Arnold Schwarzenegger
Governor

LIQUID DESICCANT BASED INTEGRATED HYBRID REFRIGERATION TECHNOLOGY FOR ENERGY-EFFICIENT REFRIGERATED WAREHOUSES

PIER FINAL PROJECT REPORT

Prepared For:
California Energy Commission
Public Interest Energy Research Program

Prepared By:

University of California, Merced

January 2011
CEC-500-2010-041

Prepared By:
University of California
Gerardo C. Diaz, Roland Winston
5200 North Lake Road, Merced, CA 95343

Commission Contract No. 500-02-004

Prepared For:
Public Interest Energy Research (PIER)
California Energy Commission

Rajesh Kapoor
Contract Manager

Michael Lozano
Program Area Lead
Industry, Agriculture, & Water

Virginia Lew
Office Manager
Energy Efficiency Research Office

Laurie ten Hope
Deputy Director
ENERGY RESEARCH AND DEVELOPMENT DIVISION

Melissa Jones
Executive Director

Acknowledgements

This work has been supported by the California Energy Commission Contract # 500-02-004 and the California Institute for Energy and Environment Contract Number MR-07-04. The authors thank Scott Critchley at Aslan Cold Storage, LLC, and David Satterberg at Del Monte Foods for continuous support to this project and for allowing access to their facilities to measure realistic operating conditions inside a refrigerated warehouse. The authors also thank Jay Kliewer at California Controlled Atmosphere for providing essential information about the vapor compression system at Aslan Cold Storage, LLC, and its operating parameters.

Please cite this Report as follows:

Diaz, Gerado (UC Merced). 2010. *Liquid Desiccant Based Integrated Hybrid Refrigeration Technology for Energy Efficient Refrigerated Warehouses*. California Energy Commission, PIER Industrial/Agriculture/Water Research Program. CEC-500-2010-041.

Preface

The California Energy Commission's Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program conducts public interest research, development, and demonstration (RD&D) projects to benefit California.

The PIER Program strives to conduct the most promising public interest energy research by partnering with RD&D entities, including individuals, businesses, utilities, and public or private research institutions.

PIER funding efforts are focused on the following RD&D program areas:

- Buildings End-Use Energy Efficiency
- Energy Innovations Small Grants
- Energy-Related Environmental Research
- Energy Systems Integration
- Environmentally Preferred Advanced Generation
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy Technologies
- Transportation

Liquid Desiccant-Based Integrated Hybrid Refrigeration Technology for Energy-Efficient Refrigerated Warehouses is the final report for the project (Contract Number 500-02-004) conducted by the University of California, Merced, and AIL Research. The information from this project contributes to PIER's Industrial/Agricultural/Water Program.

For more information about the PIER Program, please visit the Energy Commission's website at www.energy.ca.gov/research/ or contact the Energy Commission at 916-654-4878.

Table of Contents

List of Figures

Abstract

California has about 475 million cubic feet of cold storage in refrigerated warehouses, consuming about 1.8 billion kilowatt hours or 20 percent of the total electricity in the food industry. Defrosting due to ice formation on the surface of cooling coils is a major source of inefficiencies in refrigerated warehouses. Reducing or completely eliminating defrosting cycles could decease the energy consumption of refrigerated warehouses by more than 10 percent depending on the type of application. Solid desiccants (substances that remove moisture) have been used to dehumidify the loading docks at refrigerated warehouses, and reduce moisture infiltration. They are not, however, suitable for direct elimination of frost from cooling coils.

This demonstration project extracted the humidity from the air inside a commercial refrigerated warehouse by using a liquid-desiccant-to-air heat and mass exchanger located upstream of the evaporator. The absorber core is designed to lower the dewpoint temperature of the air flowing through it to a temperature below the surface temperature of the cold storage evaporator. The desiccant will be regenerated by ambient air, and the controller is used to shut down the regenerators when the ambient relative humidity is above a specified limit.

A small-scale implementation of the proposed system was demonstrated at Aslan Cold Storage, LLC, in Kingsburg, California. This plant processes produce used in the Del Monte Foods canning facilities. The results show that ice formation on the surface of the evaporator can be slowed and in some situations eliminated completely.

Keywords: Liquid desiccants, refrigerated warehouses, defrosting, energy consumption reduction, cooling coils ice formation

Executive Summary

Introduction

California has about 246 refrigerated warehouses with approximately 475 million cubic feet of cold storage. The total refrigeration load in these warehouses is estimated at about 120,000 tons, the electrical power requirement (estimated at 3 kilowatts/ton) is about 360 megawatts, and the total power consumption based on 5,000 hours/year is 1.8 billion kilowatt-hours. This sector is responsible for about 20 percent of the total electric energy consumption of the food industry. Energy conservation in refrigerated warehouses can produce significant benefits to this industry.

Purpose

Significant energy and demand savings would be realized in refrigerated warehouses if defrosting the refrigerated coils could be minimized or eliminated. Several sources estimate the energy costs from frost accumulation on evaporators at between 10 percent and 25 percent. By eliminating defrosting, the warehouse temperature will be more uniform and steady which will improve product quality.

Desiccants (substances that remove moisture) have been successfully used to reduce defrosting losses in refrigerated warehouses. The most ambitious approach was a liquid-desiccant system engineered by Kathabar, Inc., and installed on a 25,000 square foot warehouse owned and operated by Newark Refrigerated Warehouses. This frost-free system replaced conventional finned-tube refrigeration coils with a bed of porous material called Celdek that was flooded with -25° F lithium chloride solution. Return air from the warehouse was cooled as it flowed through the flooded bed. Water vapor in the air was absorbed by the desiccant instead of building frost on the cold surfaces of a coil. The desiccant was continuously chilled in a plate-and-frame heat exchanger that functioned as the evaporator of the refrigeration system. A slip stream of desiccant was diverted to a regenerator where heat removed the absorbed water.

Results

In this project, a liquid-desiccant system was tested that has the potential to completely eliminate defrosting at refrigerated warehouses, while retaining the conventional finned-tube refrigeration coils that are now widely used. A low-flow liquid-desiccant absorber was used to lower the dewpoint of the air that enters the cooling coil to a value that is several degrees lower than the surface temperature of the coil. After absorbing water, the desiccant was regenerated to its original concentration by bringing the weak desiccant in contact with low relative humidity ambient air.

A small-scale liquid-desiccant system was installed at a commercial cold storage facility. Indoor air was dehumidified with the absorber, and the diluted liquid desiccant solution was regenerated with low relative humidity ambient air without using any external heat source. The results show that humid cold air inside the refrigerated warehouse could be dehumidified by lowering its dew point temperature to a value near or below the surface temperature of the cold storage evaporator. This technology has the potential to significantly reduce or even eliminate ice-formation on the surface of the cooling coil. With the proper maintenance related to air and liquid desiccant systems, the prototype showed no reliability problems although some signs of corrosion were found in some metallic components. This corrosion most likely occurred when desiccant droplets were entrained in the air stream during abnormal operation, for example, when a poorly supported air filter made contact with the core of the desiccant absorber.

Conclusion and Recommendations

The scale-up analysis of the technology shows that even with energy savings between 10 percent and 20 percent, the cost of the liquid desiccant system to process the entire air-flow rate inside the cold storage is not economically attractive as a retrofit system. However, if a liquid desiccant system is applied to a new warehouse then it may reduce the size and cost of the evaporator by significantly increasing the density of fins on the coil (e.g., 10 fins per inch (fpi) instead of 4 fpi). In this case, cost savings and energy savings for the evaporator could offset the cost for the liquid desiccant system.

1.0 Introduction

Refrigerated warehouses are predominantly operated using vapor compression systems. In these systems, air is passed through cooling coils to obtain a desired air exit temperature that can range from -25 to 34 °F, depending on the type of application. Due to the low surface temperature of the coil, frost builds up and blocks the free flow area of air passing across the coil. A defrosting process is then utilized to remove the ice from the cooling coils to restore the normal operation of the system. The defrosting operation can consist of running hot refrigerant through the cooling coils, or spraying water on the external surface. Due to infiltrated humid air, some warehouses have electrically heated doors to remove the ice that builds up on them.

The consequences of frost buildup are significant with respect to energy consumption in refrigerated warehouses. As the ice starts to block the face area of the evaporator, the fan power consumption increases, the cooling capacity of the evaporator decreases, and the overall coefficient of performance (COP) of the vapor compression system decreases. At this point, the warehouse operator or the built-in controls start the defrosting process. Using electrical heaters for de-icing purposes incurs the need for auxiliary electricity significantly. A low-temperature refrigerated warehouse that maintains a -20 °F storage space could reduce its energy use by 40 percent if it eliminated defrosting.

By eliminating defrosting, the temperature within the warehouse will be more uniform and steady, which will improve product quality.

California has about 246 refrigerated warehouses with approximately 475 million cubic feet of cold storage volume. The total refrigeration load in these warehouses is estimated at about 120,000 tons, the electrical power requirement estimated at 3 kilowatt/ton is about 360 megawatts, and the total power consumption based on 5,000 hour/year is 1,800 million kilowatt-hours. This sector is responsible for about 20 percent of the total electric energy consumption of the food industry. Energy conservation in refrigerated warehouses can produce significant benefits to the food industry.

This project proposes the utilization of a *liquid* desiccant to remove moisture content of the air upstream of the cooling coils. The system is designed to take the dew point temperature of the air passing across the coil to a value below its surface temperature. In our demonstration, outdoor air was used to regenerate the desiccant. No modifications to the original vapor compression system were required.

Solid desiccants have been utilized in the past to dehumidify the loading docks at refrigerated warehouses, thereby reducing moisture infiltration. However, they are not suitable for the direct elimination of frost from cooling coils. This project proposes to extract the humidity from the air in a localized manner. This is done by utilizing a liquid-desiccant-to-air heat/mass exchanger located upstream of the evaporator.

This project was demonstrated at a refrigerated warehouse located in the Central Valley in California. UC Merced together with industrial partner's AIL Research, Inc., Del Monte Foods, Inc., and Aslan Cold Storage, LLC, participated in this project.

1.1. Project Goals and Objectives

The goal of this project is to reduce the energy consumption of a refrigerated warehouse by 10 percent.

The objectives of this project are:

- Reduce the frost formation in the cooling coils of a refrigerated warehouse.
- Significantly reduce or eliminate the number of defrost cycles in a refrigerated warehouse.
- Integrate a liquid-desiccant system to an existing vapor compression system used to refrigerate a warehouse.
- Monitor the performance of the system in a refrigerated warehouse.
- Perform cost analysis.

2.0 Description of the Project

The purpose of the project was to demonstrate that a liquid-desiccant system could reduce or completely eliminate defrosting at refrigerated warehouses. A low-flow liquid-desiccant absorber was used to lower the dewpoint of the air that enters the cooling coil to a value that is several degrees lower than the surface temperature of the evaporator.

2.1. Low-Flow Liquid-Desiccant Technology

Low-flow liquid-desiccant technology refers to a concept that AIL Research developed and patented in 1994 in which desiccant flow rates are reduced to between 1/20 to 1/50 the rates commonly used in industrial liquid-desiccant systems. In a seven-year, $5 million development program sponsored by the U.S. Department of Energy and the National Renewable Energy Laboratory, AIL Research has developed low-cost, manufacturable components to implement the low-flow technology.

As shown in Figure 1, a liquid-desiccant air conditioner that uses the low-flow technology has three main components: the conditioner, the regenerator, and the interchange heat exchanger. The conditioner is a parallel-plate liquid-to-air heat exchanger. A coolant, typically cooling tower water (but possibly water from a geothermal well, lake, or chilled water loop), flows within the plates and a very low-flow of liquid desiccant flows down the outer surfaces of the plates. Thin wicks on the plate surfaces create uniform desiccant films. The air to be processed flows horizontally through the gaps between the plates. As this humid air comes in contact with the desiccant, water vapor is absorbed. The heat released by this absorption is transferred to the coolant. The air leaves the conditioner drier and at a much lower temperature.

The diluted desiccant that leaves the conditioner is pumped to the regenerator. The regenerator has the same configuration as the conditioner: a parallel-plate liquid-to-air heat exchanger. Again, very thin films of desiccant flow in wicks on the outer surfaces of the plates, and air flows in the gaps between the plates. For the regenerator, however, a hot heat transfer fluid flows within the plates. This hot fluid can be supplied by solar thermal collectors or other energy sources. In the proposed application, low relative humidity ambient air was used to regenerate the liquid desiccant.

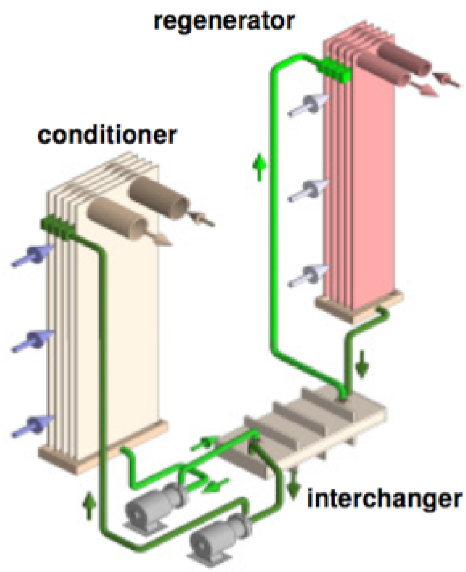

Figure **1. Low-flow liquid-desiccant components in a system**

Source: UC Merced

The 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. In both the regenerator and conditioner, the flow rate of desiccant is so low that the falling films on the plates are contained completely within the thin wicks. The air velocity over the films is far too low to entrain desiccant droplets. Since both the desiccant delivery to and collection from the plates are done without creating droplets, desiccant does not carryover from either the regenerator or the conditioner.

2.2. Description of Facilities

Aslan Cold Storage, LLC, located at 1045 Simpson Street, Kingsburg, California, was the selected site. The peaches stored in this facility are sent to Del Monte plant # 25 located at 1101 Marion Street, Kingsburg, California. Other types of fruits, such as grapes, pears, blueberries, etc., are also handled in this facility. There are five refrigerated rooms that are used for short- or long-term cold storage of produce. Room 2 was selected for this particular project. Figure 2 shows the location of Room 2 within the facilities.

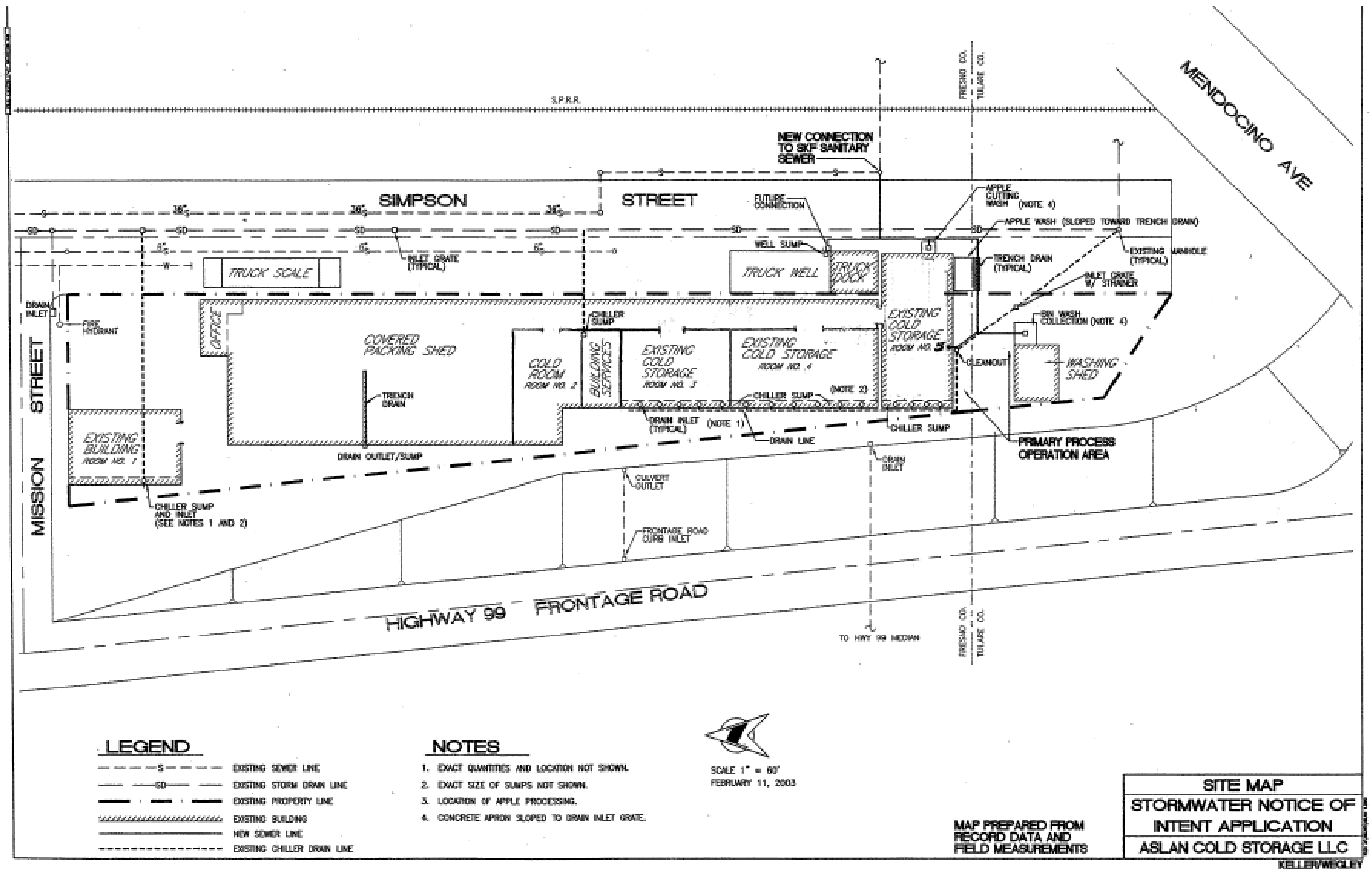

Figure 2. Cold storage site

Source: UC Merced

The 83x41 ft^2 (25x13 m^2) room has six 37″ (0.94 m) fans, and each fan utilizes a 10 hp Baldor Standard-E motor. There are two temperature sensors, one Ammonia gas detector, and one temperature/relative humidity sensor in this room. Figure 3 shows the interior area of Room 2 and some of the relevant dimensions.

Figure 3. Room 2 at Aslan Cold Storage, LLC.
Photo Credit: UC Merced

Room 2 uses a wall arrangement, and six prop fans draw air from the room through an opening at the bottom of the bunker wall and over two ammonia evaporators that are located behind the wall. The fans discharge the cooled air into the upper section of the room, and a combination of forced convection (from the fans) and natural convection (from density differences) distribute the cooled air throughout the room. The face area of each ammonia coil is approximately 15′ x 4′ (4.6x1.2 m^2), and the design velocity of the air at the face of the coils is 600 feet per minute (3.05 m/s). This implies a design air flow of 72,000 cubic feet per minute (34 m^3/s). Figure 4 shows the section above the coils in Room 2. The air inside the room is kept between 32 and 34 ^oF (0 and 1 ^oC), and the surface area of the coils is kept between 20 and 25 ^oF (-6.7 and -3.9 ^oC). The defrost cycle is triggered by a timer every 24, 12, or 8 hours. Figure 4 shows the defrost water piping above the two ammonia coils.

Five compressors of different capacities are used for all the rooms in the facility. There are two 60hp, two 125hp and one 400hp units. As shown in Figure 5, the condensers are located at the back of the facility.

Figure 4. Area above ammonia coils

Photo Credit: UC Merced

Figure 5. Condensers

Photo Credit: UC Merced

The expansion valve that allows flow of the ammonia refrigerant to the coils in Room 2 is located right behind the room. It is connected to a relay that is operated automatically from the control room that is located right next to Room 2 and near the compressors area. Figure 6 shows the location of this valve.

Figure 6. Ammonia valve

Photo Credit: UC Merced

The maintenance of the equipment is currently subcontracted to California Controlled Atmosphere located in Dinuba, California.

2.3. Sensor Location and Data Acquisition System

The proposed project included a first phase intended to collect data from the current operating conditions of the facility. The test data collected was used to improve the design of the liquid desiccant. The following list provides detailed information of each of the sensors used in this first phase.

1. Obvius Data Acquisition (DAQ) sensor
    a. Model AcquiSuite A8812
    b. DA Server
    c. Serial number AA10F3

2. Humidity and temperature transmitter
    a. Vaisala
    b. Serial number C2940001
    c. Range: -40° C to 60° C (output signal 0 to 10V)

        i. 0 to 100%RH (output signal 0 to 10V)
   d. Measurement: Inlet air RH and T

3. Temperature sensor
   a. Veris
   b. Model: AA10
   c. Range: 0 to 100°C (output signal 4 to 20mA)
   d. Measurement: Inlet water temperature

4. Temperature sensor
   a. Veris TW Series
   b. Range: 0°C to 50°C (output signal 4 to 20mA) Measurement: Room temperature

5. Flow meter
   a. Omega
   b. Model FP-5300
   c. Range: 0 to 200 gpm (output signal: frequency value)
   d. Measurement: Water flow rate

6. Frequency/pulse signal conditioner
   a. Omega
   b. Output range: 4 to 20 mA
   c. Measurement: Signal conditioner for flow meter

7. Digital resistance temperature detectors (RTD)
   a. Range: -18.05 to 537.5°C (output signal 0 to 10V)
   b. Resistor 3K ohm
   c. Measurement: Outlet water temperature

8. Temperature and humidity sensor
   a. Veris
   b. Model: T Series RTD/Thermistor
   c. Measurement: Ambient temperature and relative humidity

9. Temperature sensor
   a. Veris
   b. Model: AA10
   c. Range: -29 to 69°C (output signal 4 to 20mA)
   d. Measurement: Coil surface temperature

10. Differential pressure transducer

a. SETRA
b. Model 264
c. Range: 0 to 0.25 in.w.c (4 to 20 mA)
d. Measurement: Air pressure drop

11. Current monitoring
    a. Veris
    b. Model: Hawkeye 300 (H300)
    c. Range: 0.15 to 60A
    d. Measurement: Ammonia valve status

The sensors were mounted at different locations within the facility. The range of operation of each sensor was determined considering the operating conditions and location within the system. Figure 7 shows a schematic with the location of the sensors inside or near Room 2.

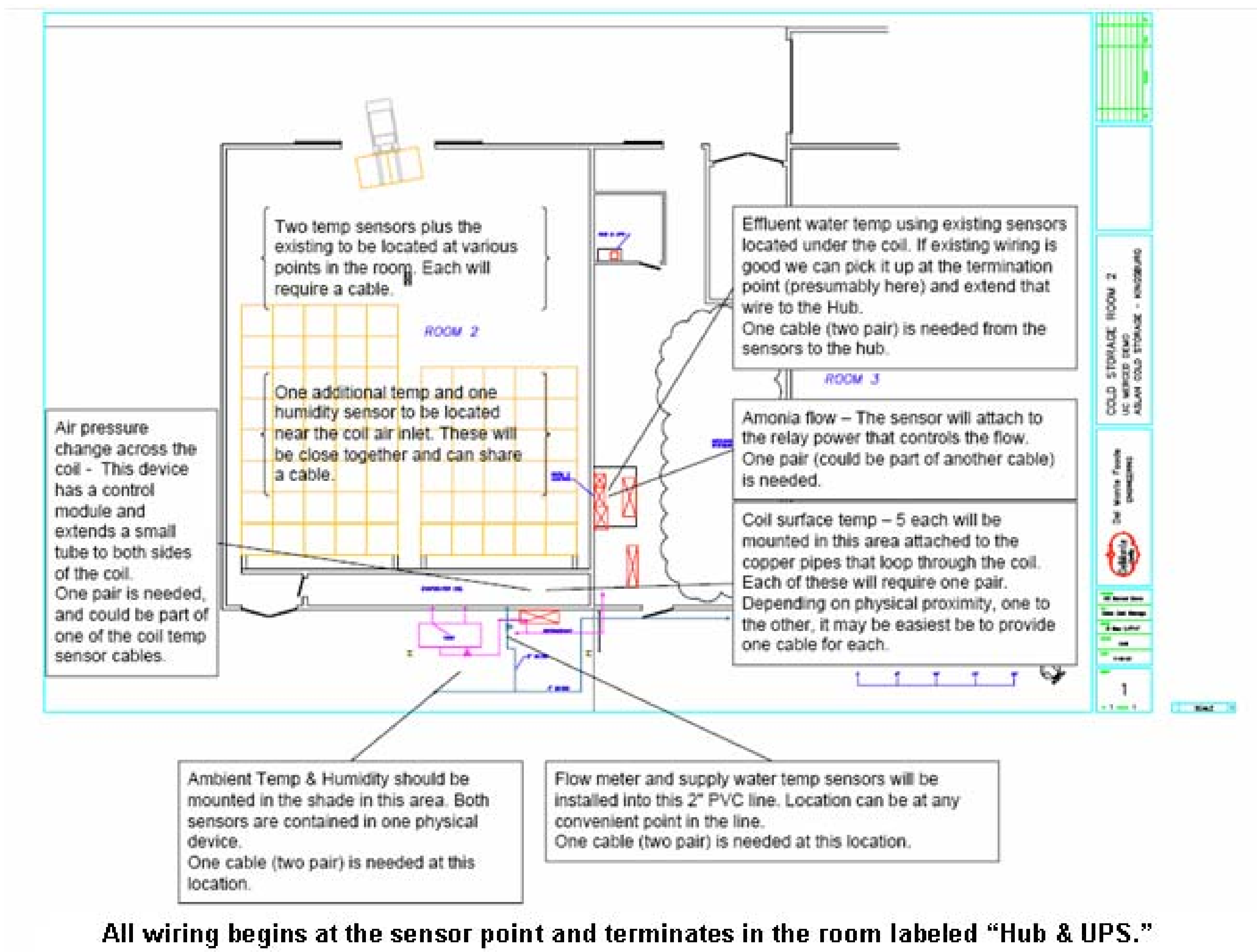

Figure 7. Location of sensors

Source: UC Merced

2.4. Simulation of Absorber/Regenerator Performance

AIL's computer model for a low-flow liquid-desiccant absorber was developed to simulate the performance of a water-cooled absorber and was modified so that it could predict the performance of the pleated-plate absorber. The original model calculates the heat and mass transfer that occurs when air flows in the gap between two parallel, desiccant-wetted walls that are water-cooled. The air flows horizontally, the desiccant flows downward, and the water flows in a two-pass circuit that is roughly counter flow to the air. The desiccant films are assumed to be well -mixed as they flow down the walls. The two walls and air gap are divided into rectangular elements that form an N x M matrix (where N and M are between 3 and 10), and energy and mass balances are performed on each element. Heat and mass transfer rates between the desiccant films and the air flow are calculated using published correlations for the Nusselt Number and its mass-transfer analog.

The preceding model was run with parameters that insulated the desiccant films from the cooling water circuit (i.e., the thermal resistance of the walls were set at very high values). With this change, the model simulates the adiabatic absorber that will be used in the refrigerated warehouse.

The modified model of the absorber was used to predict the performance of the absorber under operating conditions within the warehouse. An absorber with 100 plates that processed 833 cfm was modeled with air entering the absorber at 1°C (34° F) dry-bulb and 0° C (32° F) dew point.

As shown in Figure 8, the dew point of the processed air was reduced from 0°C (32° F) to between -6°C (21° F) to -8°C (17° F) as the calcium chloride concentration increased from 39 percent to 43 percent. In order to keep the ammonia evaporator within the warehouse free from frost, the processed air should have a dew point lower than the surface temperature of the evaporator. Based on an early review of the data collected at Room 2, the surface temperature of the ammonia evaporator is between -5°C (23° F) and -1°C (30° F) during normal operation. Based on the modeling results shown in Fig. 8, an absorber that operates with calcium chloride at 40 percent concentration should keep the dew point of processed air at least -16°C (3° F) below the evaporator surface temperature.

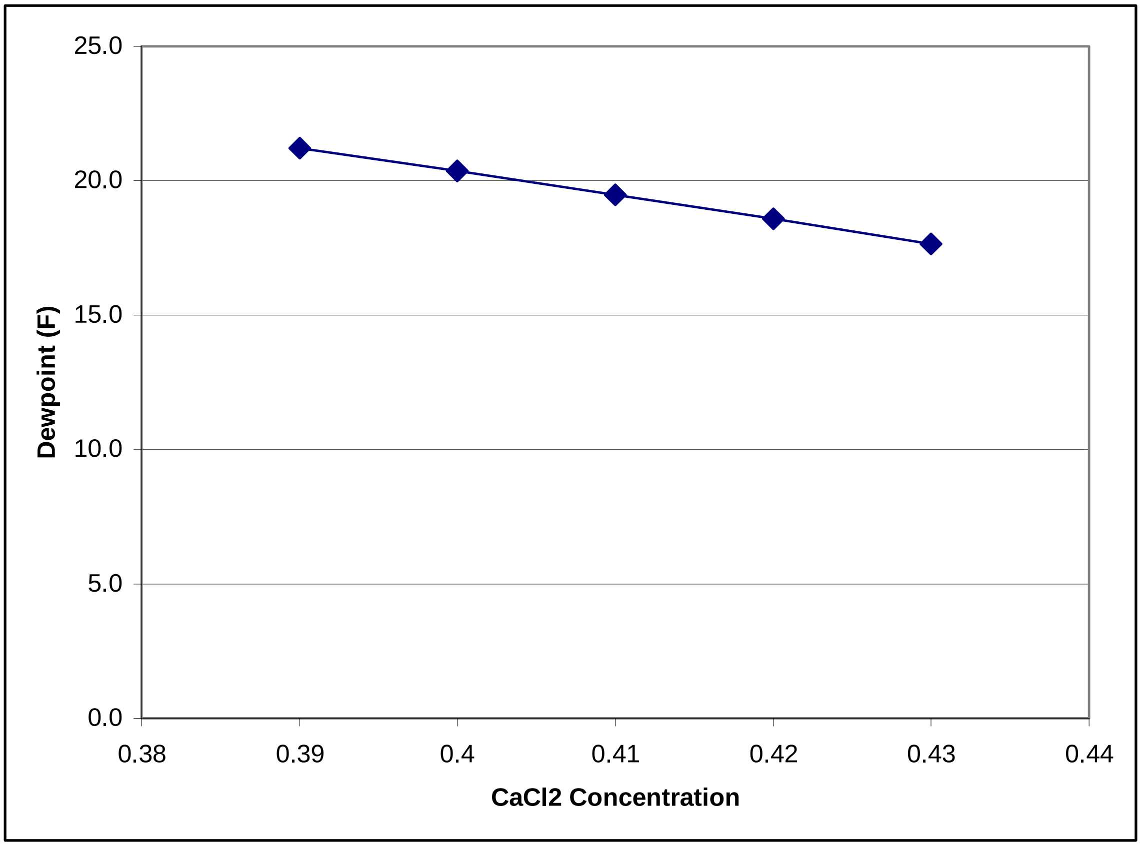

Figure 8. Final dew point of processed air as function of desiccant concentration

Source: UC Merced

The water removal rate for a 100-plate absorber is shown in Figure 9 as a function of desiccant concentration. As shown in this figure, the absorber will remove 6 lb/h of water when operating with a 40% calcium chloride solution.

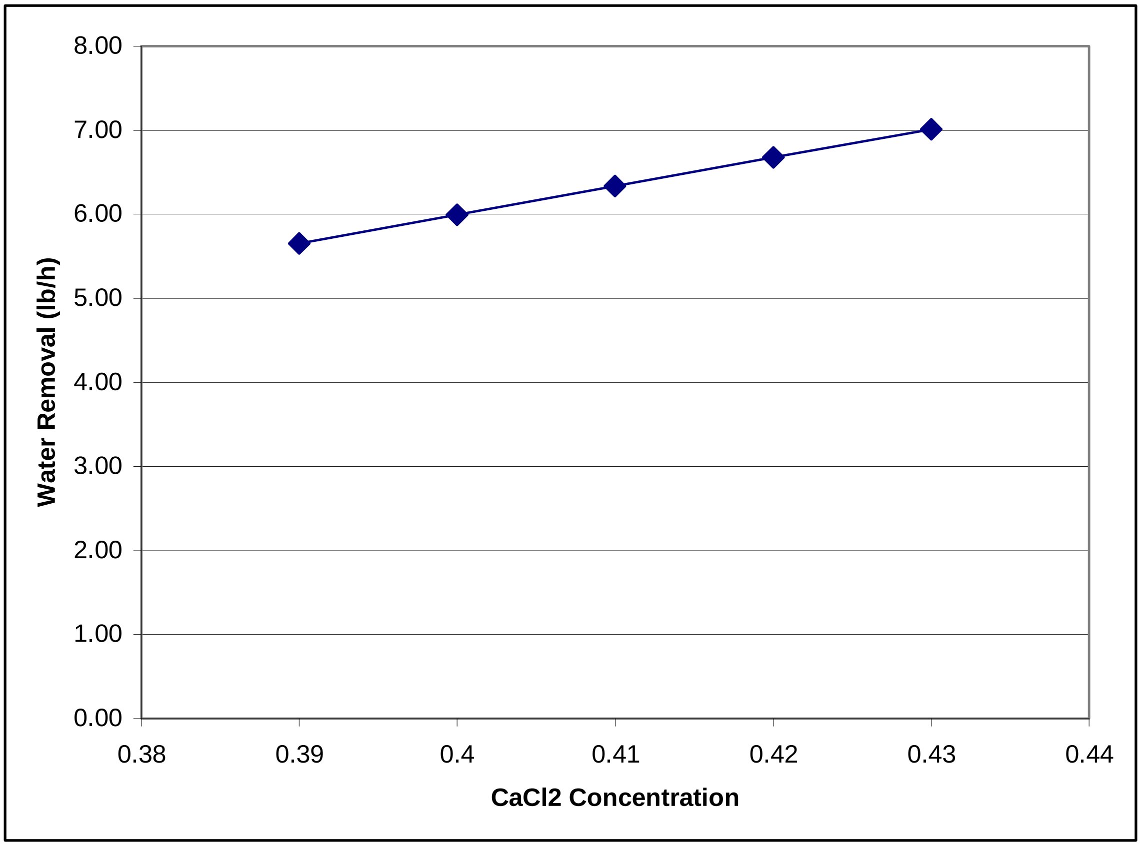

Figure 9. The water removal rate for a 100-plate absorber as a function of desiccant concentration.

Source: UC Merced

2.5. Liquid Desiccant Components Manufactured

The following components were manufactured by AIL Research for the integration of the liquid desiccant system.

2.5.1. Absorber/Regenerators

The same design was used for the absorber and regenerators. Figure 10 shows the manufactured core, which contained 100 pleated plates that provided heat/mass exchange surface, as well as, structural integrity to the core. Since only ambient air was used to regenerate the liquid desiccant, two of these cores were used as regenerators to increase the transfer surface between the ambient air and liquid desiccant. Only one core was used as the absorber.

Figure 10. Absorber/regenerator design

Photo Credit: UC Merced

The other heat exchanger obtained from AIL Research corresponds to the internal heat exchanger (IHX) that transfers energy between the warm/concentrated and the cold/diluted streams of liquid desiccant. This IHX is shown in Figure 11.

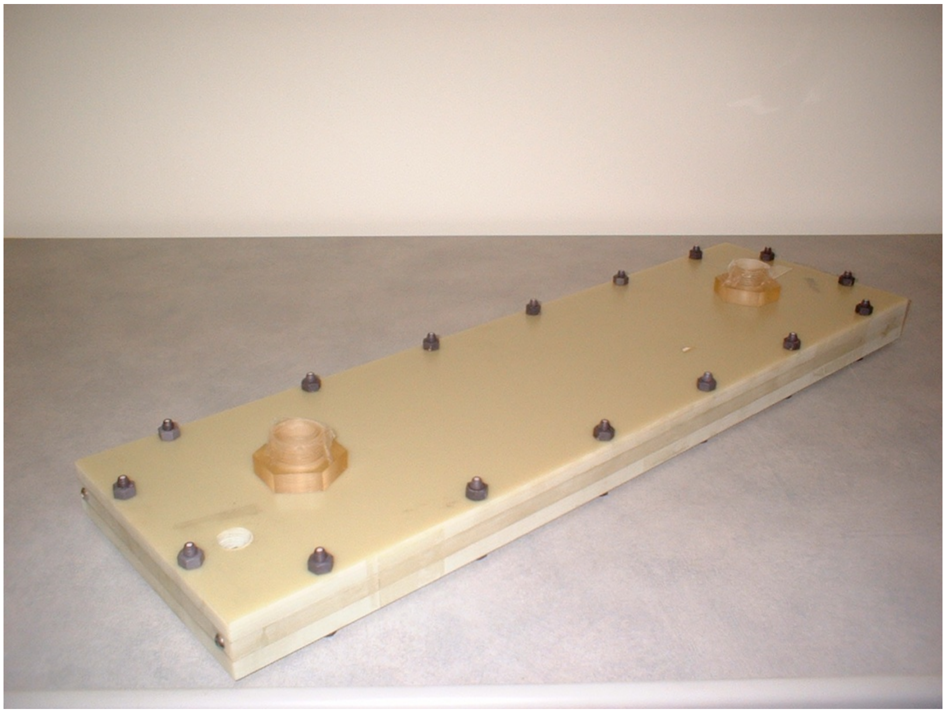

Figure 11. Internal heat exchanger

Photo Credit: UC Merced

2.6. System Integration

The components have been integrated into an indoor (absorber) and an outdoor (regenerator) stand that are connected by means of a PVC piping system. The storage tank sits in the middle and is also connected by means of PVC pipes. Figure 12 shows a schematic of the overall integration.

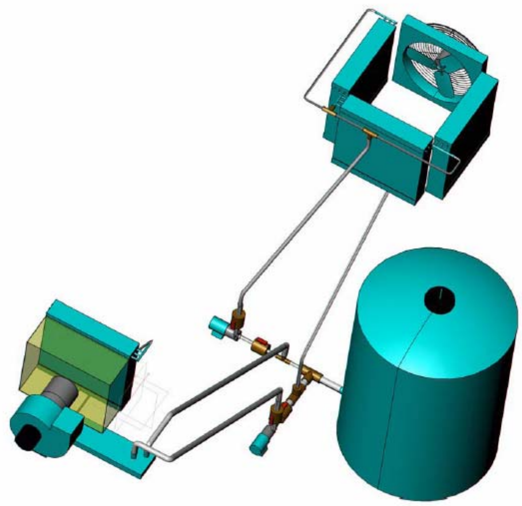

Figure 12. Integration of the absorber/internal heat exchanger (IHX), regenerator, and storage tank subsystems

Source: UC Merced

The absorber is situated above the internal heat exchanger (IHX) so that the outlet of the absorber feeds into the low-pressure inlet of the IHX by gravity. The warm concentrated liquid desiccant coming from outside of the room is cooled in the high-pressure side of the IHX before entering the inlet of the absorber. The cooling effect is performed by the cold and diluted stream of liquid desiccant coming from the outlet of the absorber. This diluted stream flows back to the storage tank area and is eventually circulated through the regenerators where moisture is removed from the liquid desiccant increasing its concentration. The flow of liquid desiccant to the absorber and regenerators is generated by separate pumps.

As shown in Figure 12, downstream of the absorber there is a polycarbonate plenum attached to a blower. Two temperature/relative-humidity sensors are located downstream of the absorber as well as four resistance temperature detectors (RTDs) that measure the temperature at the four quadrants. Inlet temperature and relative humidity were measured upstream of the absorber. The operation of the liquid desiccant system can be seen observed in the schematic shown in Figure 13.

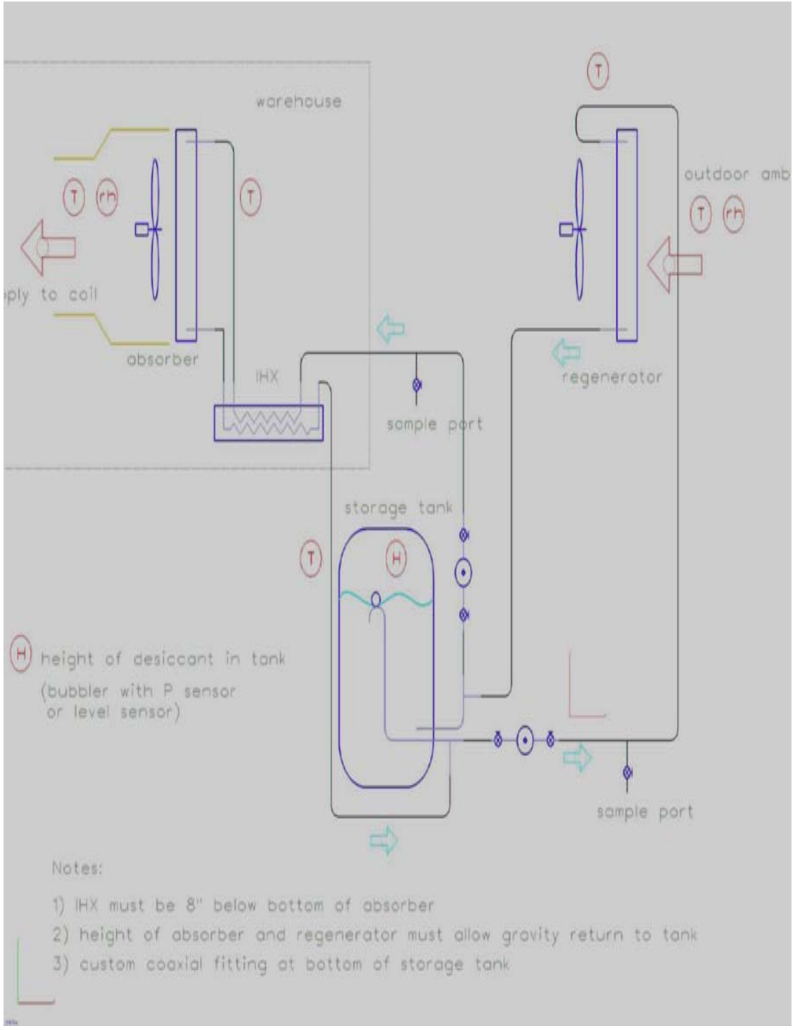

Figure 13. Operation of liquid desiccant system

Source: UC Merced

2.7. Test Plan

The test and monitoring plan included three main topics related to (a) performance operation of the refrigerated warehouse, (b) performance operation of the liquid desiccant system, (c) reliability of the liquid desiccant system.

a) Performance operation of the refrigerated warehouse

The monitoring and test plan corresponded to obtaining data to characterize the regular operating conditions inside Room 2 at Aslan Cold Storage, LLC. Since the operation of this room is determined by Aslan Cold Storage, no particular test has been designed. Avoidance of disruptions of the regular operation at the cold storage was a main goal. Data was obtained discontinuously from December 2007 to July 2009. This long period of monitoring allowed the collection of test data related to the operation of the refrigerated warehouse under different weather conditions and type of fruit stored. The variables to be monitored over time included: ammonia valve operation (on/off), inlet coil surface temperature, outlet coil surface temperature, inlet defrosting water temperature, outlet defrosting water temperature, room temperature, evaporator inlet air temperature and relative humidity, air pressure drop across evaporators, ambient temperature and relative humidity, and flow rate of defrosting water.

b) Performance operation of the liquid desiccant system

The monitoring and test plan of the liquid desiccant systems covered a shorter period of time with respect to the monitoring of the facility. The liquid desiccant system was fully operational at the beginning of June 2008. The system was completed and fully integrated during May of 2008 so some test data were collected early in the season until a leak developed. After the system was fixed, the collection of data has continued with no interruption during the 2008 season. The system was operated again between May and July 2009.

The testing of the system mainly related to recording its performance under a variety of weather conditions and operating procedures within the cold storage facility. One of the tests run corresponded to shutting down the absorber side and letting the liquid desiccant level at the tank go down due to the effect of the regenerators. This allowed the quantification of the amount of water being rejected by the regenerators. The test was performed under hot weather conditions. The inverse test, i.e. turning off the regenerators and leaving only the absorber working, was also performed during hot weather conditions. This test allowed the quantification of the performance of the absorber in terms of extracting moisture from the ambient. The variables to be monitored over time included: liquid desiccant flow rate, inlet air temperature and relative humidity upstream of the absorber, outlet air temperature and relative humidity downstream of the absorber, inlet liquid desiccant temperature to the IHX, inlet liquid desiccant temperature to the absorber, outlet liquid desiccant temperature from the absorber, liquid desiccant level inside storage tank, and operation of regenerators (on/off). All these variables were collected by a data acquisition system sampling data at a rate of one sample per minute. Twenty-six channels were simultaneously sampled every minute. The data was sent to a server and could be accessed remotely by members of this project.

c) Reliability of the liquid desiccant system

The reliability of the liquid desiccant system is addressed by collecting data about its operation over a long period of time. The type of information recorded includes malfunctions, leaks, amount of maintenance, replacement of air filters, replacement of liquid desiccant filters, component failures, and performance degradation. The period of operation of the system for the analysis of its reliability covered from May 2007 to July 2009. This system was open to the flow of external air, and so the effect of dust on the liquid desiccant was analyzed. Air filters were located upstream of the absorber and regenerators. Even with these filters some amount of dust reached the desiccant. This is an important factor since it is not easy to replicate in a laboratory experiment. For instance, the indoor air inside the warehouse contains dust from the external air, exhaust gasses from the forklifts, and any vapors emanating from the stored fruit, which could contain some sulfur, as is the case in grapes.

3.0 Results

3.1. Operating Conditions Inside Cold Storage

The operating conditions inside the cold storage are presented in the next subsections

3.1.1. Water Consumption

A number of cold storage facilities run their coils between -6.7 to -3.9^{o}C (20 to 25^{o}F). During the peach season, bins with produce are taken in and out of the facility during the day. These operating conditions allow ambient air into the refrigerated warehouse. Quite often ambient air conditions can reach temperatures above 38^{o}C (100^{o}F) with relative humidity values between 20 to 35 percent. This situation increases the load on the evaporator, which has to maintain the design temperature inside the cold storage. Moreover, since the cooling coil is operating at temperatures below freezing, ice forms at the surface of the evaporator. Cold storages tend to run between one to three defrosting cycles during the day, depending on a variety of operating conditions. The defrosting cycle consists of spraying water on the evaporator until the ice has melted. Spraying water creates a mist inside the cold storage that helps keep a high moisture level (above 75 percent) inside the room and maintains the quality of the produce. Figure 14 shows the formation of ice on the surface of the evaporator.

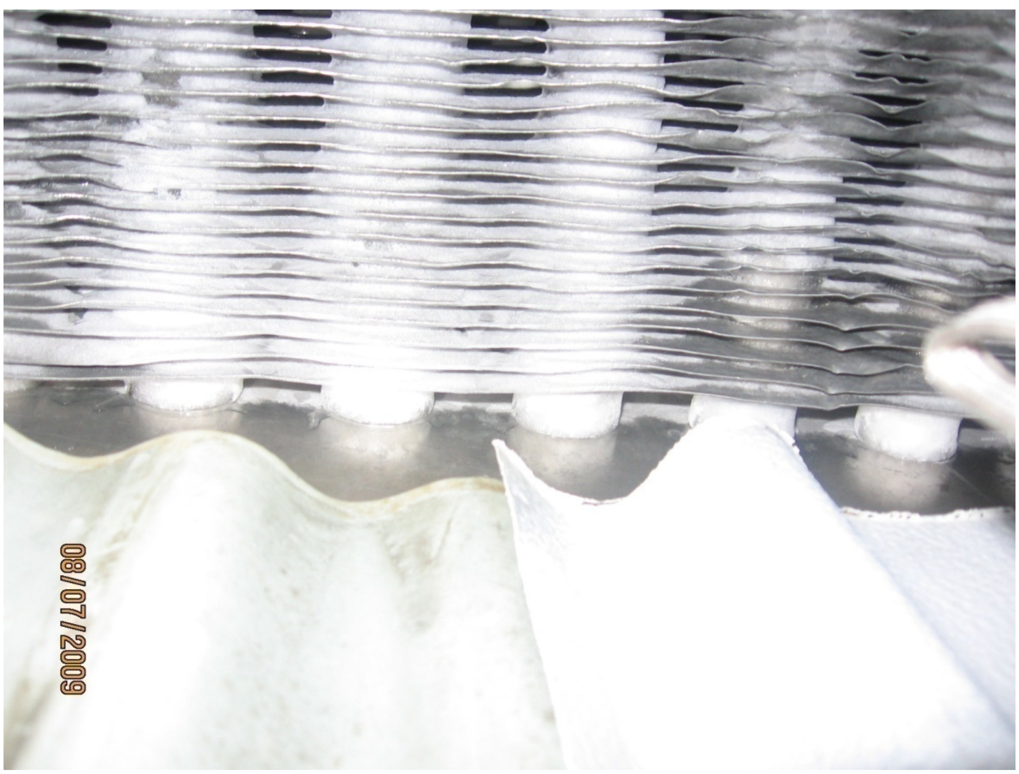

Figure 14. Ice formation on evaporator surface

Photo Credit: UC Merced

Sensors were installed to measure the amount of water used during a defrosting cycle. Figure 15 shows a typical defrosting cycle at the cold storage.

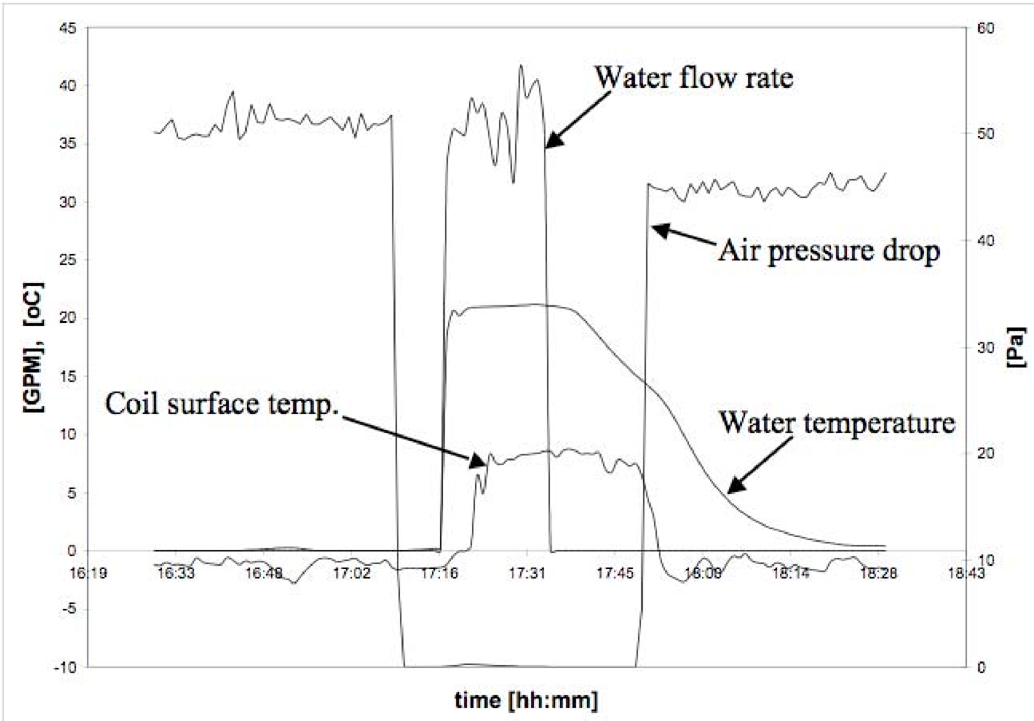

Figure 15. Flow rate, coil temperatures, and air pressure drop during defrosting

Source: UC Merced

The units were chosen to show variables of flow rate, temperature, and pressure drop plotted together on the same chart. The line indicated as "Water flow rate" shows the amount of water sprayed to melt the ice. Integration of the curve shows that more than 2,200 liters (600 gallons) of water were used per defrosting cycle. This is a significant amount of water since, in general, there are one to three defrosting cycles per day being run for each cold storage room. The line labeled "Coil surface temp." shows the temperature measured at the surface of the evaporator. It is clear that, after the defrosting cycle, the coil surface temperature increased due to the temperature of the water that was sprayed on it. This is much higher than the designed temperature of operation; so that the vapor compression refrigeration system needs to work harder to lower the temperature back to designed conditions. The line labeled "Air pressure drop" shows the value of the air pressure drop across the evaporator and provides an indication of the amount of ice formed on the surface. It is seen that the air-pressure drop before and after the defrosting cycle is different. The data in this figure were taken during November when low air-flow rates are experienced (usually one out of six fans is working). The change in pressure drop is quite significant if data from June or July are plotted, as shown in Figure 16. The red line corresponds to the air pressure drop across the evaporator. It is observed that ice formation is not regular since the air pressure drop varies significantly between defrosting cycles. The defrosting cycle is indicated by the time when the fans are turned off, which translates to zero air-pressure drop. It is seen that right after the defrosting cycle, the air pressure drop value is near 1 inch of water, and it increases until it reaches 3 inches of water, right before the next defrosting cycle is performed. This coincides with the increase in the value of the relative humidity of the air inside the cold storage. From a value of about 80%, the peaks shown in the blue curve reach magnitudes near 100 percent during the defrosting cycle. After the defrosting cycle is completed, the relative humidity goes back down until the next defrosting cycle is performed. Depending on the time of the season and type of produce being stored, the maximum value of the air pressure drop can vary significantly, since at the peak of the season six fans are used to circulate the air inside the room, and at the end of the season, only one fan may be utilized.

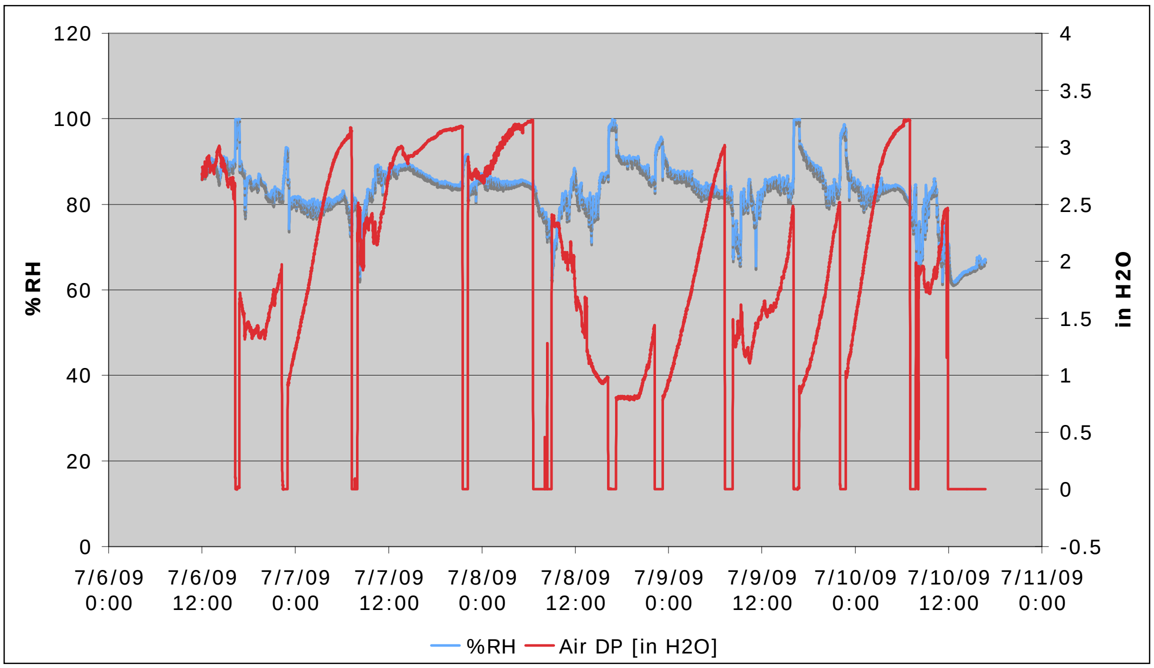

Figure 16. Change of air pressure drop across the evaporator and air relative humidity inside the cold storage

Source: UC Merced

3.1.2. Temperature Inside the Cold Storage

A number of temperature sensors were installed inside the cold storage to monitor the temperature distribution inside the room. Figure 17 shows the values of ambient temperature and relative humidity compared to temperatures measured inside the cold storage. The measurement indicated by T_room_1 is near the ceiling at the center of the room while T_room_2 is located on a wall near the ceiling. T_air_in is located upstream of the absorber for the liquid desiccant system. The measurements correspond to the period from July 6 to July 10, 2009. It is observed that the ambient temperature varies significantly during the day and night covering a range from 20° C to 35° C (68° F to 95° F). Ambient relative humidity, shown in red, covers a range between 15 percent to 60 percent. This has a significant impact in the operation of the liquid desiccant system since moisture from the cold storage room cannot be rejected at the regenerators at high ambient relative humidity without the use of external heating. In fact, since no heat is being applied to the regenerators, there is a chance that, during high ambient relative humidity conditions, they start operating as absorbers. The control system implemented was designed to turn off the regenerators when the ambient relative humidity exceeded a value of 50 percent.

Figure 17 also shows that the temperature measurements near the ceiling do not differ significantly between each other (brown and orange lines) and remain at 5° C (41° F), approximately. On the other hand, the temperature upstream of the absorber (i.e. blue line) shows lower values. The increase in temperature during the defrosting cycles is clearly seen as temperature spikes during the day.

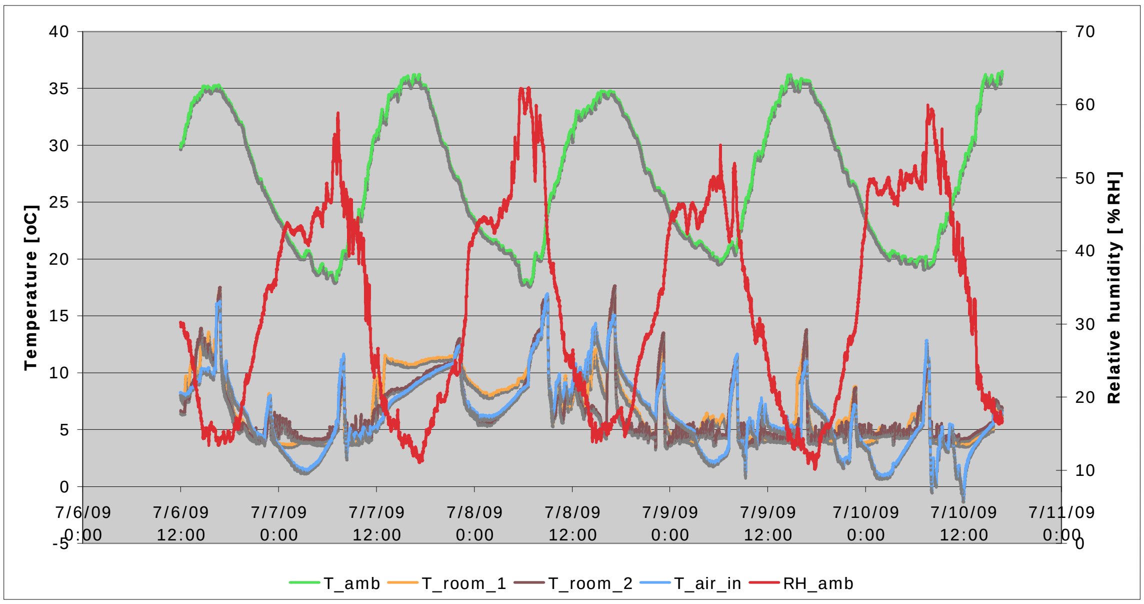

Figure 17. Ambient temperature and relative humidity compared with temperature distribution inside the cold storage

Source: UC Merced

3.2. Performance of the Liquid-Desiccant System

The liquid-desiccant system was installed at Aslan Cold Storage, LLC, in Kingsburg, California. The overall system was composed of an absorber (or dehumidifier), and internal heat exchanger (IHX), a storage tank, two regenerators, sensors, and a control unit.

The performance of the liquid-desiccant system was analyzed using different variables. As the regeneration process was performed with ambient air, the control logic for the operation of the regenerators was designed to turn them off when the ambient relative humidity was higher than a certain threshold. Figure 18 shows the status of the regenerators (0=off, 1=on) as the ambient relative humidity varied during the day. It is observed that during October 2008 the liquid desiccant system was regenerating the desiccant solution for about seven hours a day. The rest of the time, only the absorber was operating. The regeneration rate was higher than the absorbing rate since the regeneration area was larger by using two cores as opposed to using only one for the absorber. Also since two regenerators were used, the flow rate of liquid desiccant that was regenerated was roughly twice the flow rate through the absorber, i.e. 0.8 liters/min (0.21 gal/min), approximately. The higher regeneration capacity is clearly seen in Figure 19, where the level of the liquid desiccant solution inside the tank goes down during an interval of several days. It is important to note that only power for the fan and pump was used for the regeneration process; no other source of energy was used to provide heat.

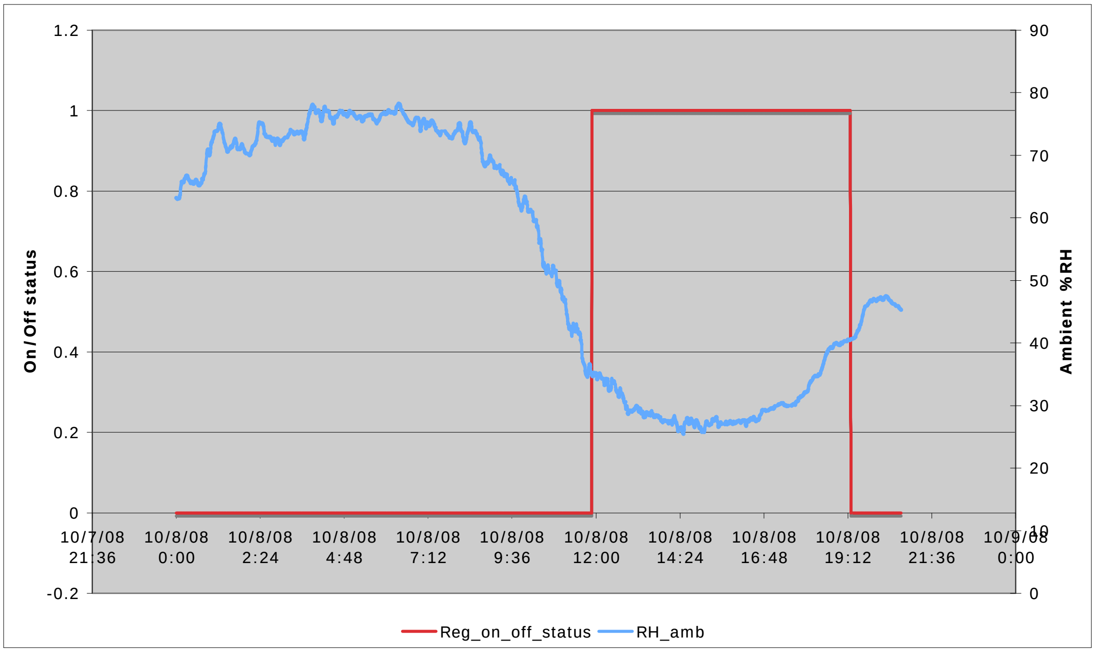

Figure 18. Operating logic for turning regenerators on and off based on ambient relative humidity

Source: UC Merced

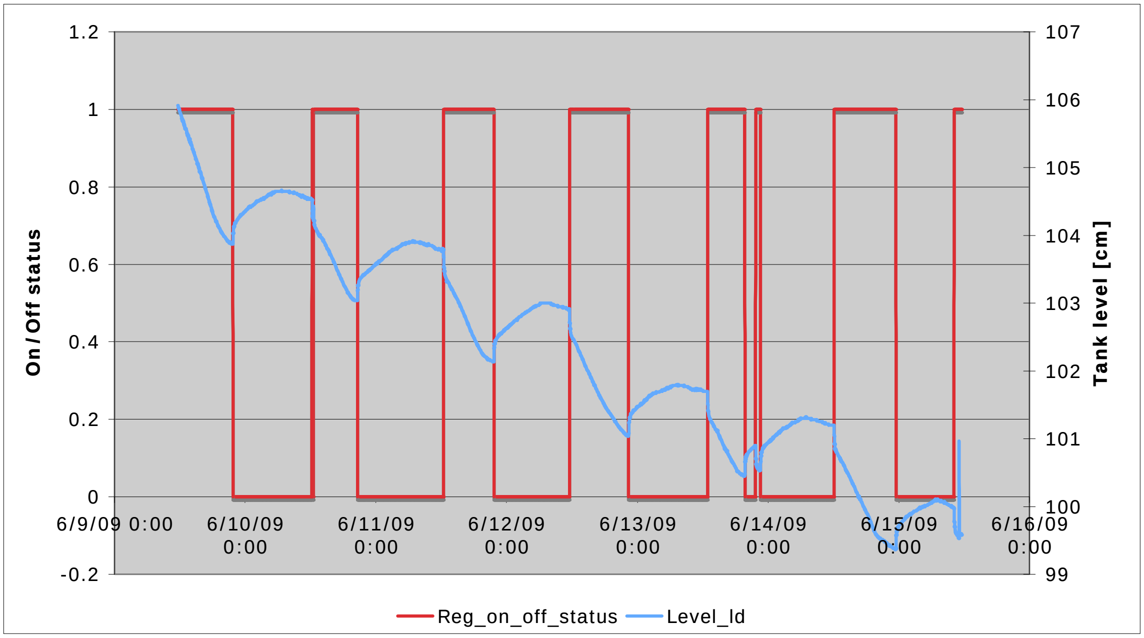

Figure 19. Liquid desiccant level inside the storage tank for the operation of several days. The status of the regenerators is also shown.

Source: UC Merced

For a flow rate of 0.2381 m^3/s (505 cfm) of air at 1°C and 99% relative humidity (RH), the water absorption rate was 0.015 kg/min. Thus, this small absorber can reject about 22 kg (48 lbs) of water during 24 hours of continuous operation. Under these operating conditions, a facility operating with 40 m^3/s (85000 cfm) could remove about 3703 kg (8163 lbs) of water per day. However, the absorber and regenerators of the liquid-desiccant system would need to be designed to process this much flow without using much space inside the cold storage. In reality, the relative humidity inside the cold storage is less than 100 percent most of the time, except during the defrosting cycles, so the amount of water needed to remove before it reaches the evaporator is less than the figure indicated above.

3.3. Defrosting Cycle

The defrosting cycle is intended to remove the ice that has formed on the surface of the evaporator. Water is sprayed on the evaporator, surface to melt the ice. If moisture is removed from the air before it reaches the evaporator, then the number of defrosting cycles could be reduced or even eliminated. A much smaller amount of water would need to be sprayed inside the cold storage to maintain the high humidity levels needed by the produce.

If the absorber of the liquid-desiccant system can process the air upstream of the evaporator, lowering the dew point temperature up to a point that is lower than the surface temperature of the evaporator, then no ice will form. This concept was tested by comparing the dew point temperature calculated for the air downstream of the absorber against the measurements of the surface temperatures at the evaporator at the refrigerated warehouse. Figure 20 shows the comparison between the refrigerant temperatures at the inlet manifold of the evaporator (red line), the coil surface temperature (blue line) and the calculated dew point temperature downstream of the absorber (yellow line). Three defrosting cycles are clearly observed.

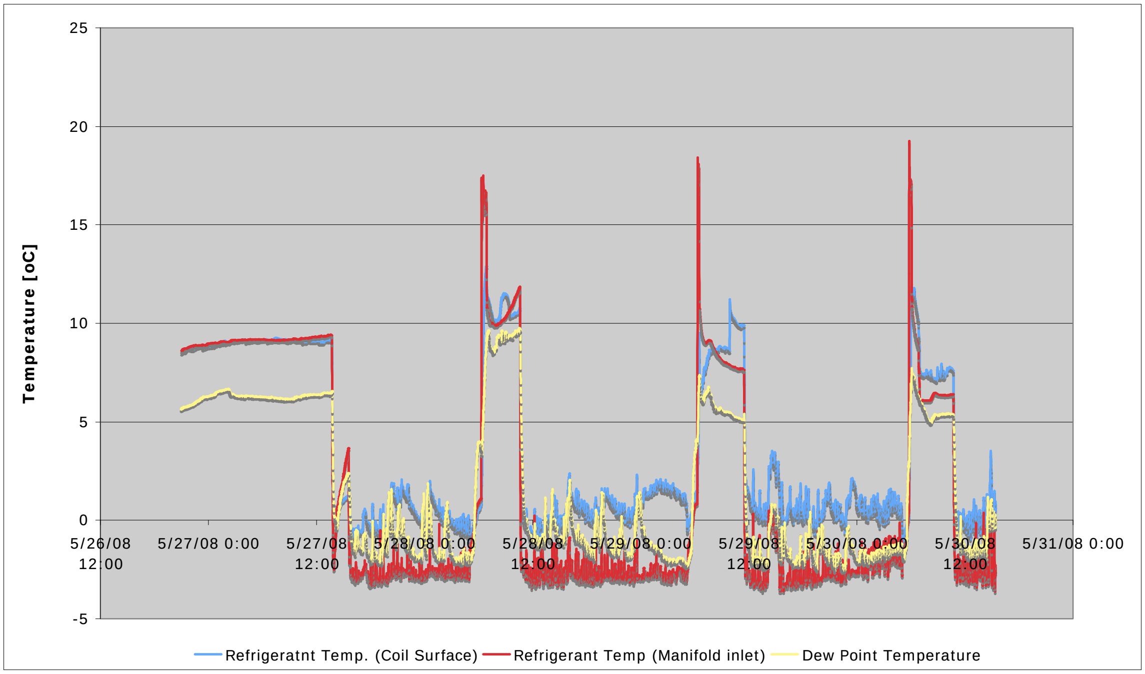

Figure 20. Dew point temperature (downstream of the absorber) vs. coil surface and manifold inlet refrigerant temperature

Source: UC Merced

The figure shows that, during the operation of the liquid desiccant system at the cold storage facility, the absorber consistently showed air dehumidified to a point where the dew point temperature was below the surface temperature of the evaporator. The dew point temperature after the absorber, shown in yellow, is below the coil surface temperature, and in blue line indicates suppression of ice formation. The red curve shows the measurement of the temperature at the evaporator manifold that has no flow of air around it. This temperature is closer to the refrigerant temperature and is in general lower than the dew point temperature of the dehumidified air. This trend was observed consistently at low air-flow rates through the evaporator (mainly at the beginning and the end of the season).

During summer conditions the formation of ice was significant. The layer of ice on the surface of the evaporator grew to a point where the evaporator was no longer reducing the temperature inside the room and the temperature started to rise. This effect can be observed in Figure 21 starting at around 2 a.m. For a relative humidity inside the room between 75 and 100 percent, seen in Figure 21, the ice formation can be characterized by the increase in air pressure drop across the evaporator. Figure 21 shows that the air pressure drop increased from 200 Pascals (Pa) right after the defrosting cycle to 800 Pa right before the start of a next defrosting cycle. An air pressure drop of 600 Pa indicates the point where enough ice had accumulated so that the temperature of the room started to increase.

The proposed liquid desiccant system targeted this type of operation in which the compressor continues to run because the cold storage temperature is increasing. However, due to the layer of ice that has formed, the evaporator is no longer effective in lowering the temperature of the air. A practical rule indicates that compressor power decreases 2 percent to 3 percent for every degree Fahrenheit that suction temperature increases. The refrigerant temperature measured at the manifold indicated that under these operating conditions of ice formation, the temperature of the refrigerant was reduced by 3° C (5.4°F), approximately, indicating an estimated increase in compressor power between 10.8 and 16.2 percent.

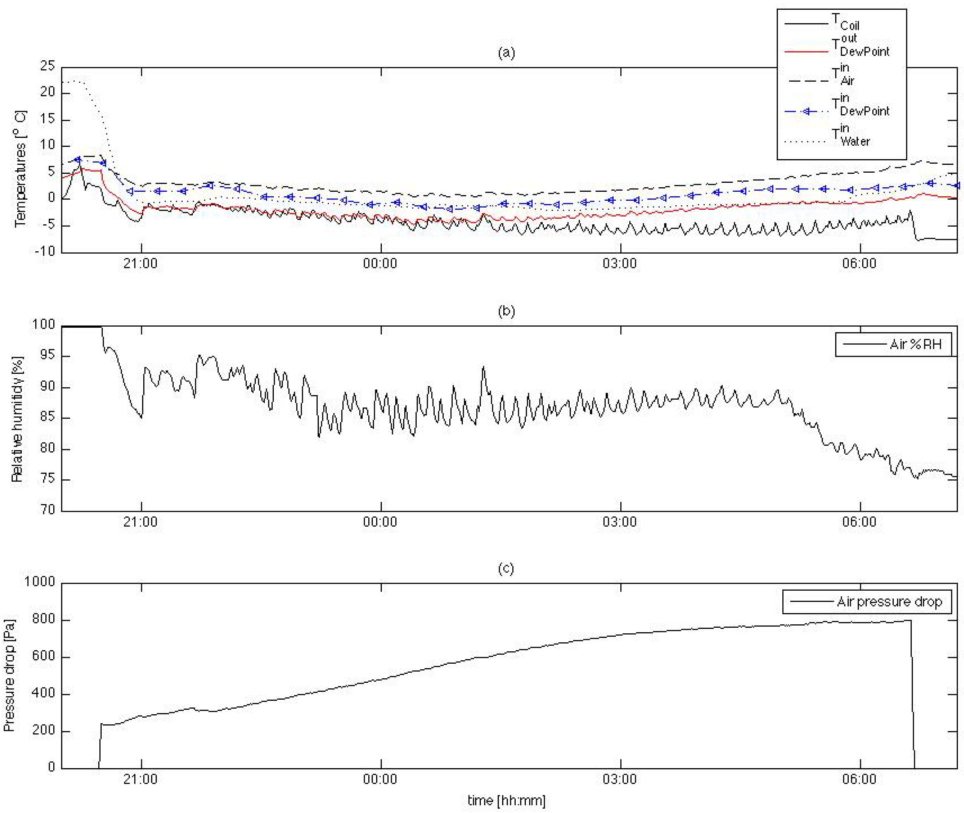

Figure 21. Operating conditions during summer

Source: UC Merced

3.4. Reliability

The liquid-desiccant system was installed during May 2008, and after a few days of operation, it was stopped due to a major leak that developed below the absorber. It was determined that there was not enough head for the liquid desiccant to overcome the pressure drop at the low-pressure side of the IHX, so the absorber was raised with respect to the IHX, and a spill containment tray was installed below the absorber/IHX stand.

The system was restarted during June 2008 and operated until December 9, 2008, when the season at Aslan Cold Storage, LLC, was almost over. The system was turned on again on June 9, 2009, and it was operated until the end of July.

With the proper maintenance, the system showed to be reliable. No major failures were observed during the operation of the system. However, it is noted that liquid desiccants are corrosive. Abnormal operation, such as the desiccant leak that was caused by inadequate drainage of the absorber, can lead to the entrainment of desiccant droplets in the process air. The exposure of metal surfaces to droplets of calcium chloride resulted in corrosion of some components.

3.4.1. Preventive Maintenance

Just as any mechanical system, the liquid desiccant installation requires proper maintenance for its correct operation. Due to the harsh industrial conditions at Aslan Cold Storage, LLC, the air filters were replaced about once a month. Figures 22(a) and (b) show the air filters of the regenerator and absorber before being replaced. The thick layer of dirt on the surface of the filter translated into a higher pressure drop of the filter, which tended to curve towards the surface of the absorber (or regenerator). Care must be taken to avoid contact of the filter with the front surface of the absorber (or regenerator) so that liquid desiccant does not start leaking through the filter. This was found to be a source of leaks that was resolved by preventing contact of the filter with the surface of the absorber by means of plastic strips. A leak developed by this mechanism is shown in Figure 23. In general these leaks were found to be easy to fix without having the system shut down for a long time.

(a) Regenerator (b) Absorber

Figure 22. Air filters before being replaced

Photo Credit: UC Merced

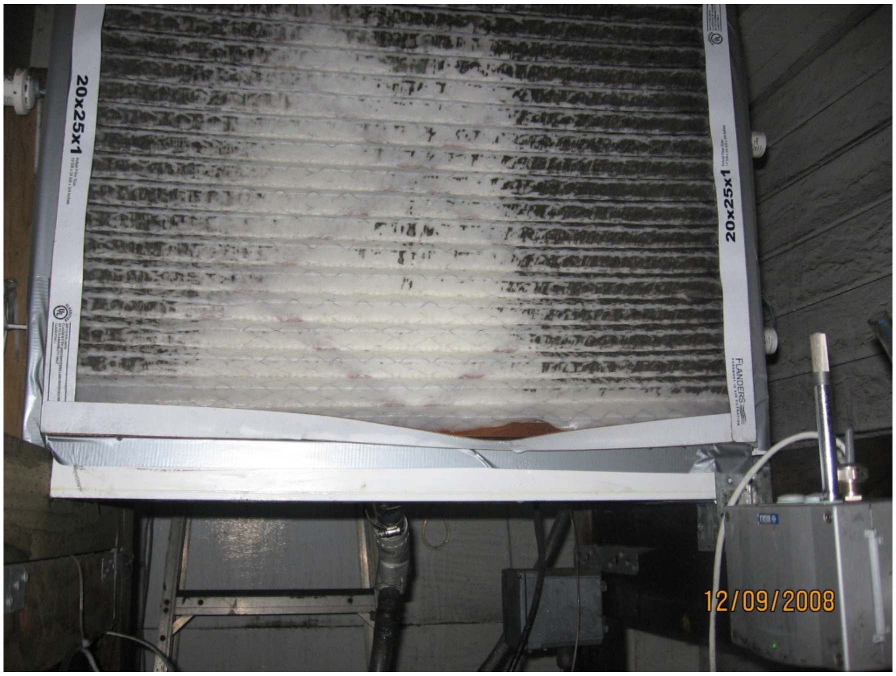

Figure 23. Leak due to contact between filter and absorber

Photo Credit: UC Merced

It is important to note that although the air filters removed most of the particulate and dust from the air, there is still some of the dirt that goes through the filter and enters in contact with the liquid-desiccant, as it is shown in Figure 24. Therefore, it is imperative to utilize filters for the liquid-desiccant side.

Figure 24. Dirt in direct contact with the liquid desiccant

Photo Credit: UC Merced

Filters for the liquid desiccant also require regular maintenance and replacement. The liquid-desiccant system operated for five months before the filter was replaced.

(a) (b)

Figure 25. Liquid desiccant filters. (a) After five months of operation. (b) New Filter

Photo Credit: UC Merced

Figure 25(a) shows the condition of the liquid-desiccant filter after five months of continuous operation. Figure 25(b) shows the condition of a brand new filter. It was found that dirty liquid desiccant filters increased the pressure drop across the assembly and generated small leaks at the top of the plastic container that holds the filter. The replacement of the filter cartridge solved this leak.

3.4.2. Reduction in Liquid-Desiccant Flow

It was noted that after two seasons of operation, the flow rate of the liquid desiccant going toward the absorber had slowly decreased to almost zero. A detailed analysis of the causes showed small particles had blocked the small passages for the liquid-desiccant flow at the absorber. The particles were forced out of the absorber by gravity, as seen in Fig. 26 inside the area of the red circle. The very low flow of liquid desiccant in the absorber involves small holes in the upper manifold that were blocked with the operation of the system. It is not clear at this point the source of the particles, but erosion effects could be one cause. A thorough cleaning of the inner passages of the absorber solved the problem, and the flow rate increased to normal levels.

Figure 26. Particles blocking flow of liquid desiccant to the absorber

Photo Credit: UC Merced

3.4.3. Corrosion

The liquid desiccant was found to be easy to work with and posed no substantial harm to human skin. However, it was noticed that the skin itches after some contact with the liquid-desiccant, so water needs to be near the location of the system so that hands can be rinsed in case of accidental contact with the desiccant solution. Over a long period of time, the effects of the contact between liquid desiccant and metals are evident. Signs of corrosion can be seen in metals that have become in contact with the solution and have not been cleaned properly. Figure 27 shows the metallic bar inside one of the flow meters that is starting to corrode after five months of operation of the system. Similar effects can be seen on the border of the reduction zone of the fan, as shown in Figure 28. A similar effect was found in some thermocouples after operation for almost two seasons, as seen in Figure 29.

Figure 27. Corrosion signs on the metallic parts of the flow meter

Photo Credit: UC Merced

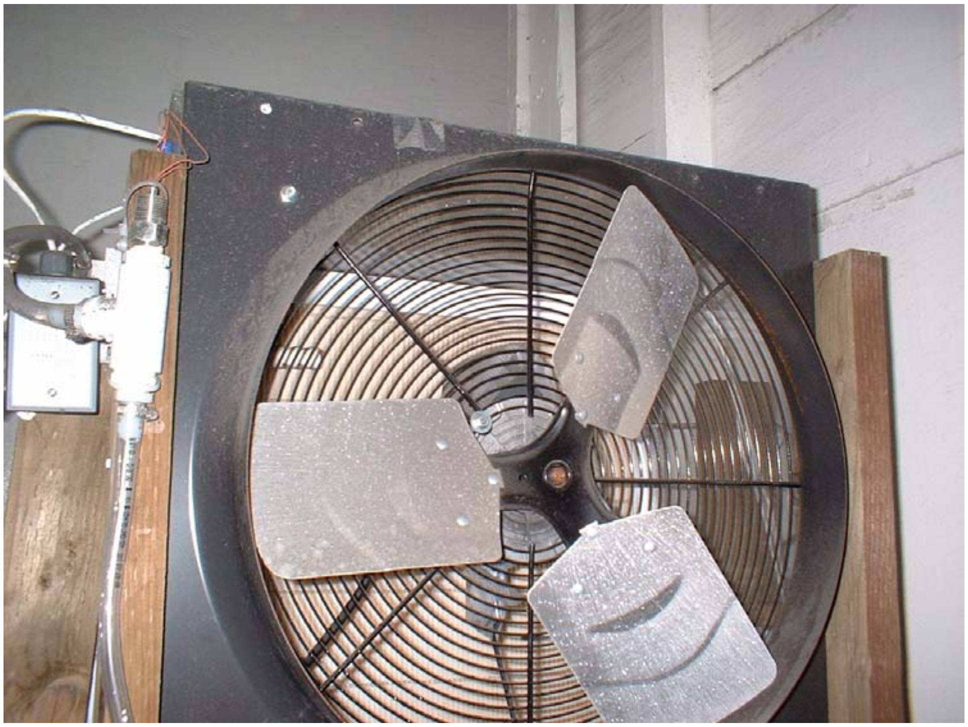

Figure 28. Corrosion of the fan

Photo Credit: UC Merced

Figure 29. Corrosion of some sensors

Photo Credit: UC Merced

4.0 The Scale-Up of the Technology

The measured performance of both the absorber and desorber that were used in the small-scale test was used to calibrate the heat and mass transfer coefficients in AILR's computer models of these components. The calibrated computer model was then used to design a liquid-desiccant system that could eliminate defrosting at Room 2 at the Aslan warehouse where the small-scale test was conducted.

The key operating parameters in Room 2 are assumed to be:

Total air flow	85,000 cfm
Air temperature	0.9° C (33.8°F)
Air relative humidity	85%
Coil temperature	-4° C (24.8°F)

The liquid-desiccant absorber must dry the process air to a dew point that is less than the coil temperature if it is to prevent the accumulation of frost on the coil. Assuming that the liquid desiccant absorber has a face velocity of 400 fpm, the performance of the absorber was modeled over a range of desiccant (calcium chloride) concentrations and absorber depths. The required supply air dew point was achieved with a 35% desiccant concentration and an absorber with a 12" depth. The air-side pressure drop across the absorber is estimated to be 0.25" water column (w.c.). Assuming a fan/motor efficiency of 0.55, the fan power to move 85,000 cfm across this pressure drop is 4.54 kW. Although significant, some of this additional fan power would be compensated for by the lower pressure drop across the frost-free cooling coils.

An absorber that operates for 24 hours at the preceding conditions will absorb 7,200 pounds of water. Since low ambient humidity is likely to occur only during midday hours, the regenerator must be able to remove 7,200 pounds of water from the desiccant during only a few hours.

In this study, it is assumed that the regenerator operates for six hours. During these six hours, the regenerator is supplied with outdoor air at 95 F and 20% rh.

A 6"deep regenerator with a face velocity of 420 fpm can remove the required water in six hours if it has an airflow of 153,000 cfm. With the shallower core, the regenerator will have a lower pressure drop than the absorber—0.13" w.c. vs. 0.25" w.c. However, experience on the small-scale test rig that operated at the Aslan warehouse indicated the need for good filtration of the regeneration air. Assuming the air filter adds 0.20" w.c., (on average), the total fan power for the regeneration air will be 10.78 kW.

In this design study, the refrigeration system provides approximately 200 tons of cooling. Assuming an EER of 7.5, the compressor power for the refrigeration system will be 320 kW. If the liquid desiccant, by eliminating defrosting, reduces compressor power by 10 percent, then the daily savings will be 704 kWh. The electricity consumption for operating the regenerator for six hours will be 64.7 kWh. As previously noted, the fan power for the absorber will be partially compensated for by a lower pressure drop across the frost-free evaporator. If one assumes that the net effect of the absorber is to increase pressure drop by 0.125" w.c., then the additional fan energy for the absorber will be 55.5 kWh. (Power for pumping the desiccant is small compared to powering fans and so they have been ignored in this study.)

Under these conservative preceding assumptions, the liquid-desiccant system produces a net energy savings of 583 kWh per day. These energy savings are unlikely to justify the cost of the liquid desiccant system (which, because of the very large volume air that must be processed at both the regenerator and absorber, will be over $250,000).

It is unlikely that a liquid-desiccant system for frost control in a refrigerated warehouse will be economically attractive as a retrofit system. However, if a liquid-desiccant system is applied to a new warehouse, then it would be possible to greatly reduce the size and cost of the evaporator by significantly increasing the density of fins on the coil (e.g., 10 fpi instead of 4 fpi). In this case, the cost savings for the evaporator could offset the cost for the liquid-desiccant system.

5.0 References

U.S. Patent No. 5,351,497, October 1994, AIL Research.

California Energy Commission RFP CIEE-FIER-2006.

Kosar and Dumitrescu, "*Humidity Effects on Supermarket Refrigerated Case Energy Performance*," ASHRAE Paper OR-05-16-1, 2005.

Turpin, J. "So, How Do You Keep Frost, Ice Out of Warehouses?" http://www.achrnews.com/CDA/ArticleInformation/features/BNP__Features__Item/0,1338,2824,00.html