category 2, Serious Eye Damage/Eye Irritation Category 2A, Specific Target Organ Systemic Toxicity (STOT) - Single Exposure Category 3, Hazardous to the aquatic environment - Acute Category 3, Acute Toxicity - Oral Category 4

Acute Toxicity Dermal Contains	100 % of the mixture consists of ingredient(s) of unknown toxicity
Acute Toxicity Inhalation Gas Contains	100 % of the mixture consists of ingredient(s) of unknown toxicity
Acute Toxicity Inhalation Vapor Contains	100 % of the mixture consists of ingredient(s) of unknown toxicity
Acute Toxicity Inhalation Dust/Mist Contains	100 % of the mixture consists of ingredient(s) of unknown toxicity

Section 3 — Composition / Information on Ingredients

Chemical Name	CAS #	%
Lithium Chloride	7447-41-8	100

Section 4 — First Aid Measures

Emergency and First Aid Procedures
Inhalation: IF INHALED: Remove victim to fresh air and keep at rest in a position comfortable for breathing.
Eyes: IF IN EYES: Rinse cautiously with water for several minutes. Remove contact lenses, if present and easy to do. Continue rinsing. If eye irritation persists: Get medical advice/attention.
Skin Contact: IF ON SKIN: Wash with plenty of soap and water. If skin irritation occurs: Get medical advice/attention. Take off contaminated clothing and wash before reuse.
Ingestion: IF SWALLOWED: Call a POISON CENTER or doctor/physician if you feel unwell.

Section 5 — Firefighting Procedures

Extinguishing Media: Use media suitable to extinguish surrounding fire.

Fire Fighting Methods and Protection: Firefighters should wear full protective equipment and NIOSH approved self-contained breathing apparatus.
Fire and/or Explosion Hazards: N/A
Hazardous Combustion Products: Hydrogen chloride, Lithium

Section 6 Spill or Leak Procedures

Steps to Take in Case Material Is Released or Spilled: Exposure to the spilled material may be irritating or harmful. Follow personal protective equipment recommendations found in Section 8 of this SDS. Additional precautions may be necessary based on special circumstances created by the spill including; the material spilled, the quantity of the spill, the area in which the spill occurred. Also consider the expertise of employees in the area responding to the spill. Avoid creating dusts. Cover material with absorbent and moisten and collect for disposal. Avoid breathing dust/fume/gas/mist/vapors/spray. Avoid contact with material.
Prevent the spread of any spill to minimize harm to human health and the environment if safe to do so. Wear complete and proper personal protective equipment following the recommendation of Section 8 at a minimum. Dike with suitable absorbent material like granulated clay. Gather and store in a sealed container pending a waste disposal evaluation. Ventilate the contaminated area. Isolate area. Keep unnecessary personnel away. Vacuum or sweep up material and place in a disposal container

Section 7 Handling and Storage

Handling: Avoid breathing dust/fume/gas/mist/vapors/spray. Wash thoroughly after handling. Do no eat, drink or smoke when using this product. Use only outdoors or in a well-ventilated area. Avoid release to the environment. Wear protective gloves/protective clothing/eye protection/face protection. Keep container tightly closed in a cool, well-ventilated place. Keep away from ... (incompatible materials to be indicated by the manufacturer). Readily absorbs moisture from air.
Storage: Store in a well-ventilated place. Keep container tightly closed. Store locked up. Suitable for any general chemical storage.
Storage Code: Green - general chemical storage

Section 8 Protection Information

	ACGIH		OSHA PEL	
Chemical Name	**(TWA)**	**(STEL)**	**(TWA)**	**(STEL)**
No data available	N/A	N/A	N/A	N/A

Control Parameters
Engineering Measures: No exposure limits exist for the constituents of this product. General room ventilation might be required to maintain operator comfort under normal conditions of use.
Personal Protective Equipment (PPE): Lab coat, apron, eye wash, safety shower.
Respiratory Protection: No respiratory protection required under normal conditions of use.
Eye Protection: Wear chemical splash goggles when handling this product. Have an eye wash station available.
Skin Protection: Avoid skin contact by wearing chemically resistant gloves, an apron and other protective equipment depending upon conditions of use. Inspect gloves for chemical break-through and replace at regular intervals. Clean protective equipment regularly. Wash hands and other exposed areas with mild soap and water before eating, drinking, and when leaving work.
Gloves: Nitrile, Natural rubber, Neoprene, PVC or equivalent.

Section 9 Physical Data

Formula: LiCl
Molecular Weight: 42.39
Appearance: White Crystals
Odor: None
Odor Threshold: No data available
pH: 6.0
Melting Point: No data available 608 C
Boiling Point: 1355 C
Flash Point: No data available
Flammable Limits in Air: N/A

Vapor Pressure: 1 mmHg at 547°C
Evaporation Rate (BuAc=1): N/A
Vapor Density (Air=1): N/A
Specific Gravity: 2.07
Solubility in Water: Soluble
Log Pow (calculated): -2.66
Autoignition Temperature: No data available
Decomposition Temperature: No data available
Viscosity: No data available
Percent Volatile by Volume: N/A

Section 10 Reactivity Data

Reactivity:	No data available
Chemical Stability:	Stable under normal conditions.
Conditions to Avoid:	Exposure to moisture
Incompatible Materials:	Acids, Halogens
Hazardous Decomposition Products:	HCl,
Hazardous Polymerization:	Will not occur

Section 11 Toxicity Data

Routes of Entry	Inhalation, ingestion, eye or skin contact.
Symptoms (Acute):	N/A
Delayed Effects:	No data available

Acute Toxicity:

Chemical Name	CAS Number	Oral LD50	Dermal LD50	Inhalation LC50
Lithium Chloride	7447-41-8	Oral LD50 Rat 1530 mg/kg Oral LD50 Mouse 1165 mg/kg	Not determined	Not determined

Carcinogenicity:

Chemical Name	CAS Number	IARC	NTP	OSHA
No data available	7447-41-8	Not listed	Not listed	Not listed

Chronic Effects:

Mutagenicity:	No evidence of a mutagenic effect.
Teratogenicity:	No evidence of a teratogenic effect (birth defect).
Sensitization:	No evidence of a sensitization effect.
Reproductive:	No evidence of negative reproductive effects.
Target Organ Effects:	
Acute:	See Section 2
Chronic:	N/A

Section 12 Ecological Data

Overview:	Moderate ecological hazard. This product may be dangerous to plants and/or wildlife. Keep out of waterways.
Mobility:	No data
Persistence:	No data
Bioaccumulation:	No data
Degradability:	No data
Other Adverse Effects:	No data

Chemical Name	CAS Number	Eco Toxicity
N/A	7447-41-8	

Section 13 Disposal Information

Disposal Methods:	Dispose in accordance with all applicable Federal, State and Local regulations. Always contact a permitted waste disposer (TSD) to assure compliance.
Waste Disposal Code(s):	Not Determined

Section 14 Transport Information

Ground - DOT Proper Shipping Name:
N/A

Air - IATA Proper Shipping Name:
Not regulated for air transport by IATA.

Section 15 Regulatory Information

TSCA Status: All components in this product are on the TSCA Inventory.

Chemical Name	CAS Number	§ 313 Name	§ 304 RQ	CERCLA RQ	§ 302 TPQ	CAA 112(2) TQ
No data available	7447-41-8	No	No	No	No	No

Section 16 Additional Information

Revised: 09/09/2015 **Replaces: 09/03/2014** **Printed: 10-29-2015**

The information provided in this (Material) Safety Data Sheet represents a compilation of data drawn directly from various sources available to us. Carolina Biological Supply makes no representation or guarantee as to the suitability of this information to a particular application of the substance covered in the (Material) Safety Data Sheet.

Glossary

ACGIH	American Conference of Governmental Industrial Hygienists
CAS	Chemical Abstract Service Number
CERCLA	Comprehensive Environmental Response, Compensation, and Liability Act
DOT	U.S. Department of Transportation
IARC	International Agency for Research on Cancer
N/A	Not Available
NTP	National Toxicology Program
OSHA	Occupational Safety and Health Administration
PEL	Permissible Exposure Limit
ppm	Parts per million
RCRA	Resource Conservation and Recovery Act
SARA	Superfund Amendments and Reauthorization Act
TLV	Threshold Limit Value
TSCA	Toxic Substances Control Act
IDLH	Immediately dangerous to life and health

Material Safety Data Sheet

Catalog Number: 150134
Revision date: 26-Apr-2006

1. IDENTIFICATION OF THE SUBSTANCE/PREPARATION AND COMPANY INFORMATION

Catalog Number: 150134

Product name: LITHIUM CHLORIDE ANHYDROUS

Synonyms: Chlorku litu; Chlorure de lithium; Lithiumchlorid
Supplier:
MP Biomedicals, LLC
29525 Fountain Parkway
Solon, OH 44139
tel: 440-337-1200

Emergency telephone number: CHEMTREC: 1-800-424-9300 (1-703-527-3887)

2. COMPOSITION/INFORMATION ON INGREDIENTS

Components	CAS Number	Weight %	ACGIH Exposure Limits:	OSHA Exposure Limits:
LITHIUM CHLORIDE ANHYDROUS	7447-41-8	90 - 100%	None	None

3. HAZARDS IDENTIFICATION

EMERGENCY OVERVIEW: The product causes burns of eyes, skin and mucous membranes, Harmful if swallowed.

Category of Danger:
Corrosive , Harmful , Lachrymator

Principle routes of exposure: Skin
Inhalation: Vapors or dusts will cause burns of respiratory passages.
May be harmful if inhaled.
Ingestion: Can burn mouth, throat, and stomach
Harmful if swallowed.
Skin contact: Causes skin burns
May be harmful if absorbed through the skin.
Eye contact: Causes eye burns
Vapors extremely irritating to eyes an respiratory tract

ANSI Classification Corrosive, Irritant - eye, severe
Statements of hazard CAUSES BURNS TO SKIN AND EYES
VAPORS OR MISTS CAN IRRITATE OR BURN THE RESPIRATORY TRACT.
HARMFUL IF SWALLOWED. MAY BE HARMFUL IF ABSORBED THROUGH SKIN OR INHALED.
CAUSES EYE IRRITATION.
Statement of Spill or Leak - ANSI Label Contain and/or absorb spill with inert material (e.g. sand, vermiculite), then place in suitable container. Do not flush to sewer or allow to enter waterways. Use appropriate Personal Protective Equipment.

Statement of First Aid If swallowed, do NOT induce vomiting unless directed to do so by medical personnel. Never give anything by mouth to an unconscious person. Call a physician. In case of contact, immediately flush eyes or skin with plenty of water for at least 15 minutes while removing contaminated clothing and shoes. Wash clothing before reuse. Call a physician. If inhaled, remove to fresh air. If not breathing, give artificial respiration. If breathing is difficult, give oxygen. Call a physician. If swallowed, DO NOT induce vomiting. Get medical attention immediately. In case of contact, flush eyes with running water for at least 15 minutes. Consult a physician for irritation or any other symptom.
Precautions - ANSI Label Do not taste or swallow. Wash thoroughly after handling. Do not swallow. Avoid contact with skin and eyes Do not breathe vapors or spray mist

4. FIRST AID MEASURES

General advice: In the case of accident or if you feel unwell, seek medical advice immediately (show the label where possible).
Inhalation: Call a physician immediately Move to fresh air Consult a physician
Skin contact: Wash off immediately with plenty of water for at least 15 minutes Call a physician immediately
Ingestion: Do not induce vomiting without medical advice. Never give anything by mouth to an unconscious person. Consult a physician Do not induce vomiting. Call a physician immediately.
Eye contact: Rinse immediately with plenty of water, also under the eyelids, for at least 15 minutes Call a physician immediately Rinse immediately with plenty of water, also under the eyelids, for at least 15 minutes.
Protection of first-aiders: No information available
Medical conditions aggravated by exposure: None known

5. FIRE FIGHTING MEASURES

Suitable extinguishing media:	Dry sand (except on molten lithium)., Dry graphite, Dry lithium chloride, Zirconium silicate, Lith-X (graphite based) media
Specific hazards:	Flammable solid. Evolves hydrogen and ignites on contact with water. Combustion may produce irritants and toxic gases.
Unusual hazards:	None known
Special protective equipment for firefighters:	As in any fire, wear self-contained breathing apparatus pressure-demand, MSHA/NIOSH (approved or equivalent) and full protective gear
Specific methods:	Water mist may be used to cool closed containers.
Flash point:	Not determined
Autoignition temperature:	Not determined

NFPA rating:

NFPA Health:	2
NFPA Flammability:	1
NFPA Reactivity:	1

6. ACCIDENTAL RELEASE MEASURES

Personal precautions:	Use personal protective equipment.
Environmental precautions:	Do not flush into surface water or sanitary sewer system.
Methods for cleaning up:	Soak up with inert absorbent material.

7. HANDLING AND STORAGE

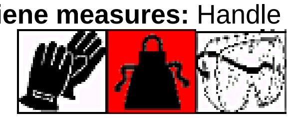
Storage:

ROOM TEMPERATURE
DESICCATE

Handling:	Use only in area provided with appropriate exhaust ventilation.
Safe handling advice:	Wear personal protective equipment.
Technical measures/storage conditions:	Keep containers tightly closed in a cool, well-ventilated place.
Incompatible products:	Oxidising and spontaneously flammable products

8. EXPOSURE CONTROLS / PERSONAL PROTECTION

Engineering measures: Ensure adequate ventilation.
PERSONAL PROTECTIVE EQUIPMENT
Respiratory protection: Breathing apparatus only if aerosol or dust is formed.
Hand protection: Pvc or other plastic material gloves
Skin and body protection: Usual safety precautions while handling the product will provide adequate protection against this potential effect.
Eye protection: Safety glasses with side-shields
Hygiene measures: Handle in accordance with good industrial hygiene and safety practice.



9. PHYSICAL AND CHEMICAL PROPERTIES

Appearance and Odor	White
Physical state:	Powder
Formula:	LiCl
Molecular weight:	42.4
Melting point/range:	605ºC
Boiling point/range:	1355°C
Density:	2.1 g/ml
Vapor pressure:	0
Evaporation rate:	No data available
Vapor density:	No data available
Solubility (in water):	Soluble
Flash point:	Not determined
Partition coefficient (n-octanol/water):	Log P(oct) = -2.66
Autoignition temperature:	Not determined

10. STABILITY AND REACTIVITY

Stability:	Stable under recommended storage conditions.
Polymerization:	None under normal processing.
Hazardous decomposition products:	Lithium oxides (Li_2O) , Hydrogen chloride gas, Chlorine
Materials to avoid:	Strong oxidising agents Strong acids Bromine trifluoride
Conditions to avoid:	Exposure to air or moisture over prolonged periods.

11. TOXICOLOGICAL INFORMATION

Product Information
Acute toxicity

Components	**RTECS Number:**	**Selected LD50s and LC50s**
LITHIUM CHLORIDE ANHYDROUS	OJ5950000	Oral LD50 Rat : 526 mg/kg Oral LD50 Mouse : 1165 mg/kg

Chronic toxicity:	Chronic exposure may cause nausea and vomiting, higher exposure causes unconsciousness.
Local effects:	Symptoms of overexposure may be headache, dizziness, tiredness, nausea and vomiting.
Specific effects:	May include moderate to severe erythema (redness) and moderate edema (raised skin), nausea, vomiting, headache.
Primary irritation:	No data is available on the product itself.
Carcinogenic effects:	Possible carcinogen
Mutagenic effects:	Substances which cause concern for man owing to possible mutagenic effects but for which the available information is not adequate for making a satisfactory assessment.
Reproductive toxicity:	Experiments have shown reproductive toxicity effects on laboratory animals.

12. ECOLOGICAL INFORMATION

Mobility:	No data available
Bioaccumulation:	No data available
Ecotoxicity effects:	No data available
Aquatic toxicity:	May cause long-term adverse effects in the aquatic environment.

Components	U.S. DOT - Appendix B - Marine Pollutan	U.S. DOT - Appendix B - Severe Marine Pollutants	United Kingdom - The Red List:
LITHIUM CHLORIDE ANHYDROUS	Not Listed	Not Listed	Not Listed

Components	Germany VCI (WGK)	World Health Organization (WHO) - Drinking Water	Ecotoxicity - Fish Species Data
LITHIUM CHLORIDE ANHYDROUS	Not Listed	Not Listed	Not Listed

Components	Ecotoxicity - Freshwater Algae Data	Ecotoxicity - Microtox Data	Ecotoxicity - Water Flea Data
LITHIUM CHLORIDE ANHYDROUS	Not Listed	Not Listed	Not Listed

Components	EPA - ATSDR Priority List	EPA - HPV Challenge Program Chemical List	California - Priority Toxic Pollutants
LITHIUM CHLORIDE ANHYDROUS	Not Listed	Not Listed	Not Listed

Components	California - Priority Toxic Pollutants	California - Priority Toxic Pollutants
LITHIUM CHLORIDE ANHYDROUS	Not Listed	Not Listed

13. DISPOSAL CONSIDERATIONS

Waste from residues / unused products:	Waste disposal must be in accordance with appropriate Federal, State, and local regulations. This product, if unaltered by use, may be disposed of by treatment at a permitted facility or as advised by your local hazardous waste regulatory authority. Residue from fires extinguished with this material may be hazardous.
Contaminated packaging:	Do not re-use empty containers
Methods for cleaning up:	Soak up with inert absorbent material.

14. TRANSPORT INFORMATION

UN/Id No:	Not regulated

DOT:

Proper shipping name: Not Regulated

Components	U.S. DOT - Appendix A Table 1 - Reportable Quantities
LITHIUM CHLORIDE ANHYDROUS	Not Listed

TDG (Canada):

WHMIS hazard class: D2b toxic materials

: :  :

IMDG/IMO

IMDG - Hazard Classifications Not Applicable

Components	U.S. DOT - Appendix B - Marine Pollutan	U.S. DOT - Appendix B - Severe Marine Pollutants
LITHIUM CHLORIDE ANHYDROUS	Not Listed	Not Listed

IMO-labels:

15. REGULATORY INFORMATION

International Inventories

Components	
LITHIUM CHLORIDE ANHYDROUS	
Inventory - United States TSCA - Sect. 8(b)	Present
Canada DSL Inventory List -	Present
Australia (AICS):	Present
Inventory - China:	Present
EU EINECS List -	231-212-3
Inventory - Japan:	1-231
Korean KECL:	KE-22552
Philippines PICCS:	Present

U.S. regulations:

Components	California Proposition 65 -	Massachusetts Right to Know List:	New Jersey Right to Know List:	Pennsylvania Right to Know List:
LITHIUM CHLORIDE ANHYDROUS	Not Listed	[present]	sn 1119	[present]

Components	Florida substance List:	Rhode Island Right to Know List:	Illinois - Toxic Air Contaminants	Connecticut - Hazardous Air Pollutants
LITHIUM CHLORIDE ANHYDROUS	[present]	Flammable	Not Listed	Not Listed

Components	SARA 313 Emission reporting/Toxic Release of Chemicals	CERCLA/SARA - Section 302 Extremely Haz	NTP:	IARC:
LITHIUM CHLORIDE ANHYDROUS	Not Listed	Not Listed	None	None

SARA 313 Notification: The above is your notification as to the SARA 313 listing for this product(s) pursuant to Section 313 of Title III of the Superfund Ammendments and Reauthorization Act of 1986 and 40 CFR Part 372.

If you are unsure if you are subject to the reporting requirements of Section 313, or need more information, please call the EPA Emergency Planning and Community Right-To-Know Information Hotline: (800) 535-0202 or (202) 479-2499 (in Washington, DC or Alaska).

State Notification: The above information is your notice as to the Right-to-Know listings of the stated product(s). Individual states will list chemicals for a variety of reasons including, but not limited to, the compounds toxicity; carcinogenic, tumorigenic and/or reproductive hazards; and the compounds environmental impact if accidentally released.

16. OTHER INFORMATION

Prepared by: Health & Safety
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End of Safety Data Sheet

APPENDIX D

Report of Recent NREL Demonstration Tests with the Same LDAC Technology as Used in the Project Phases II/A and II/B

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

Eric Kozubal, Lesley Herrmann, and Michael Deru
National Renewable Energy Laboratory

Jordan Clark
University of Texas, Austin

Andy Lowenstein
AIL Research

NREL is a national laboratory of the U.S. Department of Energy, Office of Energy Efficiency & Renewable Energy, operated by the Alliance for Sustainable Energy, LLC.

Technical Report
NREL/TP-5500-60695
September 2014

Contract No. DE-AC36-08GO28308

NATIONAL RENEWABLE ENERGY LABORATORY

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Acknowledgments

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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.

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

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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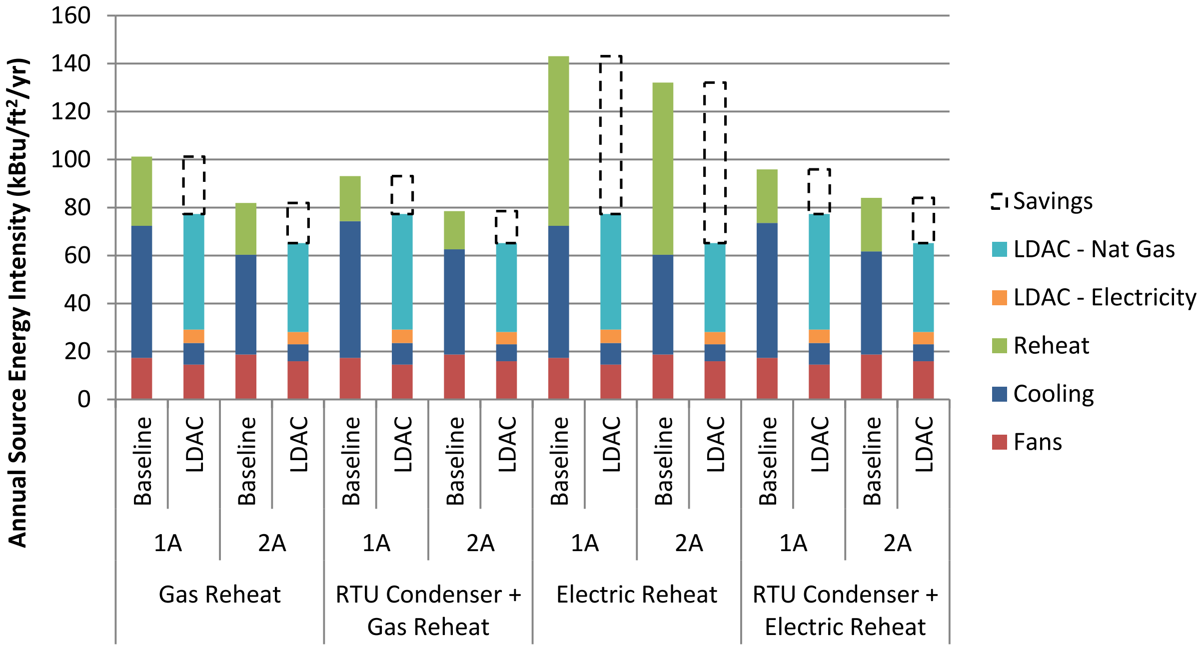

Figure ES–1 Annual source end use energy consumption and savings – single-stage regenerator (kBtu/ft^2/yr)
(Credit: Lesley Herrmann/NREL)

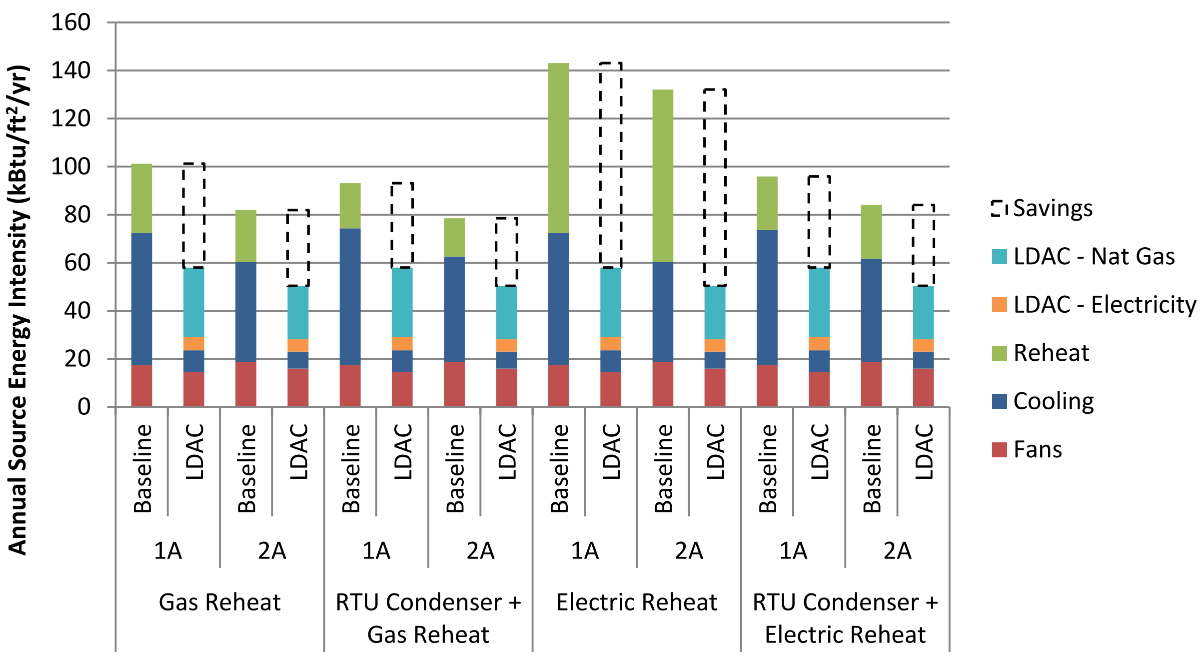

Figure ES–2 Annual source end use energy consumption and savings – two-stage regenerator (kBtu/ft^2/yr)
(Credit: Lesley Herrmann/NREL)

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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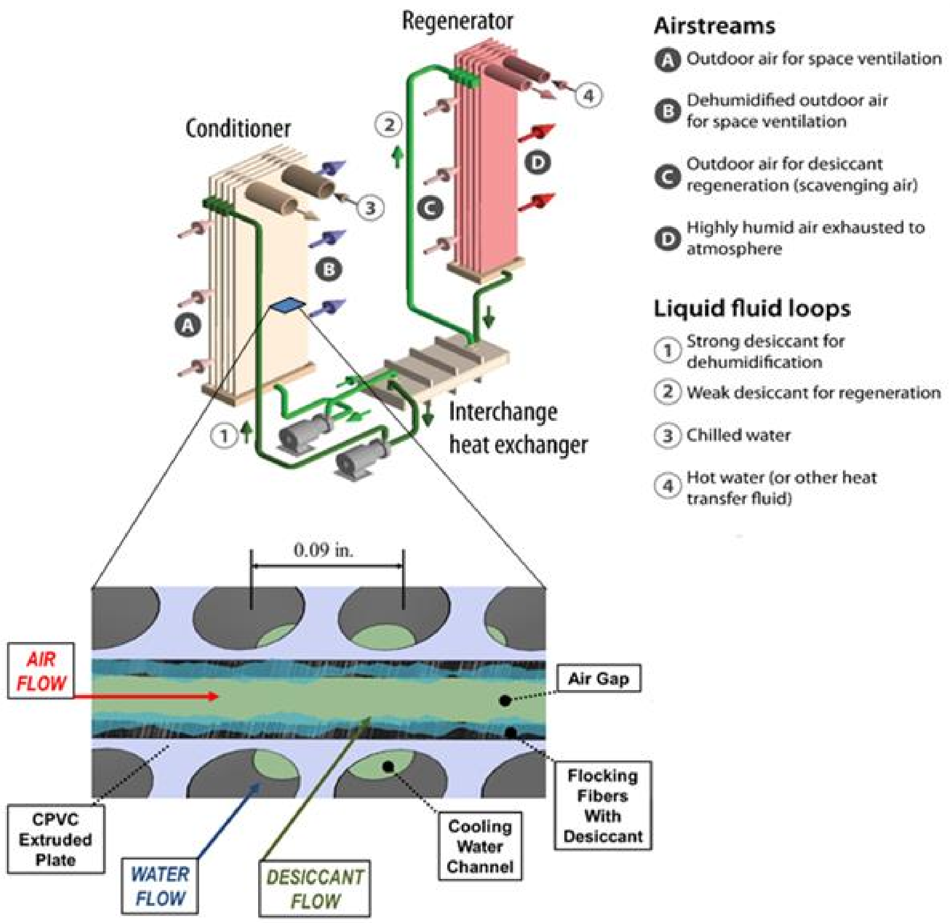

Figure 1–1 Core components of a low-flow LDAC
(Credit: Adapted from Lowenstein et al. 2006, adapted and used with permission)

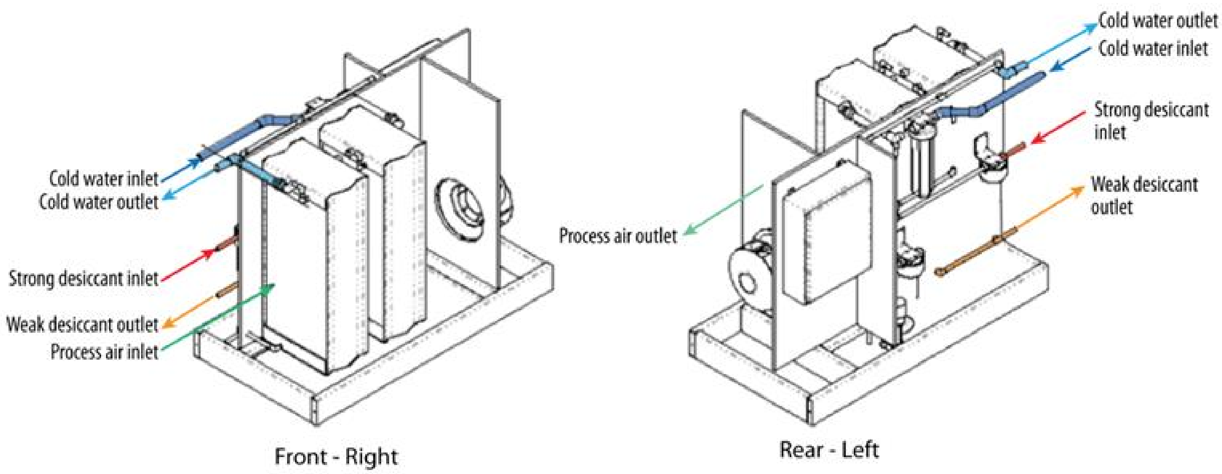

Figure 1–2 Cutaway view of LDAC conditioner (regenerator not shown)
(Credit: Andy Lowenstein, AIL Research, with permission)

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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).

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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).

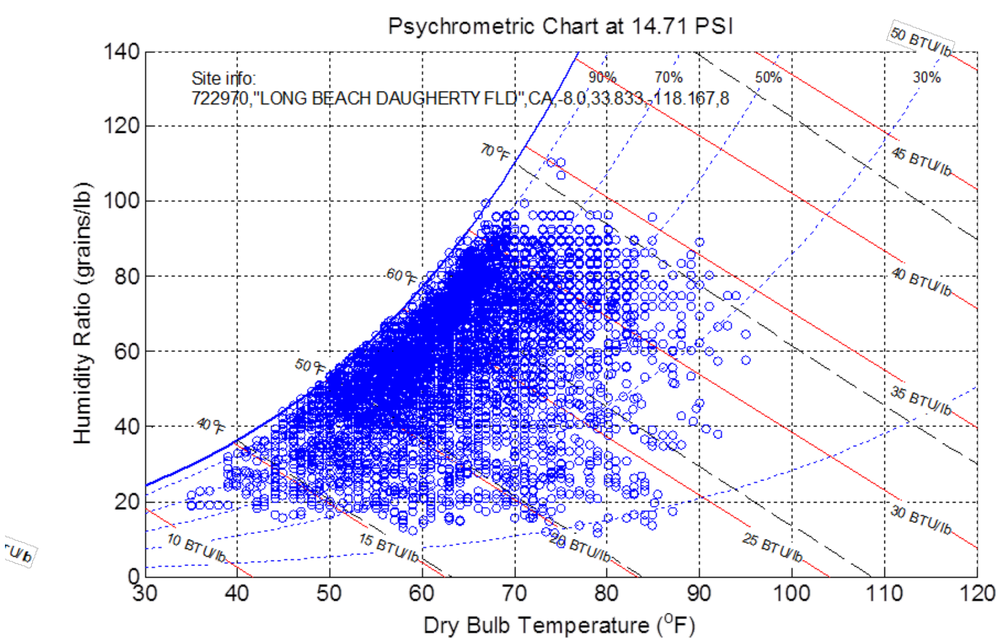

Figure 2–4 Climate analysis, Long Beach, California
(Credit: Eric Kozubal/NREL)

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).

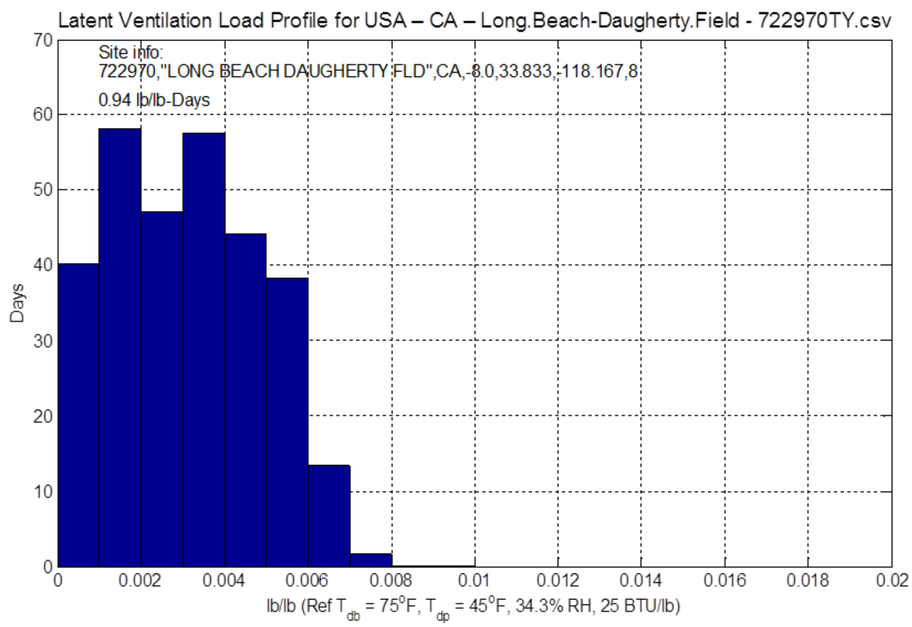

Figure 2–5 Pound-per-pound days, Long Beach, California
(Credit: Eric Kozubal/NREL)

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

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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.

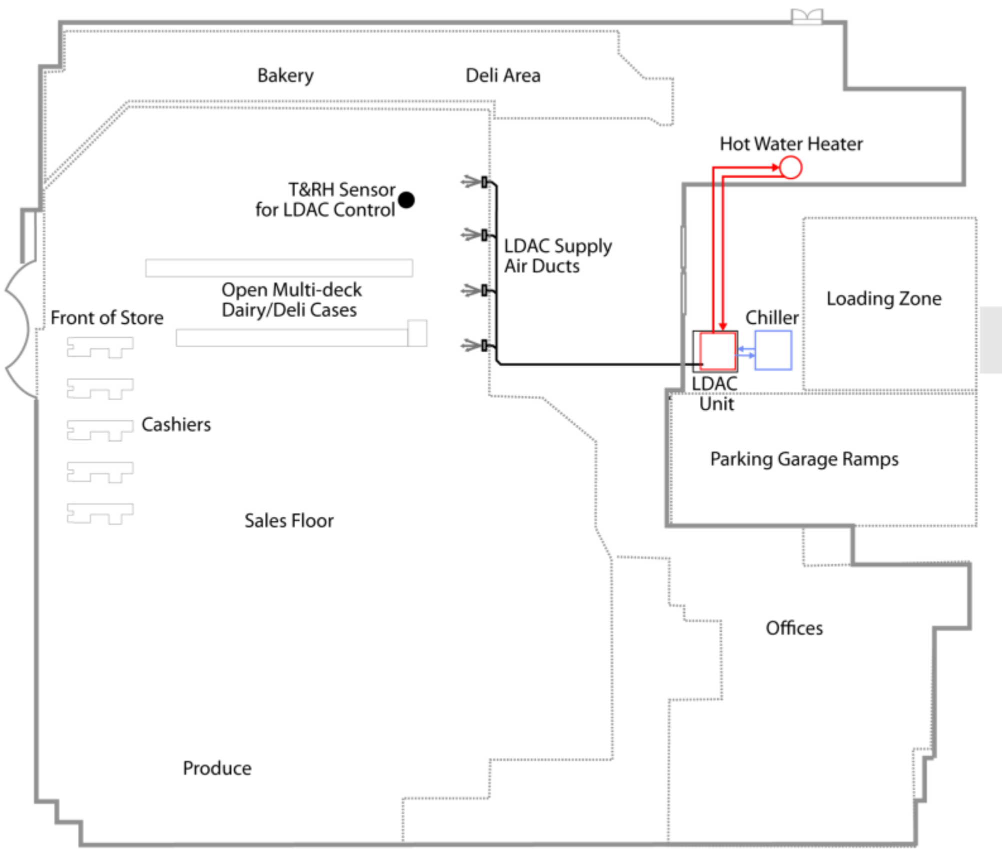

Figure 2–7 Building layout at Whole Foods, Encinitas, California
(Credit: NREL, adapted from Whole Foods Market, with permission)

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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.

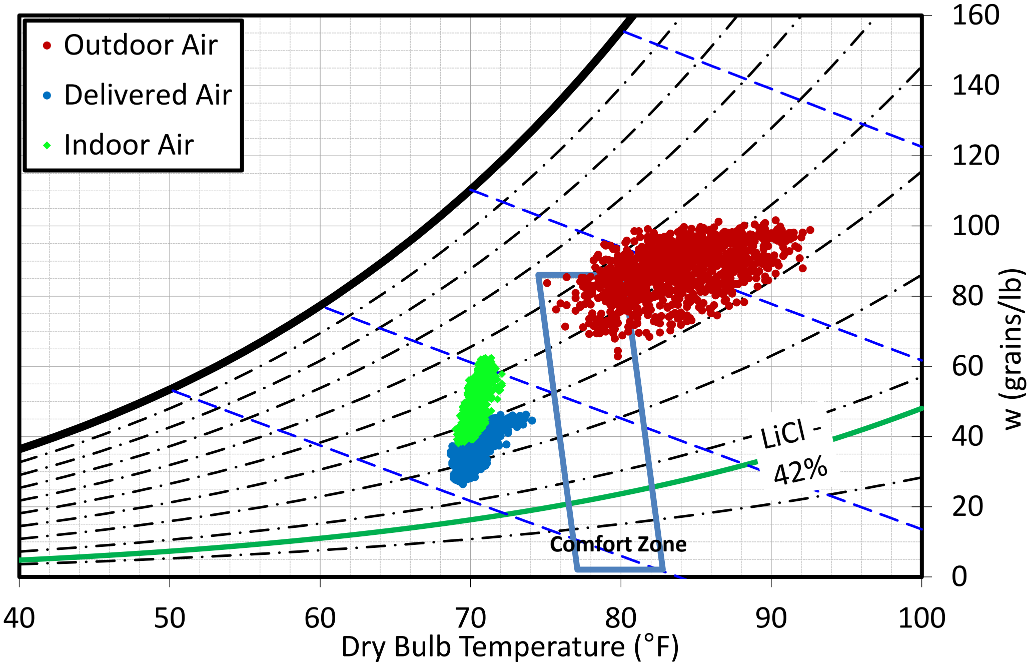

Figure 2–8 Psychrometric chart showing outdoor and delivered air from the LDAC, Encinitas, California, August–October 2012
(Credit: Eric Kozubal/NREL)

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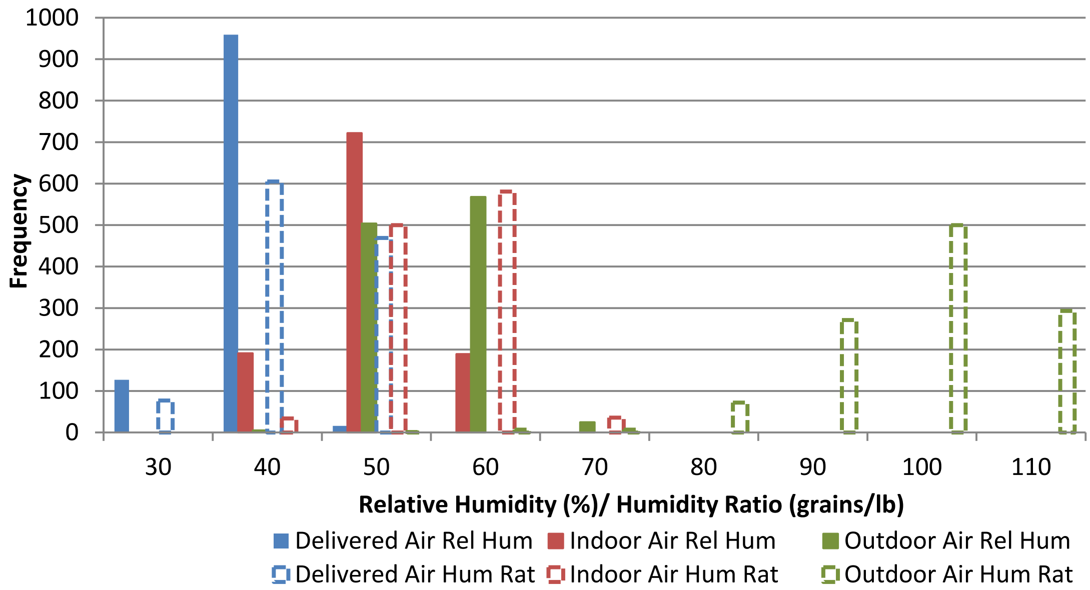

Figure 2–9 Histogram of humidity, Encinitas, California, August–October 2012
(Credit: Lesley Herrmann/NREL)

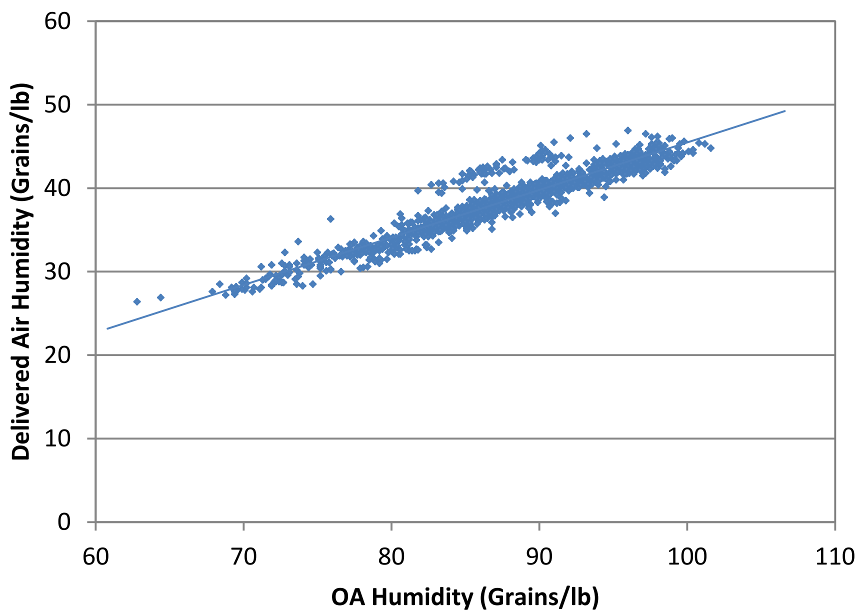

Figure 2–10 LDAC delivered humidity versus OA humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

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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.

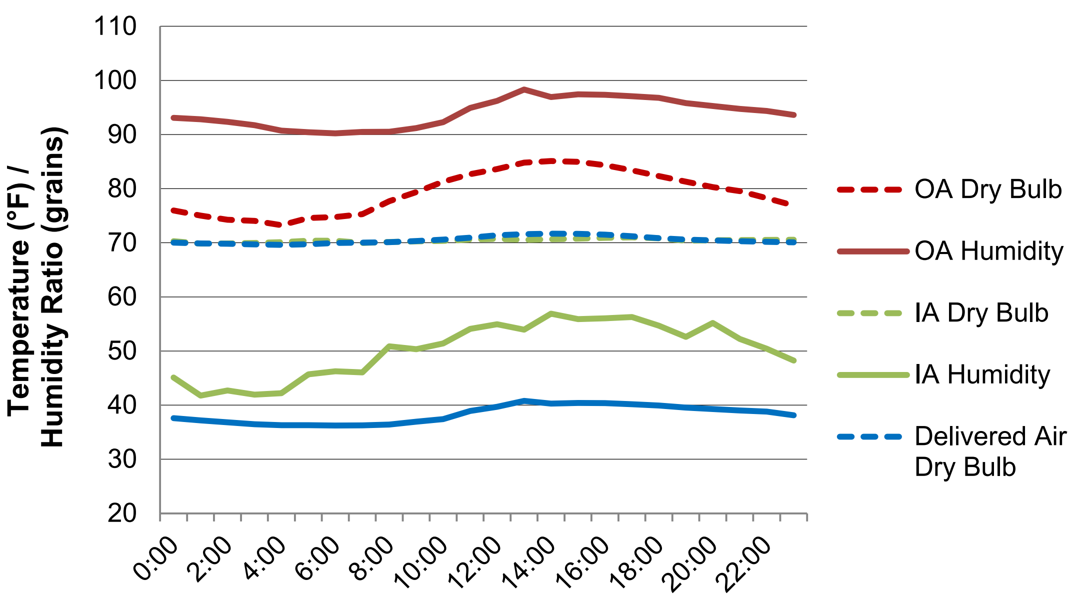

Figure 2–11 Store averaged conditions during each hour of the day, Encinitas, California, August–October 2012
(Credit: Eric Kozubal/NREL)

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.

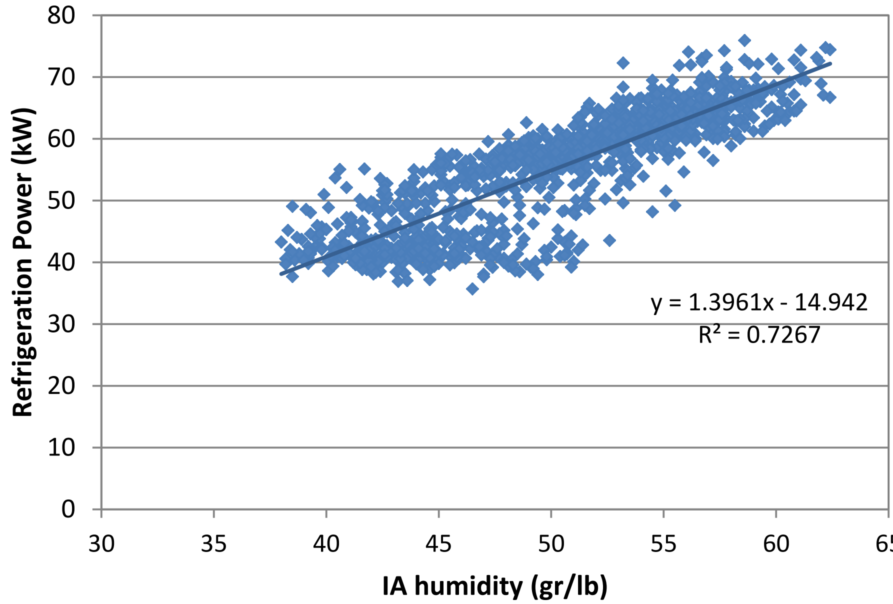

Figure 2–12 Refrigeration power versus indoor air humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

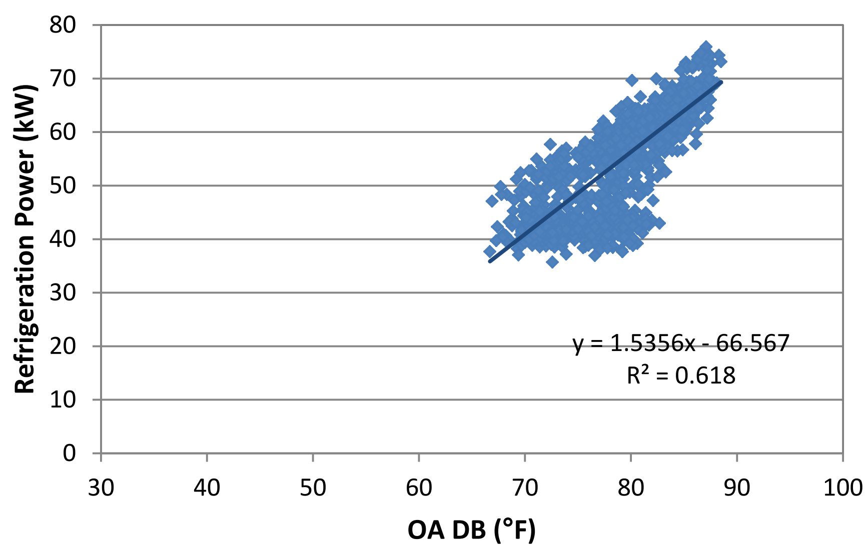

Figure 2–13 Refrigeration power versus OA DB, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

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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.

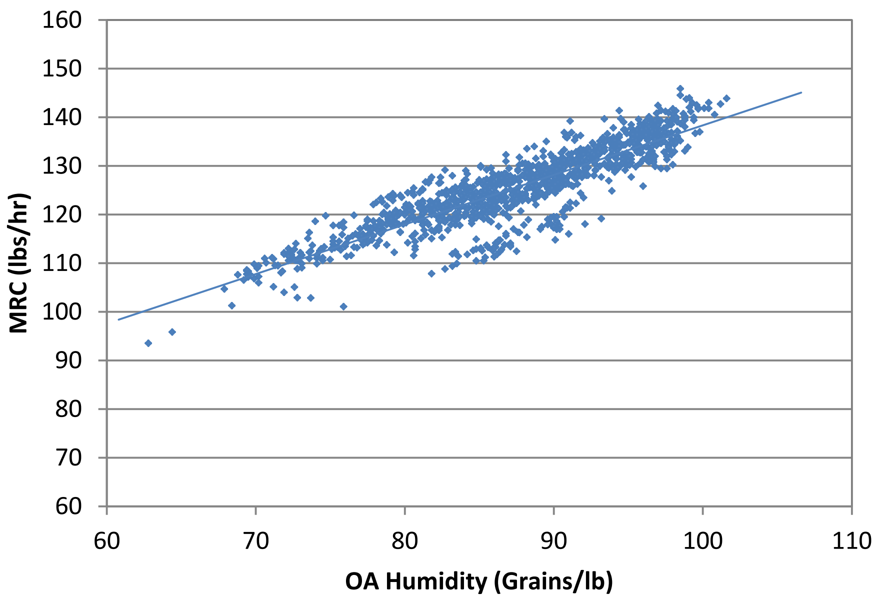

Figure 2–14 MRC versus OA humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

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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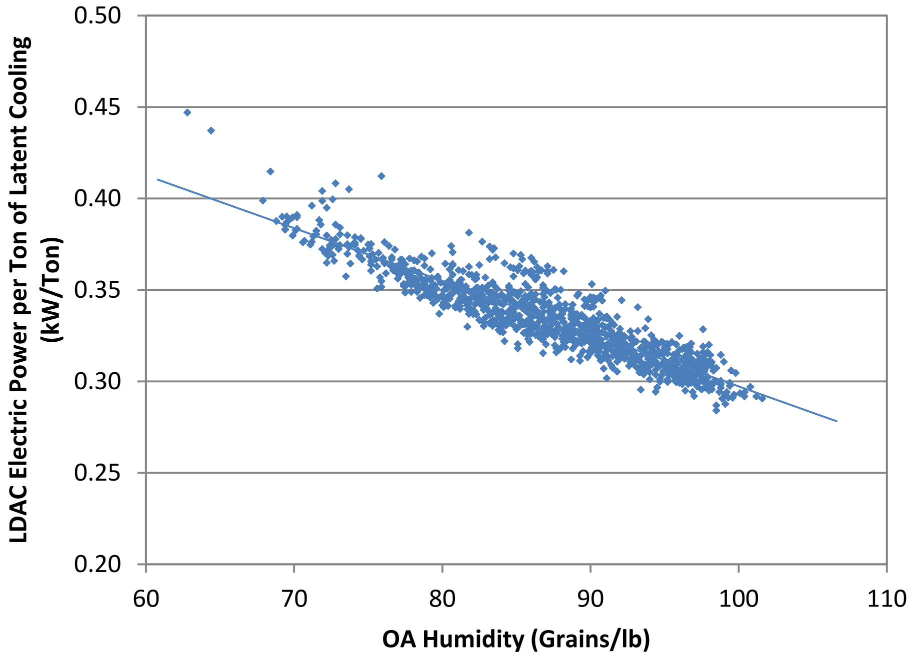

Figure 2–15 LDAC-specific electricity use versus OA humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

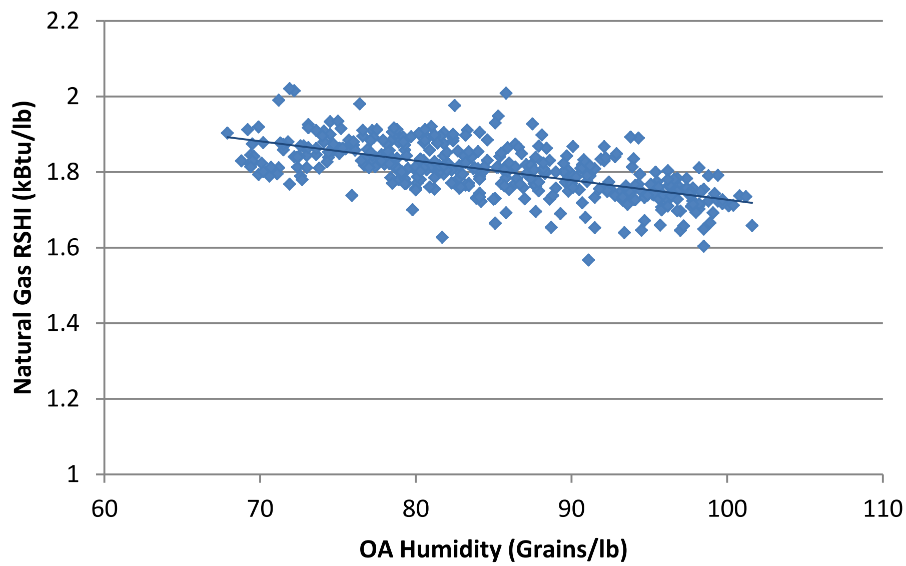

Figure 2–16 Natural gas RSHI versus OA Humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

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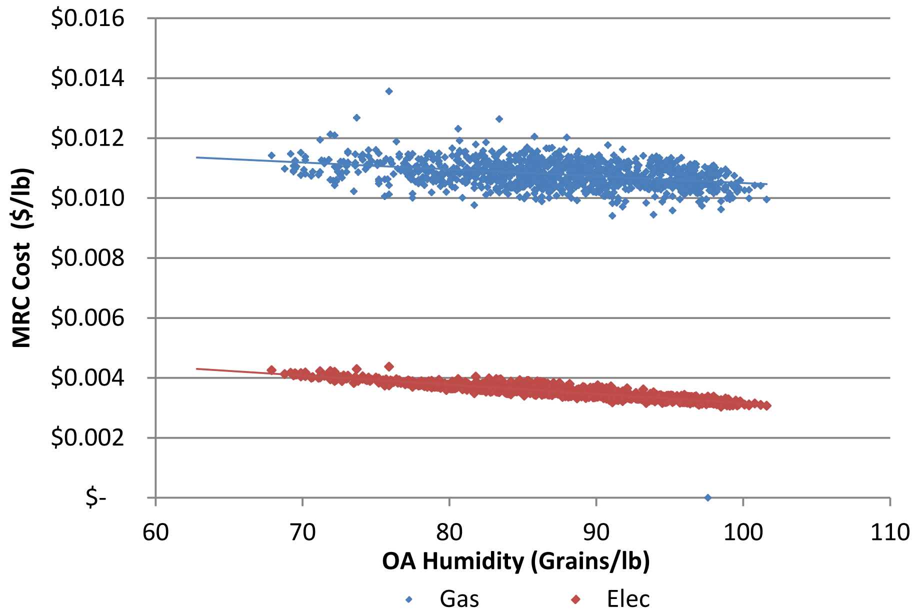

Figure 2–17 MRC cost versus OA humidity, Encinitas, California, August–October 2012
(Credit: Joe Ryan, with permission)

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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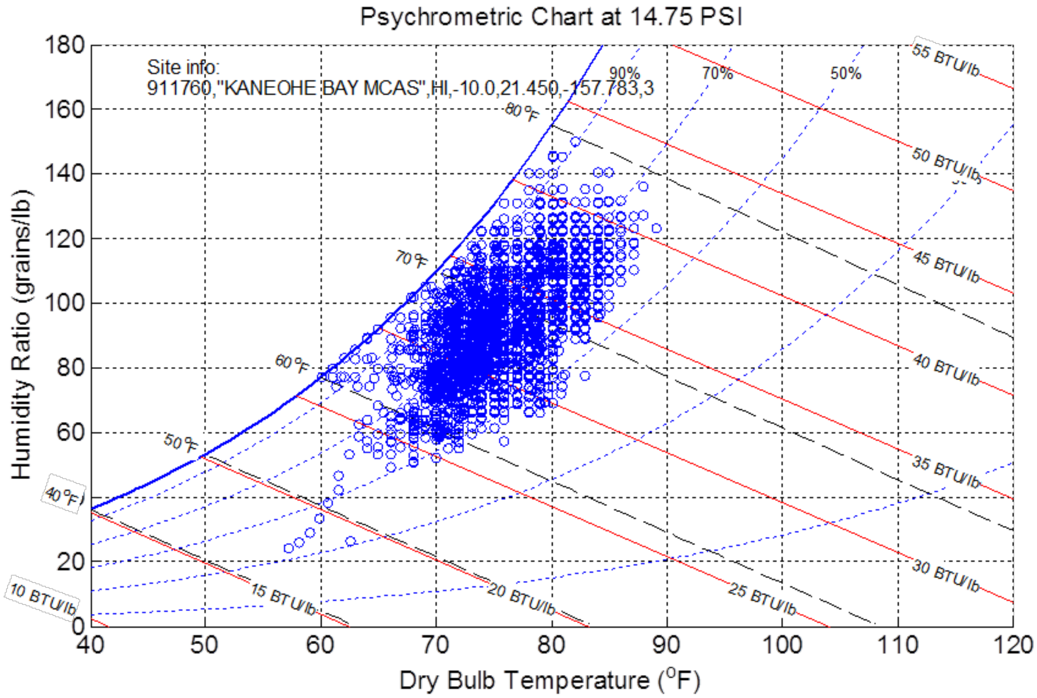

Figure 2–21 Climate analysis, Kailua, Hawaii
(Credit: Eric Kozubal/NREL)

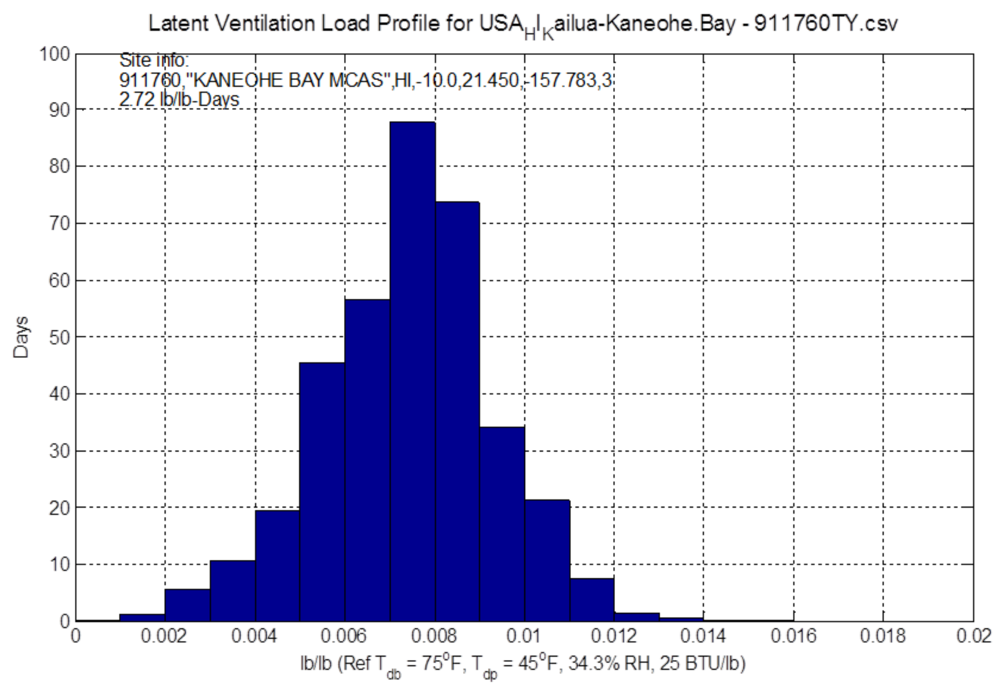

Figure 2–22 Pound-per-pound days, Kailua, Hawaii
(Credit: Eric Kozubal/NREL)

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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.

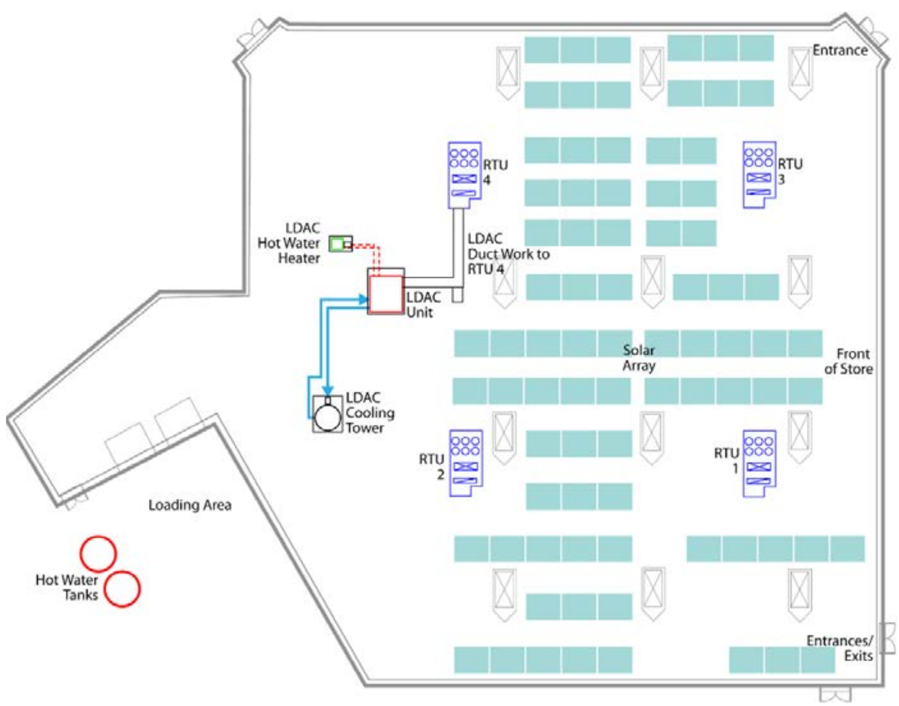

Figure 2–24 Roof equipment layout at Whole Foods Market, Kailua, Hawaii
(Credit: NREL, adapted from Whole Foods Market, with permission)

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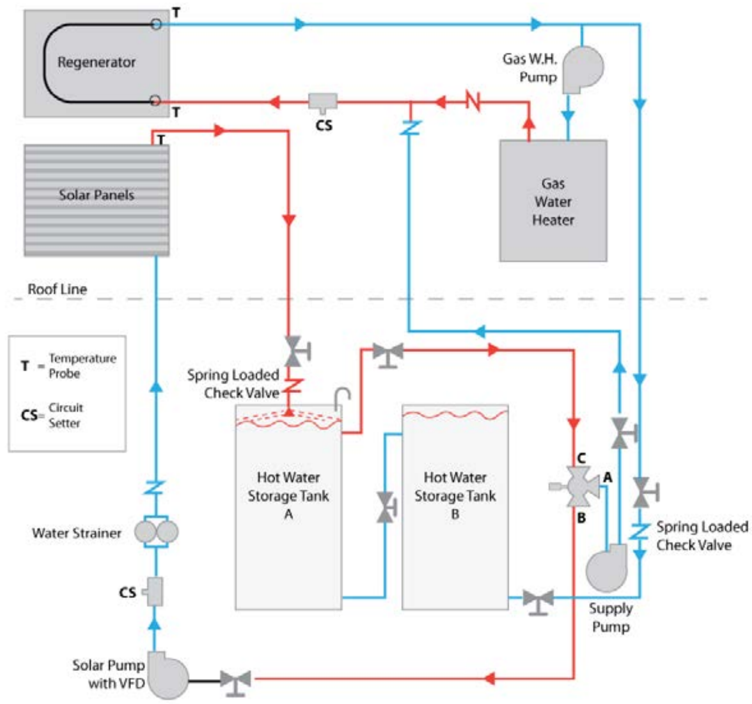

Figure 2–25 Solar thermal piping schematic at Whole Foods Market, Kailua, Hawaii
(Credit: J&J Mechanical, with permission)

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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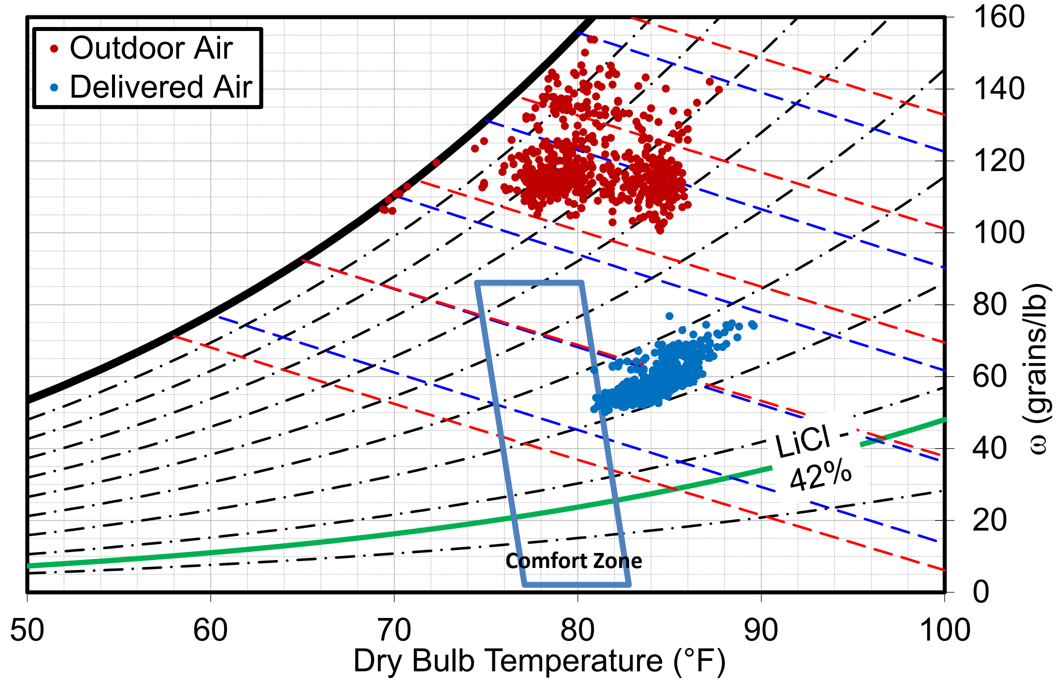

Figure 2–26 OA and delivered air plotted on a psychrometric chart, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

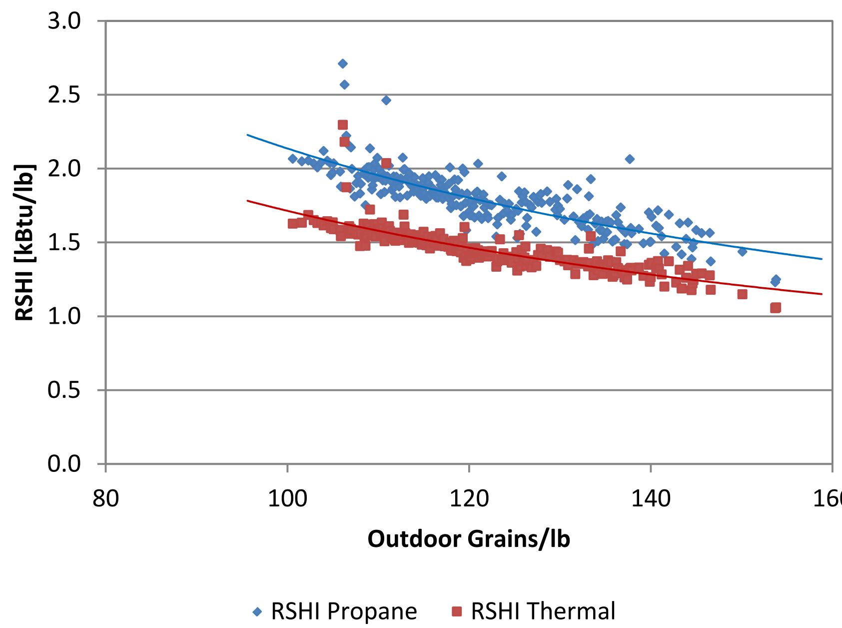

Figure 2–27 Regenerator RSHI, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

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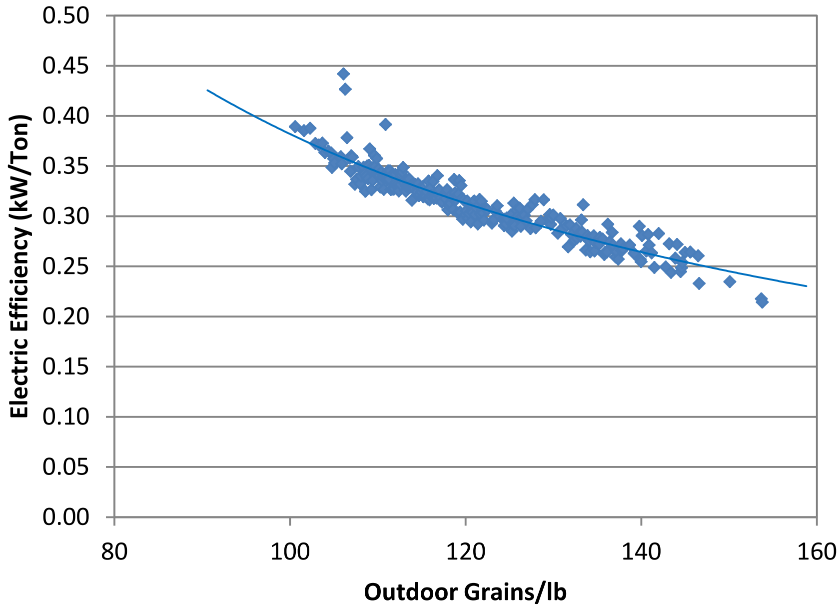

Figure 2–28 LDAC electrical efficiency, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

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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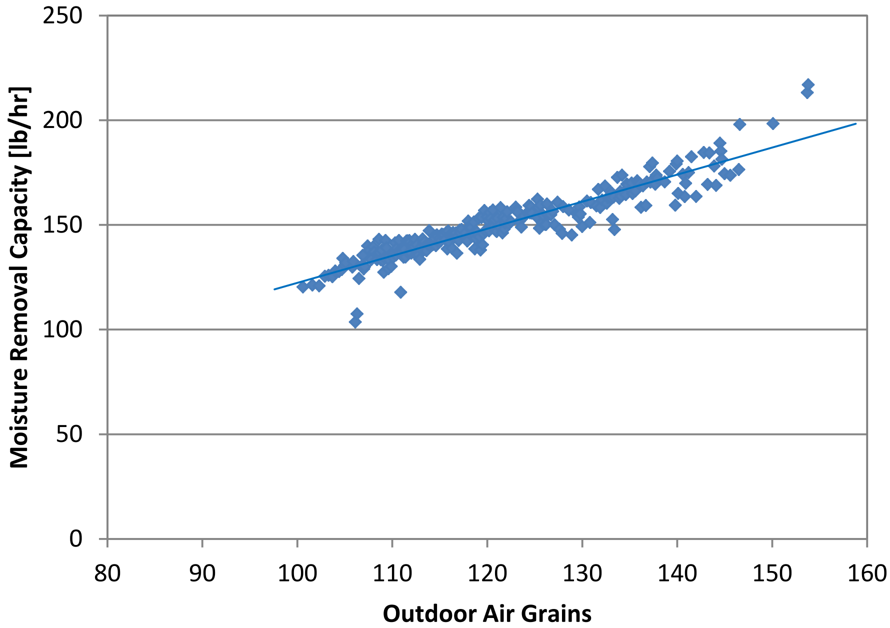

Figure 2–29 LDAC MRC, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

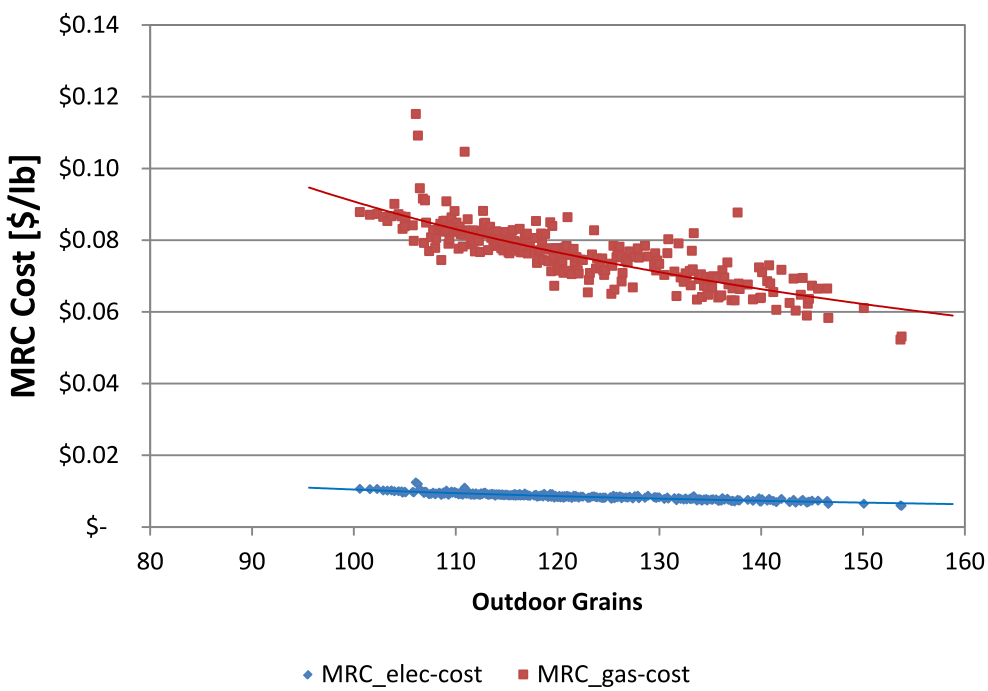

Figure 2–30 LDAC MRC cost, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

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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).

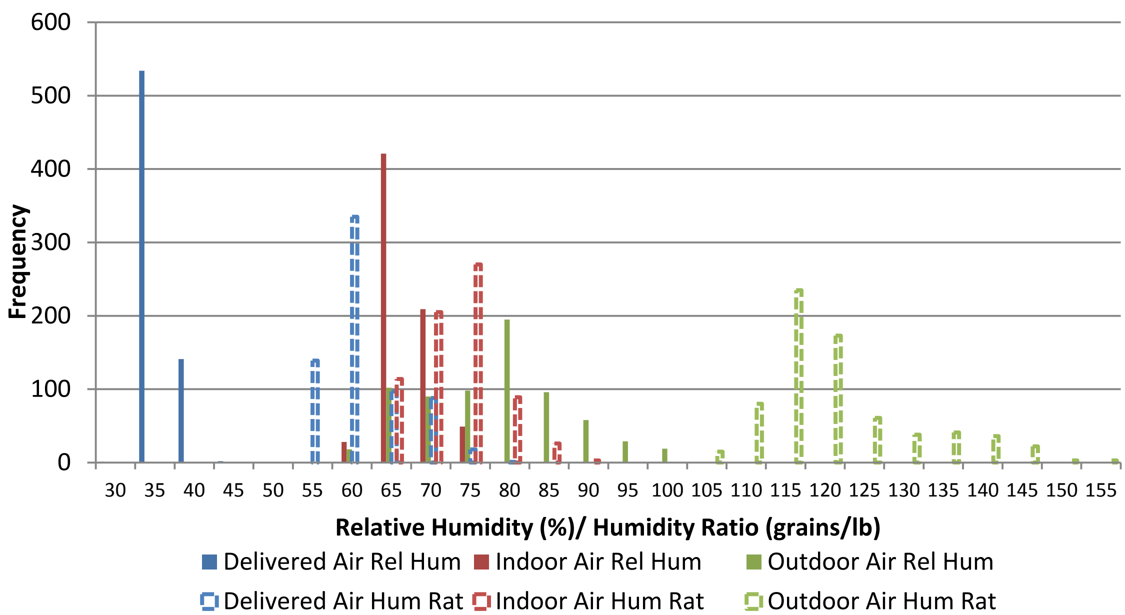

Figure 2–31 Store and site humidity levels, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Lesley Herrmann/NREL)

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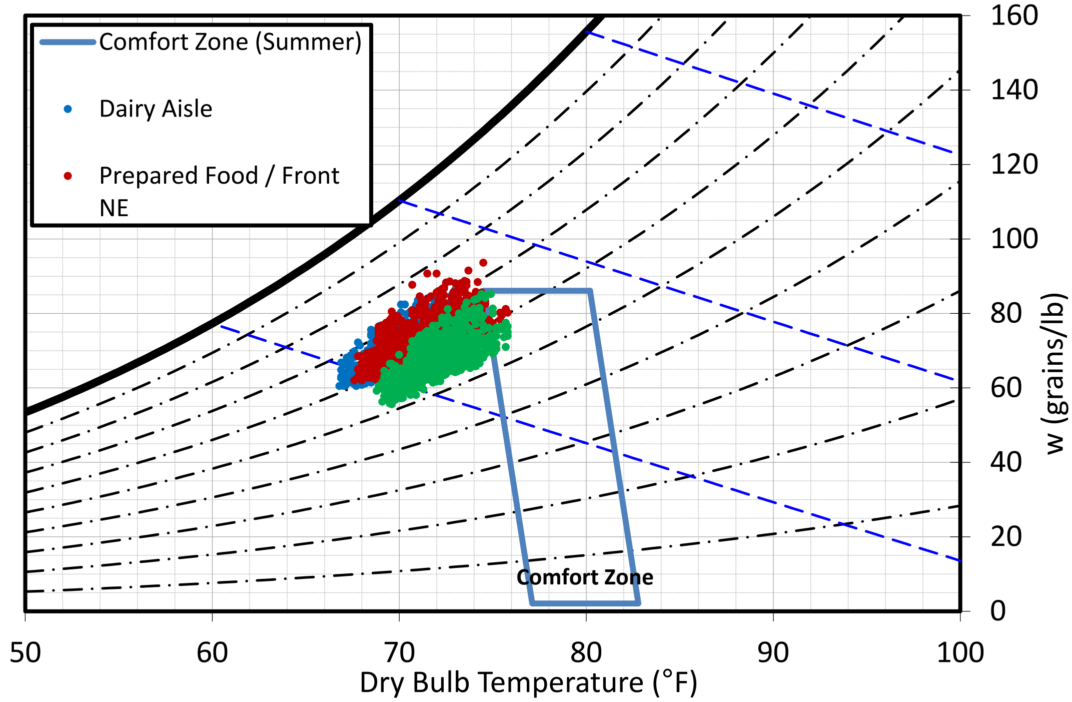

Figure 2–32 Psychrometric chart showing indoor air conditions, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Eric Kozubal/NREL)

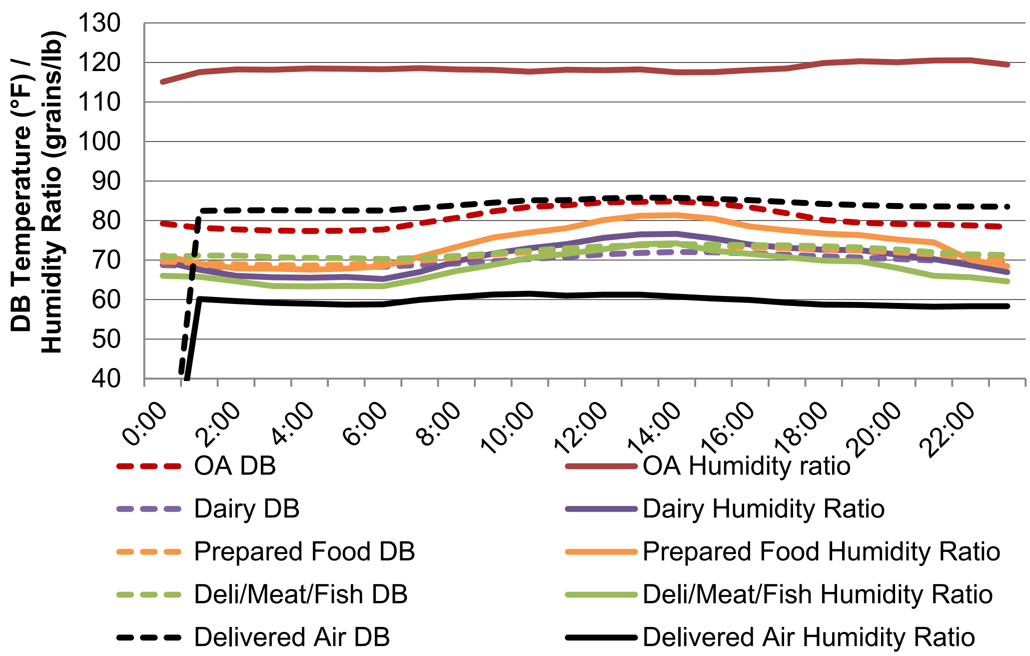

Figure 2–33 Daily average conditions, Whole Foods, Kailua, Hawaii, July–August, 2013
(Credit: Joe Ryan, with permission)

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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

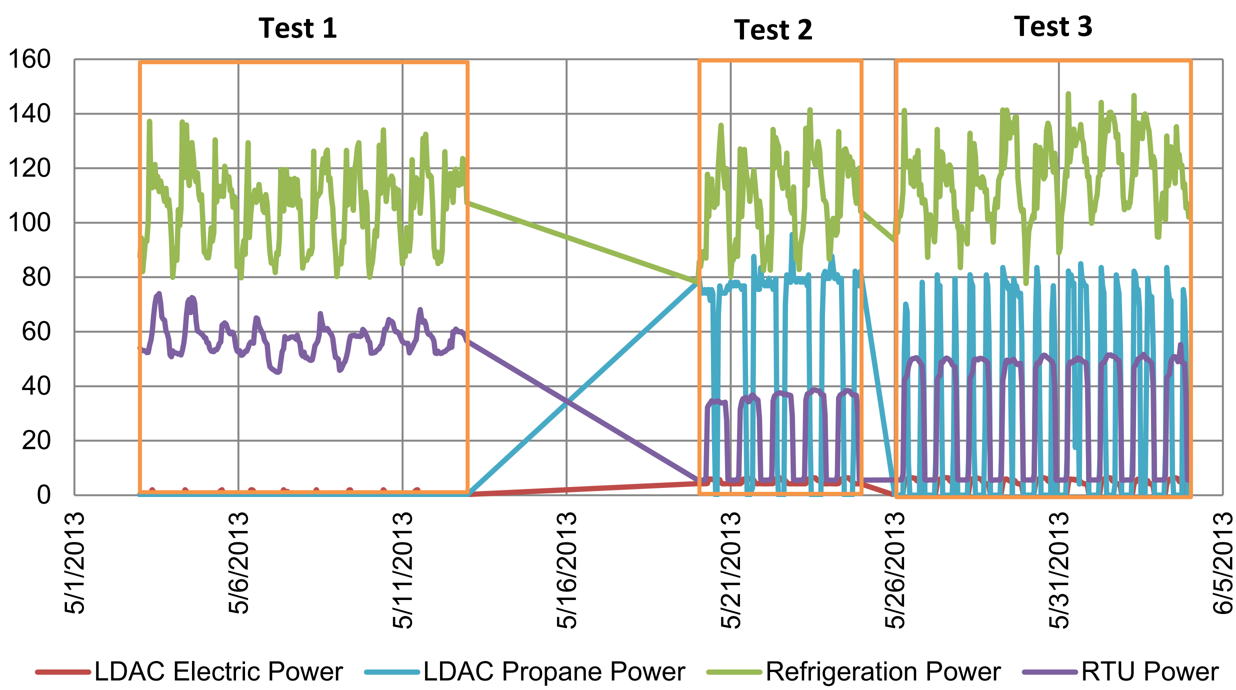

Figure 2–35 Refrigeration and HVAC systems power for tests 1–3,
Whole Foods, Kailua, Hawaii, May–June 2103
(Credit: Eric Kozubal/NREL)

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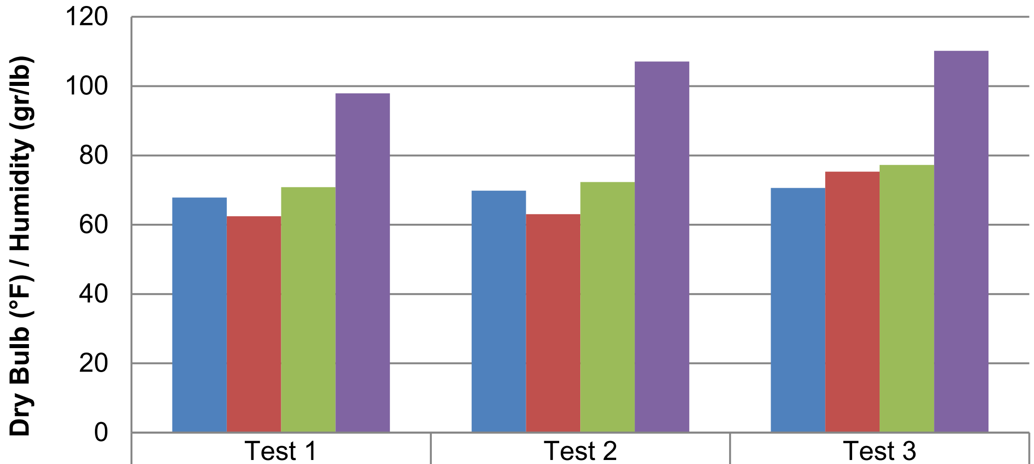
Table 2–14 Whole Foods Market, Kailua, Hawaii—
Nightly Average Electric Power Savings for Tests 2 and 3, May–June, 2103

	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.

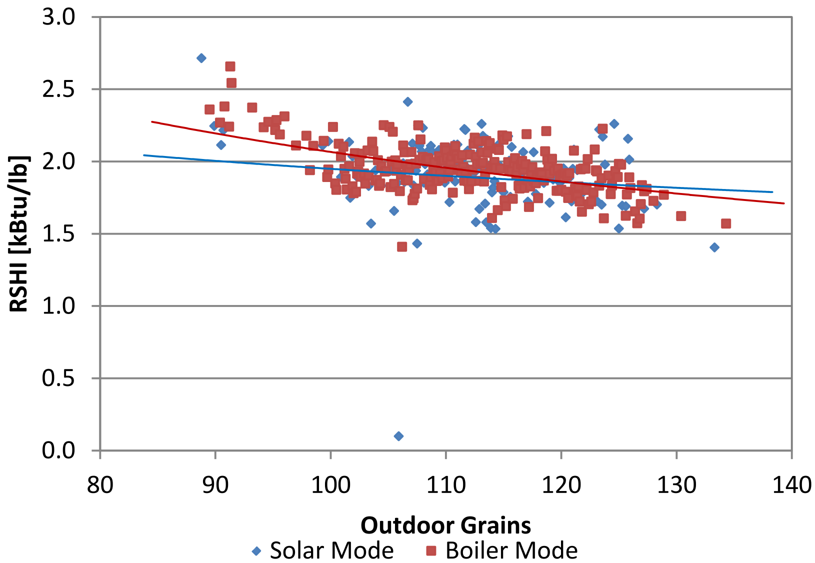

Figure 2–37 Regenerator thermal RSHI, Whole Foods, Kailua, Hawaii, May–June 2103
(Credit: Joe Ryan, with permission)

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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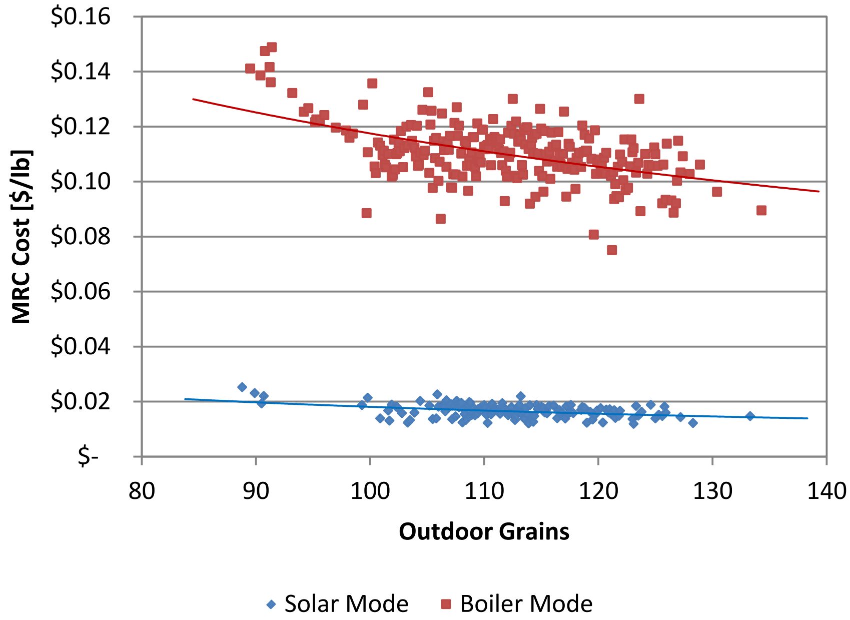

Figure 2–38 LDAC MRC cost, Whole Foods, Kailua, Hawaii, May–June 2103
(Credit: Joe Ryan, with permission)

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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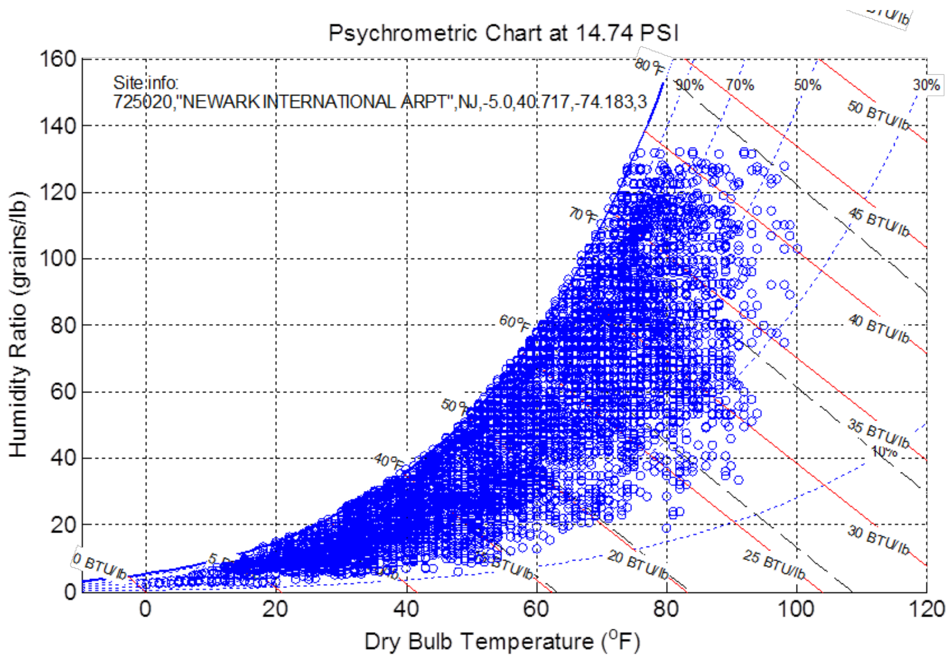

Figure 2–40 Climate analysis, Hoboken, New Jersey

(Credit: Eric Kozubal/NREL)

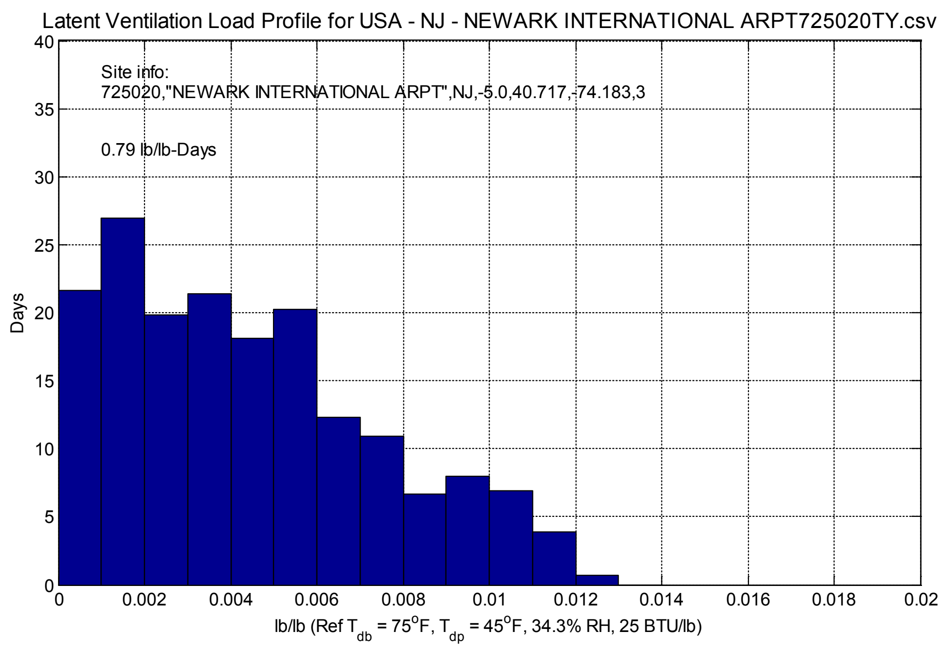

Figure 2–41 Pound-per-pound days, Hoboken, New Jersey

(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)

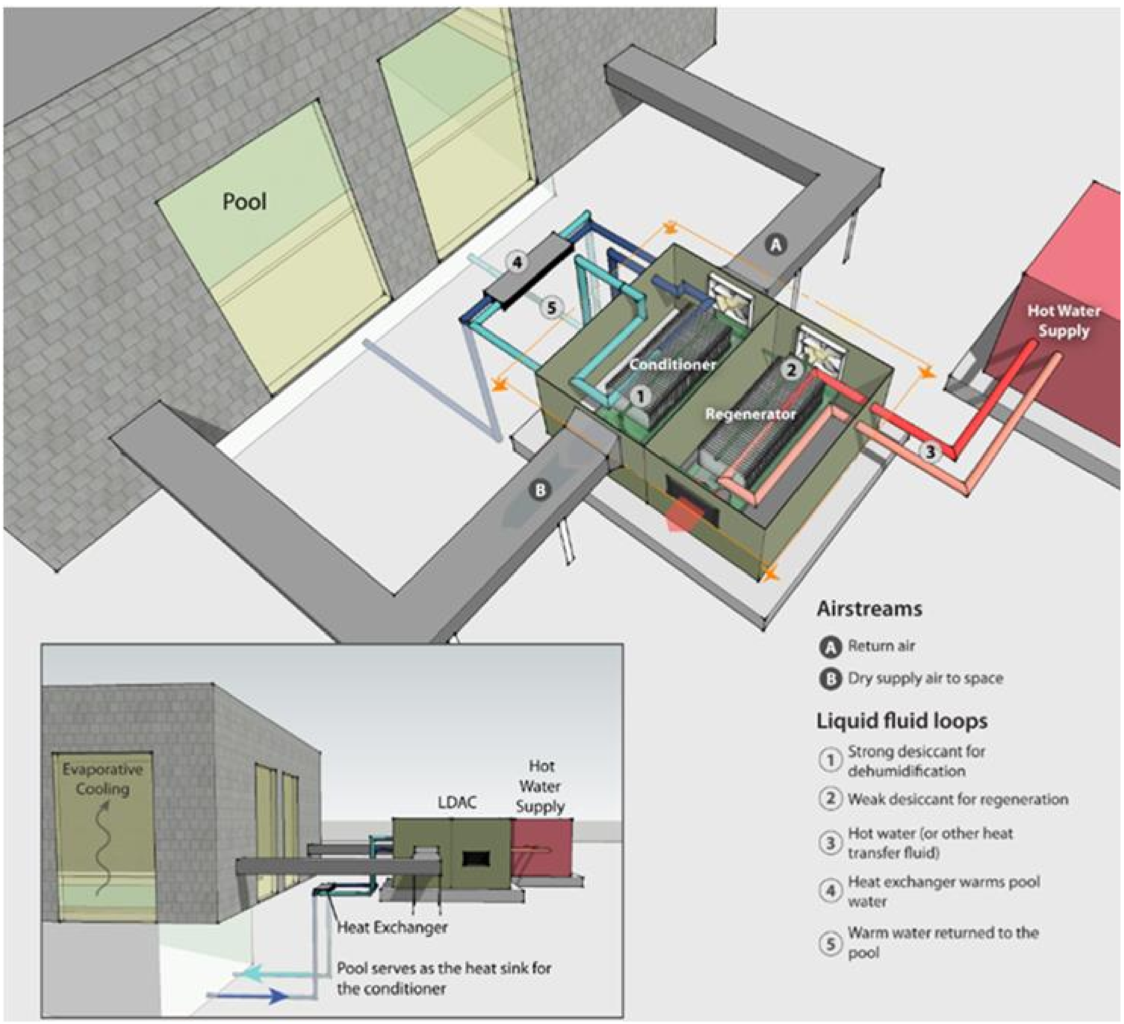

Figure 2–43 Schematic of an LDAC integrated with a pool facility

(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.

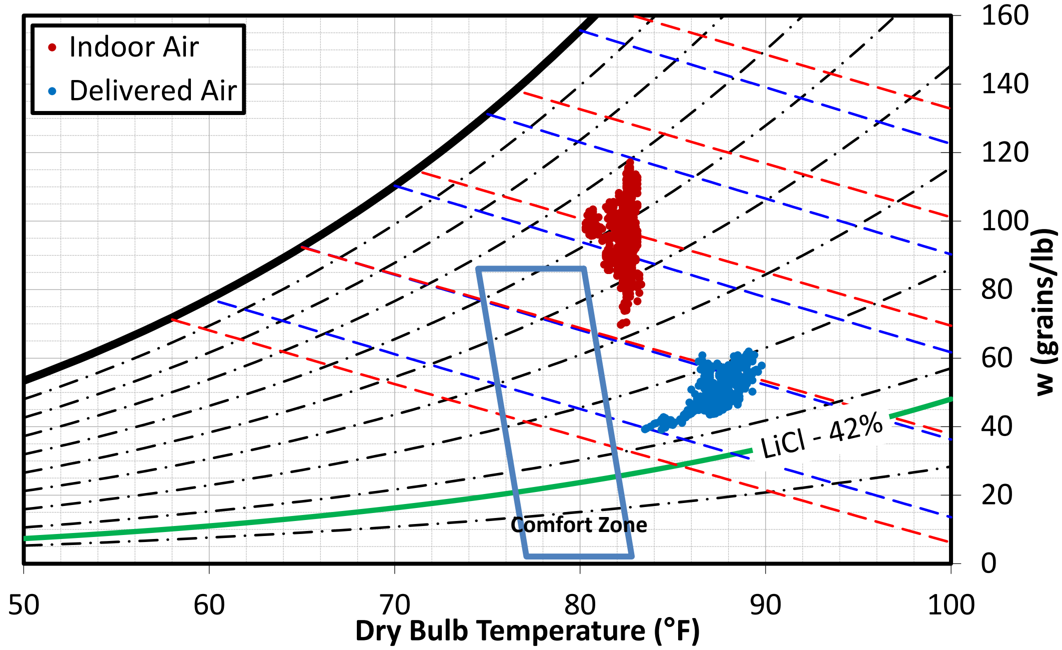

Figure 2–44 Indoor air and delivered air plotted on a psychrometric chart, Schaeffer Pool, Stevens Institute, August–September 2012

(Credit: Eric Kozubal/NREL)

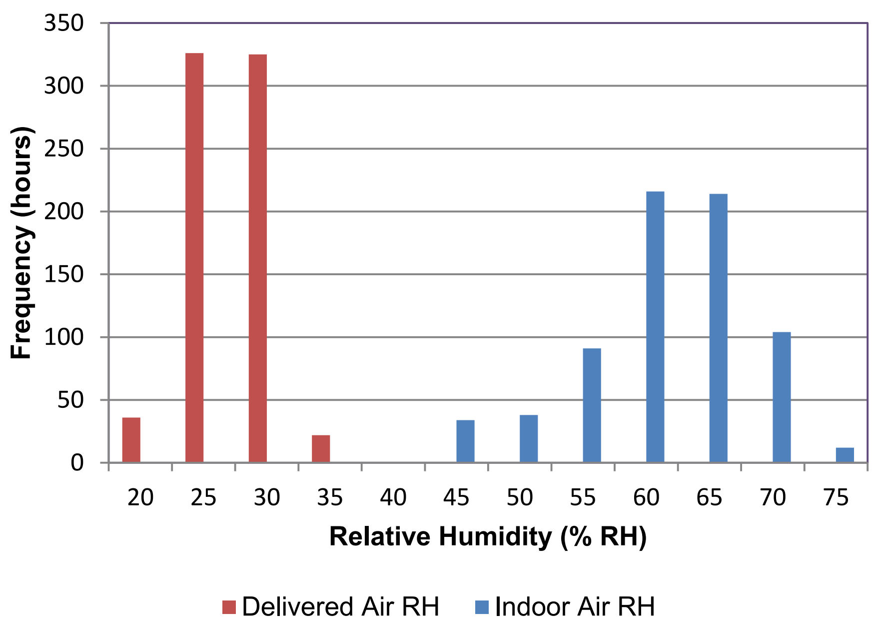

Figure 2–45 Histogram of air conditions, Schaeffer Pool, Stevens Institute, August–September 2012

(Credit: Joe Ryan, with permission)

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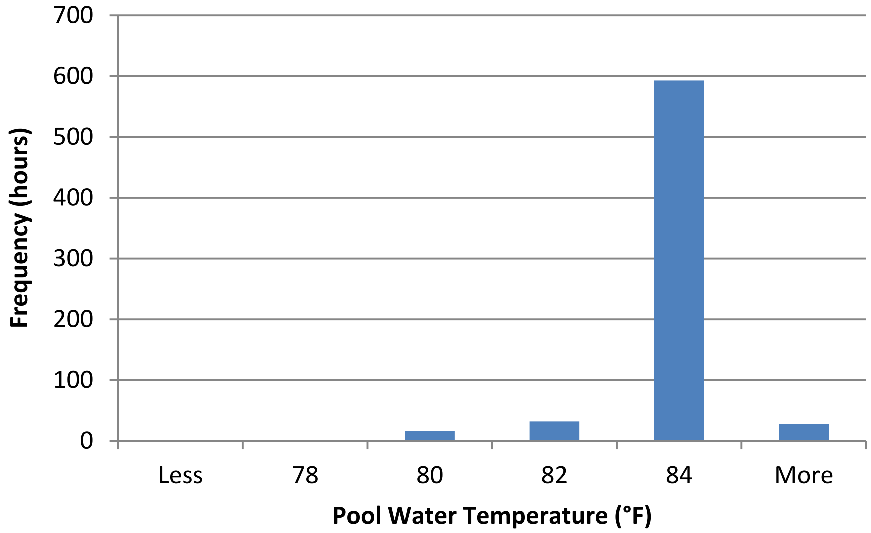

Figure 2–46 Pool water temperature, Schaeffer Pool, Stevens Institute, August–September 2012

(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.

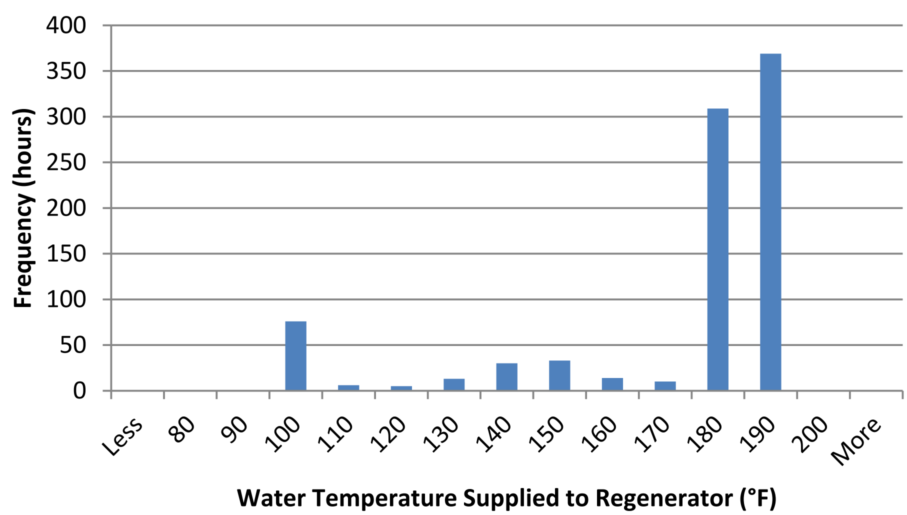

Figure 2–47 Hot water supply temperature, Schaeffer Pool, Stevens Institute, August–September 2012

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

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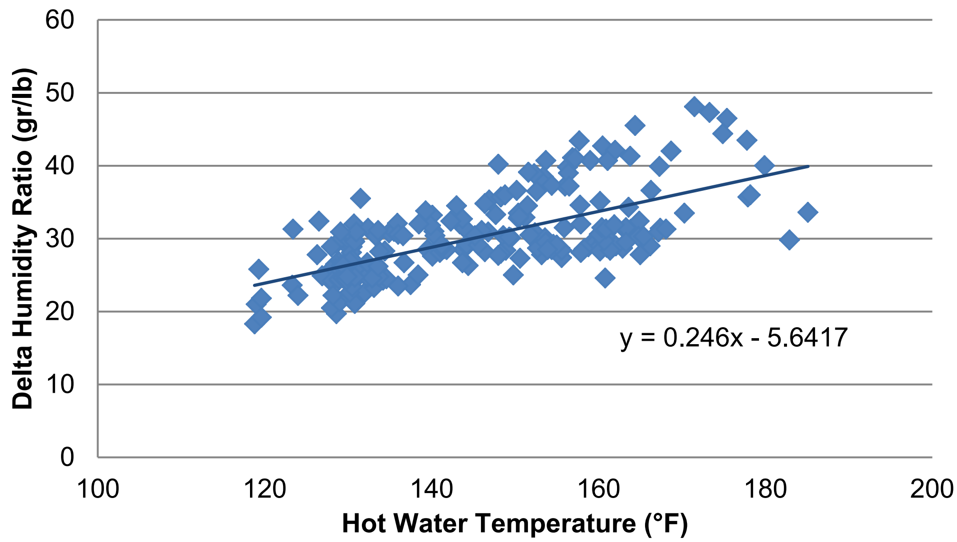

Figure 2–48 Delivered air delta humidity versus supply hot water temperature, Schaeffer Pool, Stevens Institute, August–September 2012

(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)

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

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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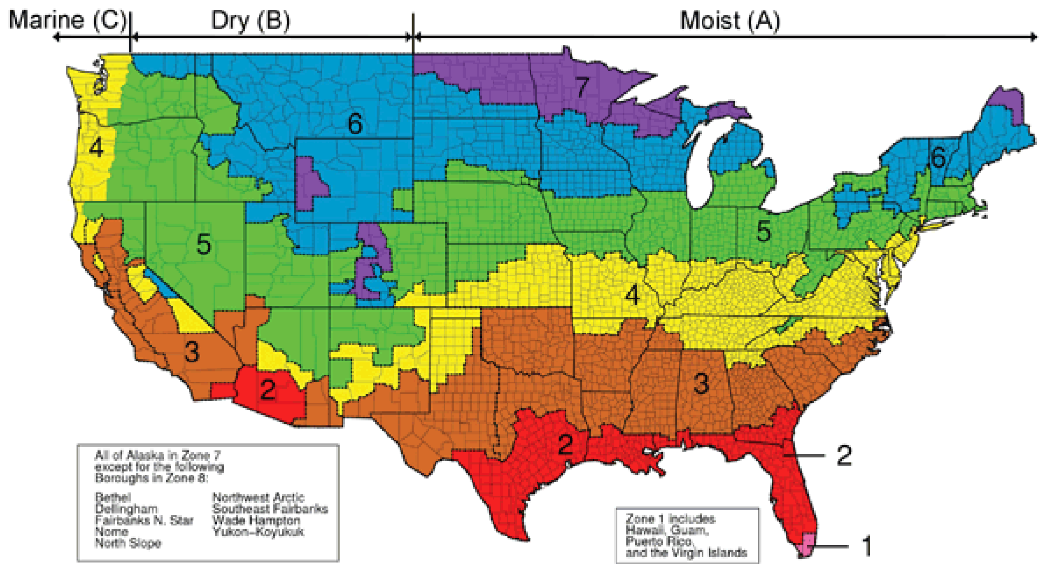

Figure 3–53 U.S. climate zone map
(Credit: DOE 2004)

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).

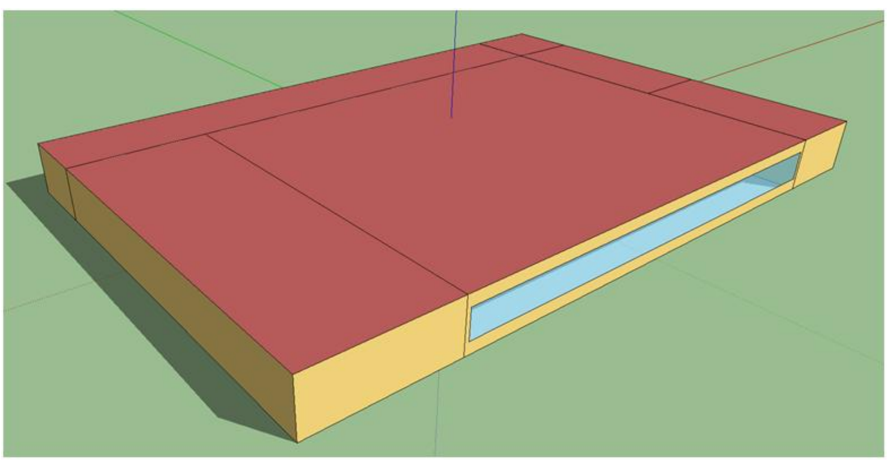

Figure 3–54 Supermarket model rendering

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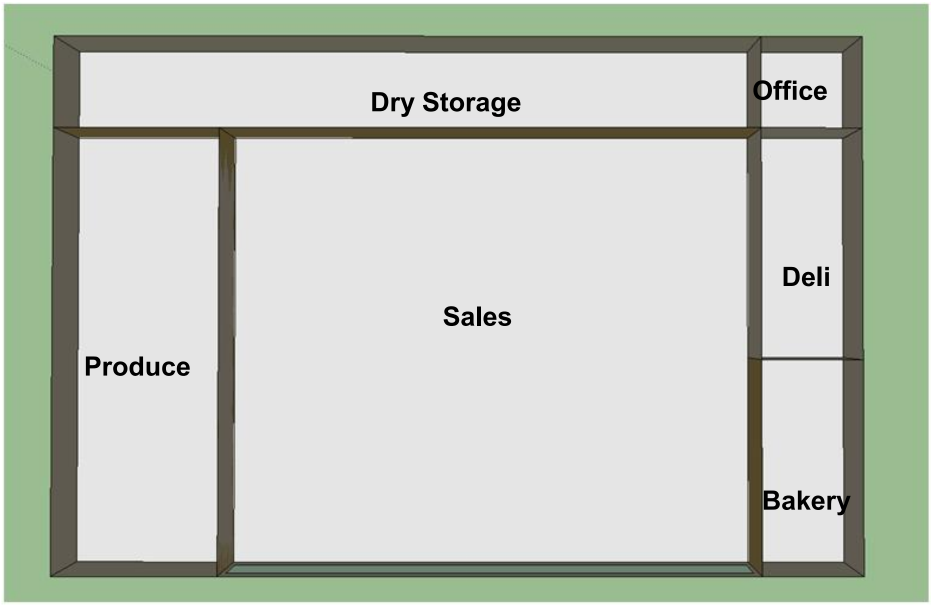

Figure 3–55 Supermarket model floor plan

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·ft^2·°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

Space Type	OA Supply		Exhaust
	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

Zone	Case		Walk-in Freezers	
	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-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 ft³)
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.

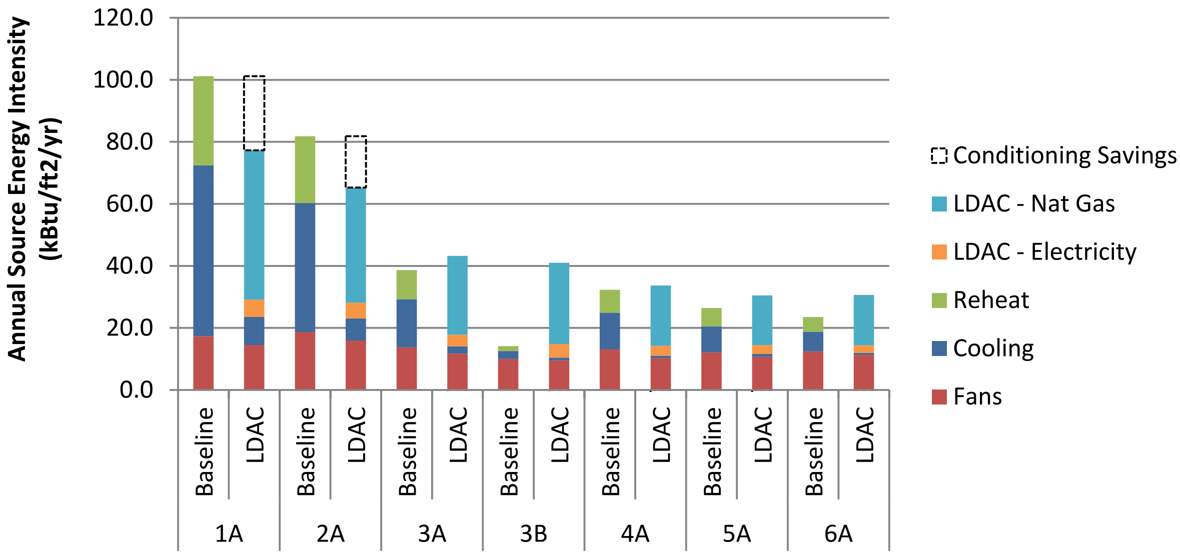

Figure 3–56 Annual ventilation and air conditioning source energy intensity and savings – natural gas reheat coils – single-stage regenerator
(Credit: Lesley Herrmann/NREL)

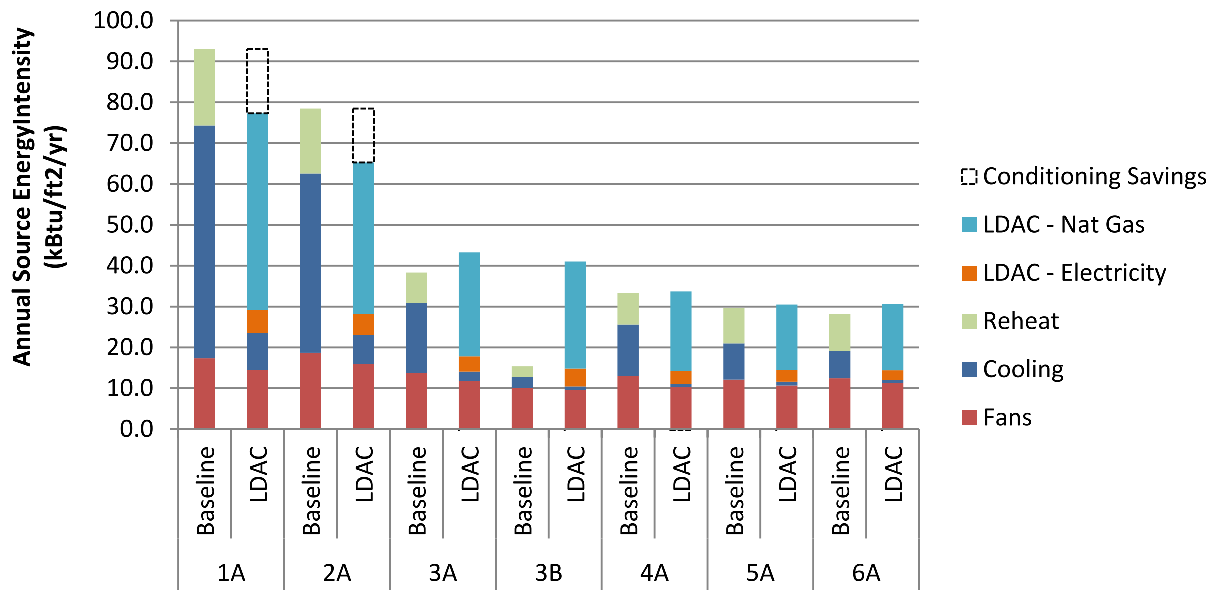

Figure 3–57 Annual ventilation and air conditioning source energy intensity and savings – RTU condenser hot-gas reheat with auxiliary natural gas reheat coils – single-stage regenerator
(Credit: Lesley Herrmann/NREL)

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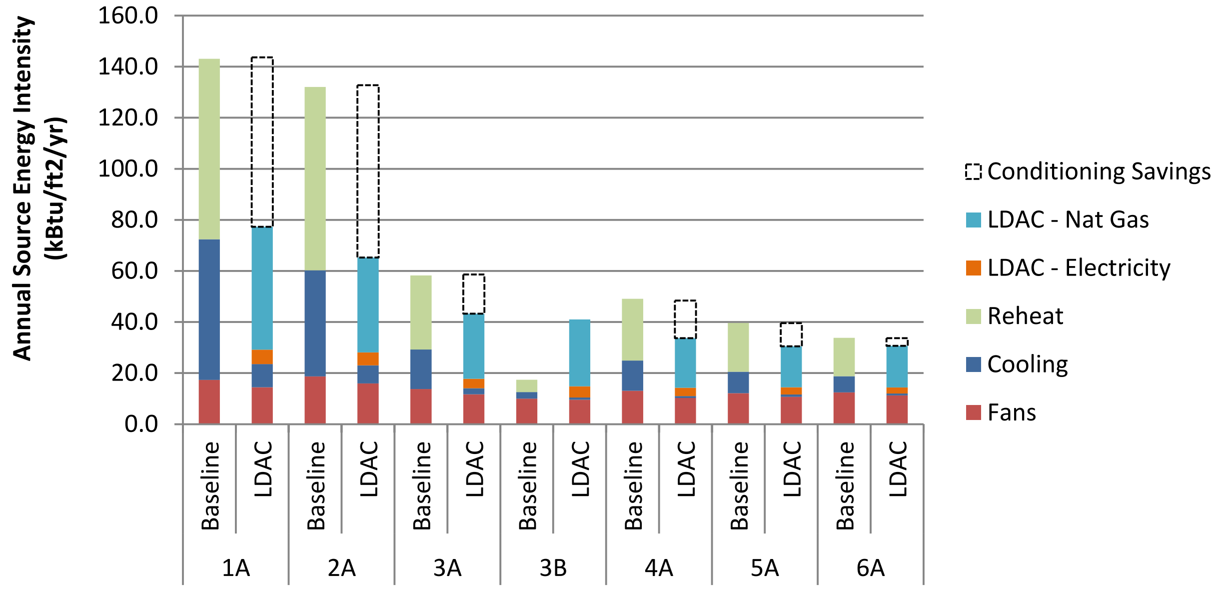

Figure 3–58 Annual ventilation and air conditioning source energy intensity and savings – electric reheat coils – single-stage regenerator
(Credit: Lesley Herrmann/NREL)

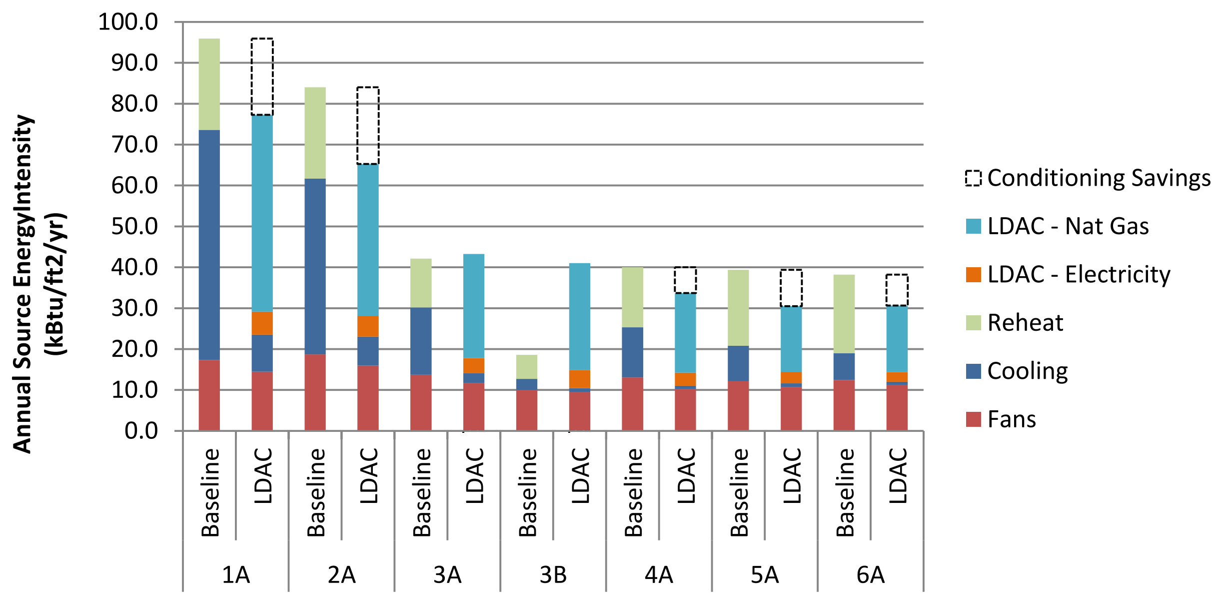

Figure 3–59 Annual conditioning source energy intensity and savings – RTU condenser hot-gas reheat with auxiliary electric reheat coils – single-stage regenerator
(Credit: Lesley Herrmann/NREL)

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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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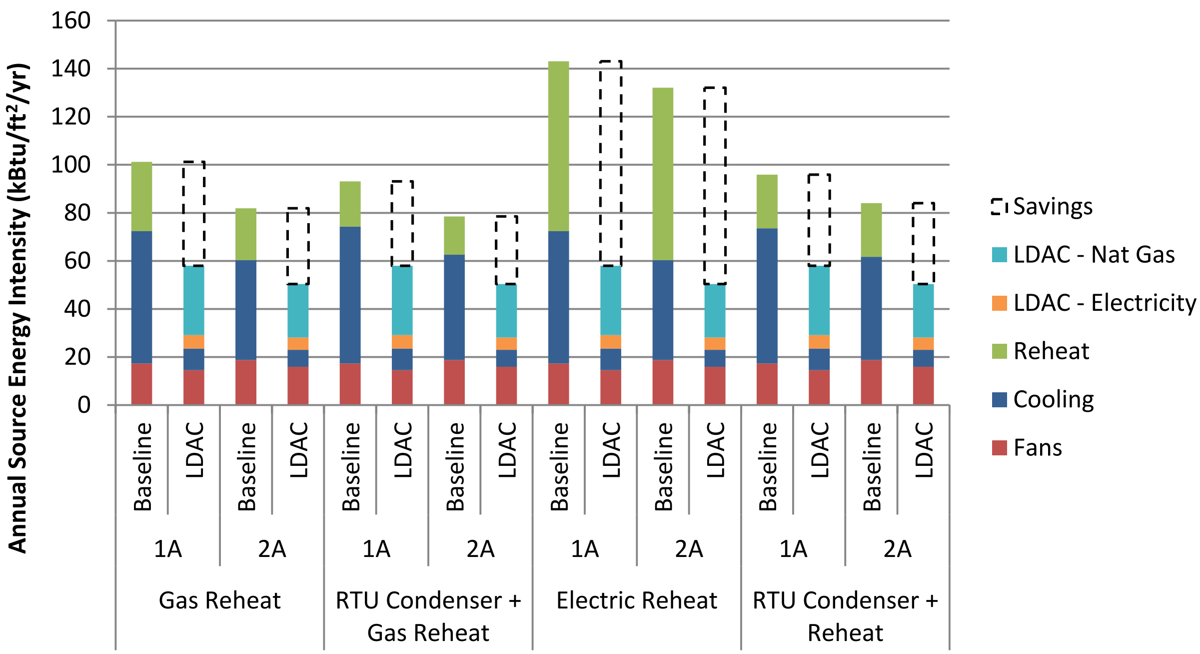

Figure 3–60 Annual source ventilation and air conditioning energy consumption and savings – two-stage regenerator (kBtu/ft^2/yr)
(Credit: Lesley Herrmann/NREL)

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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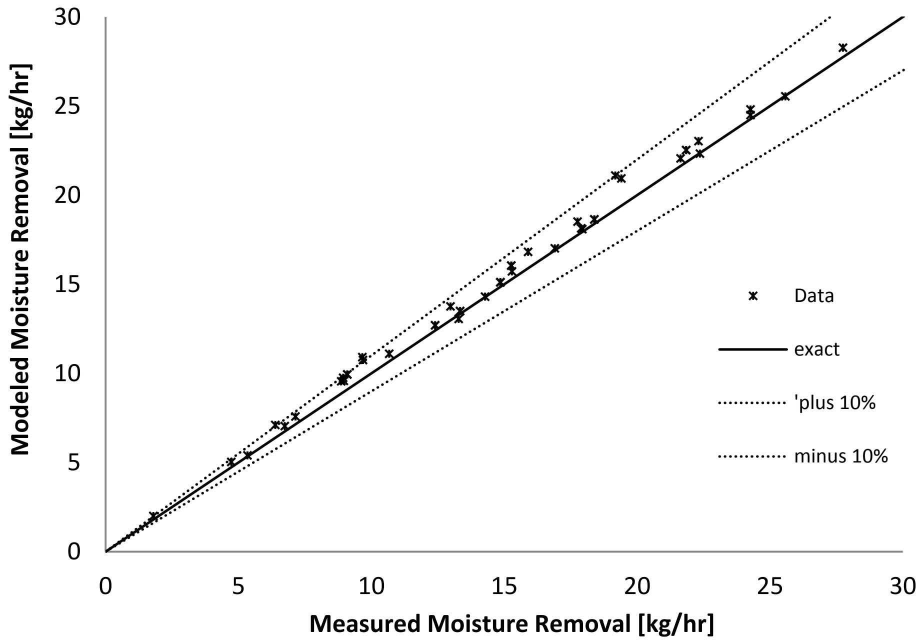

Figure A–1 Comparison of conditioner model and laboratory data showing good agreement

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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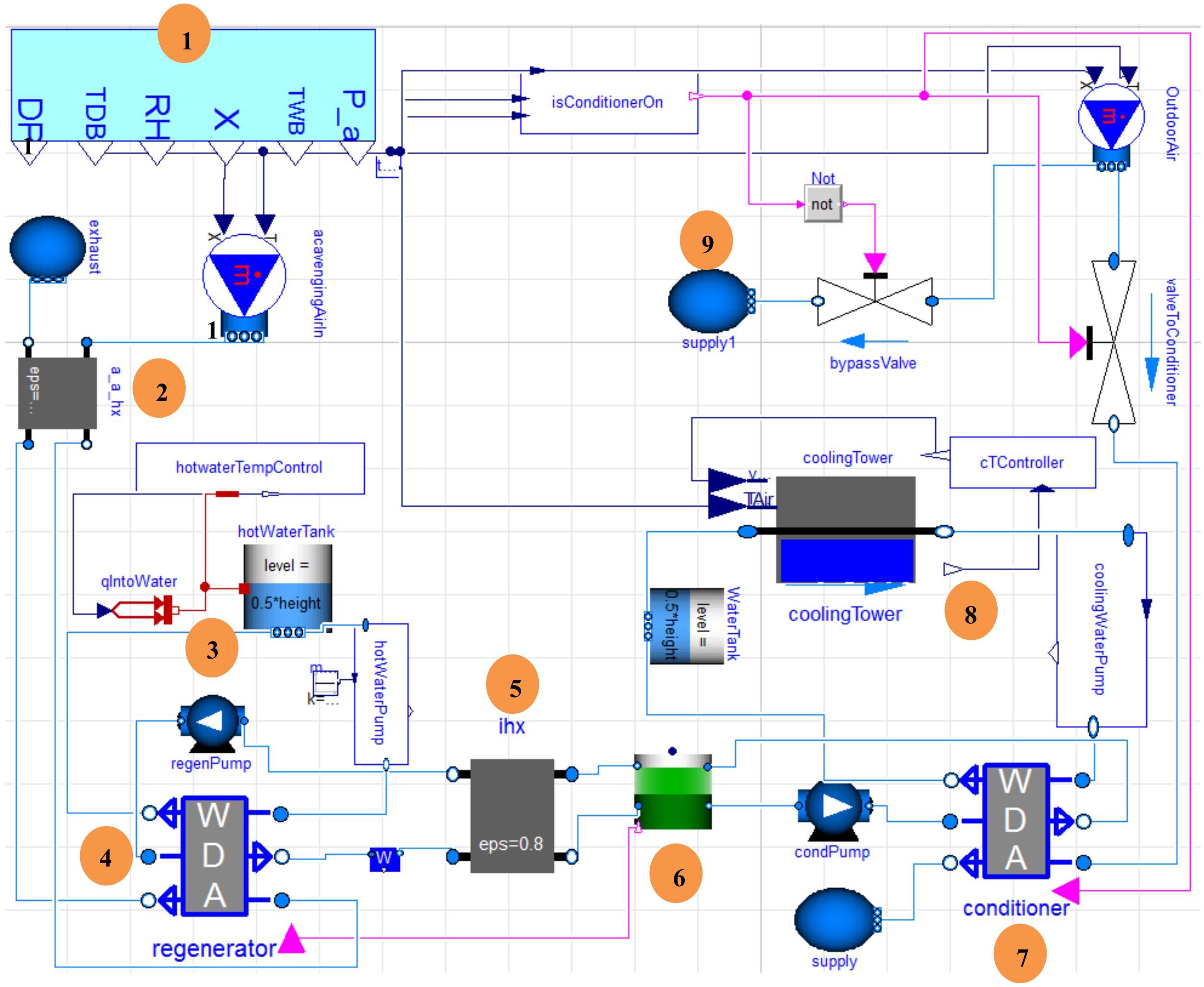

Figure A–2 Schematic of system-level LDAC model

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)

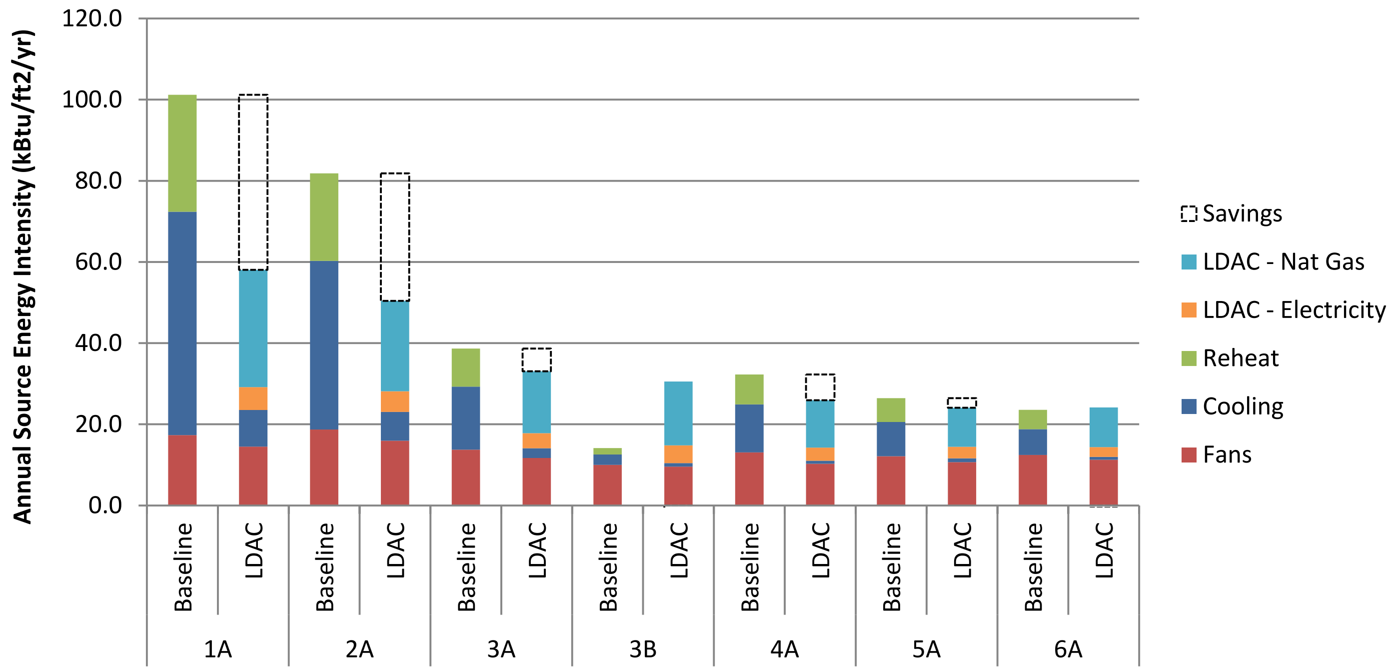

Figure B–1 Annual ventilation and air conditioning source energy intensity and savings – natural gas reheat coils – two-stage regenerator
(Credit: Lesley Herrmann/NREL)

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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

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

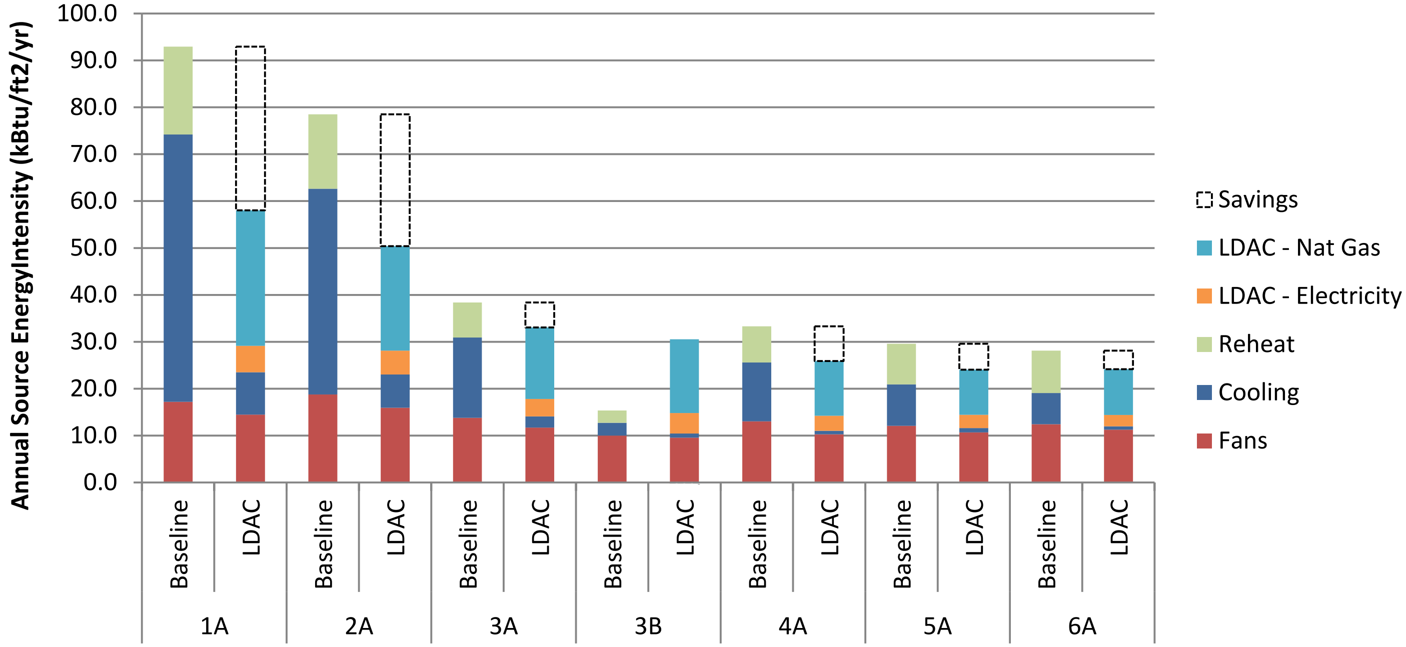

Figure B–2 Annual ventilation and air conditioning source energy intensity and savings – RTU condenser hot-gas reheat with auxiliary natural gas reheat coils – two-stage regenerator
(Credit: Lesley Herrmann/NREL)

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)

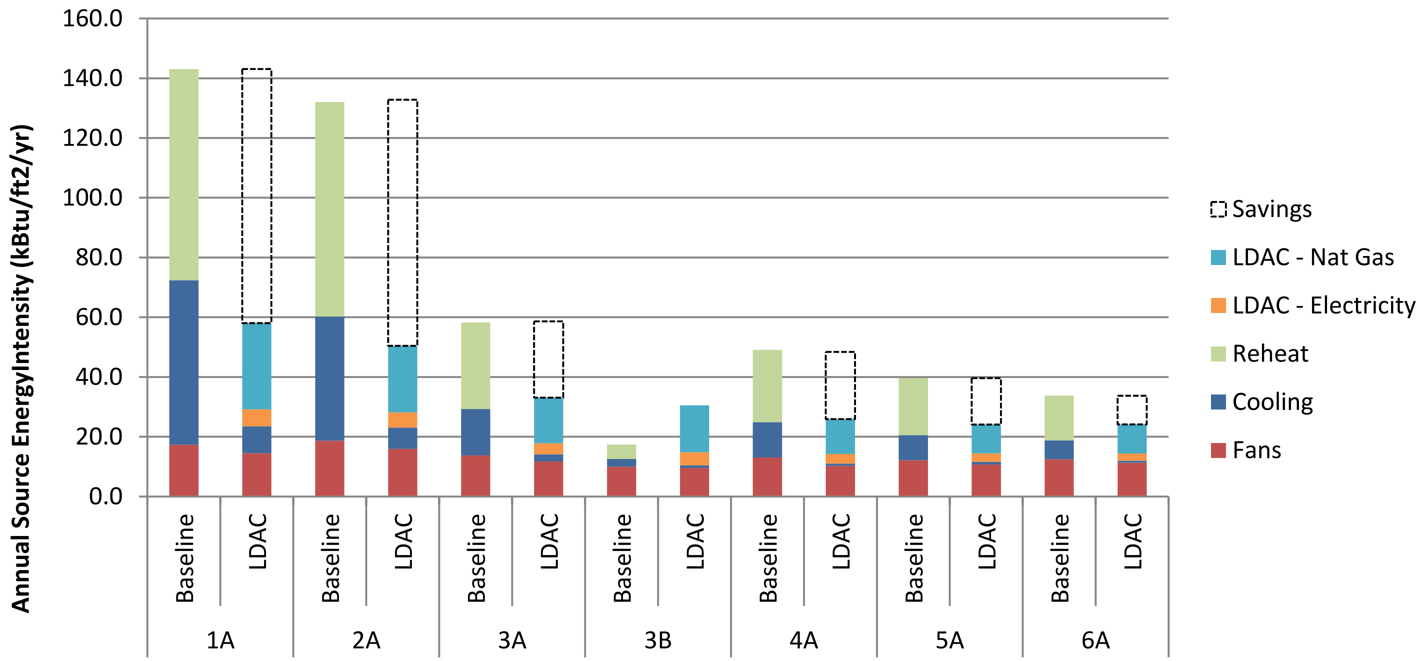

Figure B–3 Annual Ventilation and air conditioning source energy intensity and savings – electric reheat coils – two-stage regenerator
(Credit: Lesley Herrmann/NREL)

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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

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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/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	17.3	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)

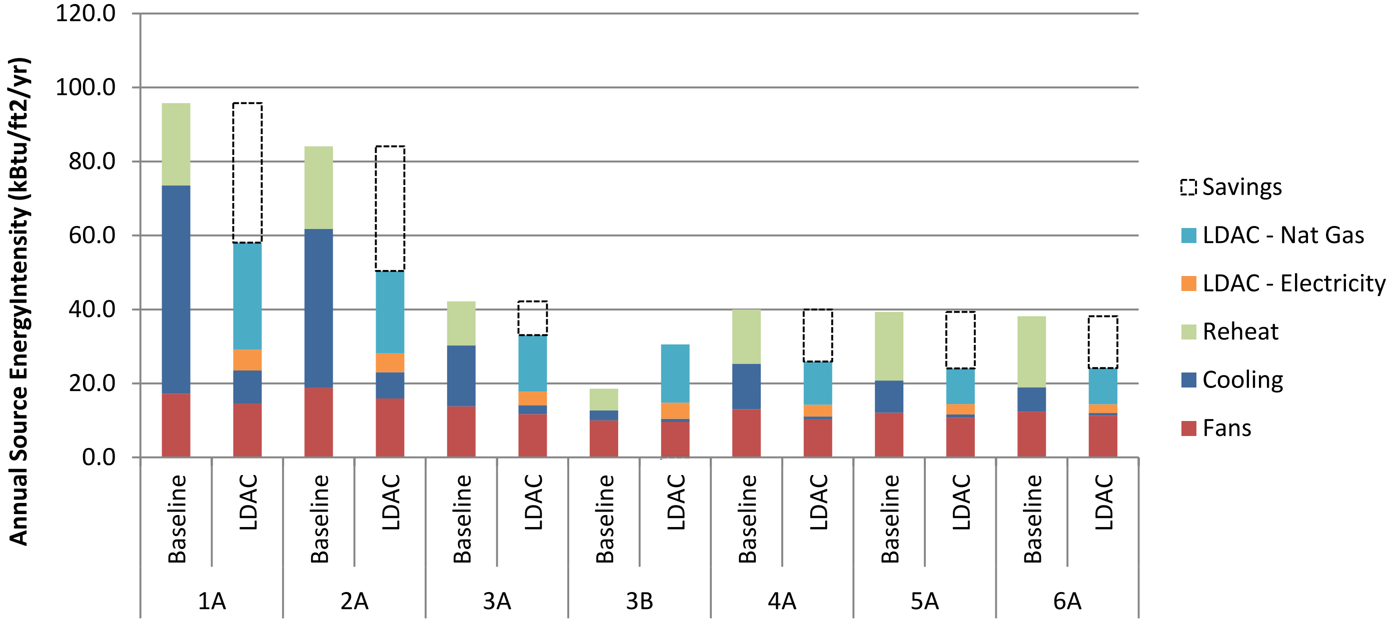

Figure B–4 Annual conditioning source energy intensity and savings – RTU condenser hot-gas reheat with auxiliary electric reheat coils – two-stage regenerator
(Credit: Lesley Herrmann/NREL)

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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/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	17.2	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

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