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Performance of an adiabatic cross-flow liquid-desiccant absorber inside a refrigerated warehouse

*Sergio M. Pineda, Gerardo Diaz**

School of Engineering, University of California – Merced, 5200 North Lake Rd., Merced, CA 95343, USA

ARTICLE INFO

Article history:
Received 5 November 2009
Received in revised form
27 July 2010
Accepted 10 August 2010
Available online 19 August 2010

Keywords:
Cold storage
Desiccant
Defrosting
Energy consumption
Reduction
Moisture

ABSTRACT

Liquid-desiccant systems have been extensively studied as a way of reducing the latent load on air conditioning systems. Most of the studies have targeted the removal of moisture from air at ambient conditions. The literature about the use of liquid desiccants in low temperature applications is scarce. In this study, a small-scale liquid-desiccant absorber is installed inside a commercial refrigerated warehouse. Its performance under realistic operating conditions inside a pre-cooling room is analyzed. The results show that the dew point temperature of the air downstream of the absorber is comparable to the evaporator surface temperature suggesting the potential to delay the formation of ice on the cooling coil. An internal heat exchanger is used to lower the temperature of the inlet liquid-desiccant flow to the absorber and the regeneration process is performed using only ambient air. The analysis of the reduction in water and energy consumption for a scaled-up system is also performed.

Performance d'un absorbeur à déshydratant liquide à écoulements croisés adiabatiques dans un entrepôt frigorifique

Mots clés : Entreposage frigorifique ; Déshydratant ; Dégivrage ; Consommation d'énergie ; Diminution ; Humidité

1. Introduction

The projection of the world's energy consumption predicts an increase in the demand of energy of approximately 50 percent from 2005 to 2030 (EIA, 2008). Fossil fuels such as petroleum, natural gas and coal are likely to remain as the main supply of the energy used worldwide until 2030. Consequently, in response to this high demand for energy, and because oil production is concentrated in a small group of countries, oil prices are expected to remain relatively high (EIA, 2006). Furthermore, the increase of fossil fuel usage in the absence of national policies and/or international agreements that limit or

* *Corresponding author*. Tel.: +1 209 228 7858.
E-mail address: gdiaz@ucmerced.edu (G. Diaz).

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doi:10.1016/j.ijrefrig.2010.08.005

Nomenclature

C_p	specific heat (kJ kg^{-1} K^{-1})
h_{fg}	enthalpy of vaporization, (kJ kg^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
Q	heat transfer rate (kW)
RH	relative humidity (%)
ΔT	temperature difference (°C)
Greek symbols	
Δw	humidity ratio difference (kg_w kg_a^{-1})
Subscripts and superscripts	
a	air
Abs	absorber
IHX	internal heat exchanger
ld	liquid desiccant

reduce greenhouse gas emissions is expected to result in the increase of emissions from 28.1 billion metric tons in 2005 to 42.3 billion metric tons in 2030 (EIA, 2008). Thus, policies that encourage more efficient use and production of energy are desperately needed.

The efficient use of electricity combined with innovative technologies are fundamental tools for reaching suitable levels of energy consumption and greenhouse gas emissions. One example is the refrigerated warehouse sector where the energy consumption is clearly intensive and where a large potential for additional energy saving exists.

A cold storage or refrigerated warehouse works as a traditional vapor-compression refrigeration cycle. In general, the evaporator is located inside the cold storage, where sensible and latent heat are removed from the indoor air to lower the temperature of the produce inside the warehouse. Heat is rejected outside of the facility by using cooling towers. In order to maintain the quality of the produce, a high relative humidity level is desired inside the cold storage. In addition, produce is transported in and out of the facility allowing hot and humid ambient air inside the warehouse. These operating conditions translate into a continuous formation of ice on the surface of the evaporator which is usually removed by means of running a defrosting cycle several times a day. In a common defrosting cycle implementation, water is sprayed on the surface of the evaporator to melt the ice. Thus, at the end of the defrosting cycle the surface of the evaporator is near the temperature of the inlet water and the compressors are turned on to bring the temperature of the evaporator surface back to its design conditions. This procedure is inefficient and involves the use of large quantities of water to defrost the cooling coils.

Liquid desiccants are natural or synthetic substances capable of absorbing moisture from the air due to the difference of vapor pressure between the surrounding air and the desiccant surface. They are hygroscopic at low temperatures (absorb water) and hydrophobic at higher temperatures (reject water). Liquid desiccants have the capability of providing localized dehumidification since the regeneration stage can be located far away from the dehumidification zone. Concentrated solutions of liquid desiccants are often sprayed into air streams or applied as falling films on the surface of dehumidifiers to absorb water vapor from the incoming air. The diluted liquid-desiccant solution is then pumped to a regeneration stage where heat is applied to reject the moisture content of the solution increasing its desiccant concentration. The concentrated desiccant solution is pumped back to the absorber to repeat the cycle.

In the air conditioning systems field, the use of liquid-desiccant systems has become more popular in the past decades due to the need for reduction in the consumption of energy and water (Feyka and Vafai, 2007; Jain and Bansal, 2007). The capability of these systems in handling latent heat in the space that will be conditioned by a dehumidifying process also allows control of the humidity without the overcool/reheat scheme as is done in a regular ventilating and air conditioning system (VAC) (Katejanekarn and Kumar, 2008).

Most liquid-desiccant applications target cooling and dehumidification at ambient conditions (Yin and Zhang, 2008; Daou et al., 2006; Kosar et al., 1998; Harriman et al., 1999). However, a few studies focus on the application of liquid desiccants to refrigerated spaces. A literature review of the theoretical and physical models of a refrigerated warehouse, as well as, the comparisons between a traditional vapor-compression cycle and a liquid-desiccant refrigeration cycle are found in Daou et al. (2006). Elsayed et al. (2006) analyzed the performance of an air cycle refrigerator combined with a desiccant rotor for an air conditioning application. More recently, the same authors numerically analyzed the performance of an air cycle refrigerator integrated with a desiccant system for cooling and dehumidifying a warehouse Elsayed et al. (2008). They found that the coefficient of performance can increase more than one hundred percent with respect to a conventional system.

The purpose of this work is to analyze the experimental results obtained for the performance of a small-scale prototype liquid-desiccant absorber installed at a commercial refrigerated warehouse.

2. Description of the installation

The operation of a liquid-desiccant absorber inside a commercial refrigerated warehouse was analyzed by means of installing a small-scale prototype liquid-desiccant system

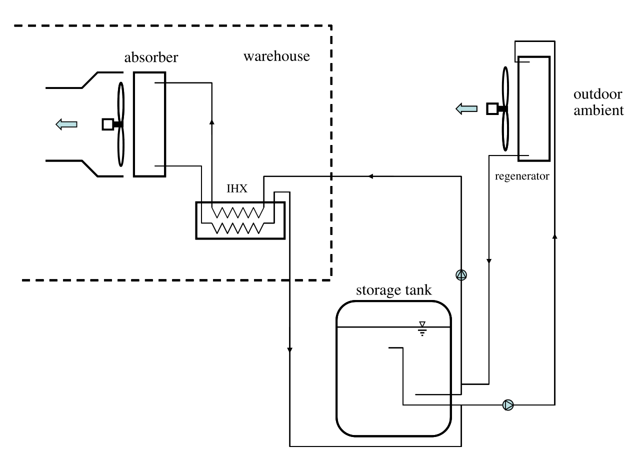

Fig. 1 – Small-scale liquid-desiccant prototype system.

at a representative cold storage site in California's San Joaquin valley. The purpose was to measure the performance of an adiabatic absorber under realistic operating conditions inside a commercial cold storage facility. The inlet air conditions to the absorber and facility evaporator were similar so that the air properties downstream of the absorber, T_{Abs}^{out}, were compared with the temperature measured at the surface of the evaporator. If the dew point temperature of the outlet air from the absorber was below or near the surface temperature of the evaporator then a potential exists to eliminate or at least delay the formation of ice at the surface of the cooling coil. Temperature and relative humidity sensors were installed to measure inlet and outlet air conditions at the absorber, and temperature sensors were used to measure the surface temperature of the coil.

Calcium chloride solution was used as the working fluid for the liquid-desiccant system which consisted of an absorber, internal heat exchanger (IHX), storage tank, two regenerators, and control unit. The operation of the system is depicted on Fig. 1 and consisted of the following processes: concentrated calcium chloride solution was pumped from a storage tank of approximately 1.14 m^3 of capacity to the IHX. The desiccant was cooled at the IHX by using the flow of diluted desiccant coming out of the absorber. The outlet of the high-pressure side of the IHX was fed into the inlet of the absorber. The concentrated liquid-desiccant solution falling as a film down the walls of the dehumidifier absorbed moisture from the cold humid air inside the cold storage. The diluted liquid-desiccant solution passed through the low-pressure side of the IHX and then was pumped to the regenerators that were located outside of the cold storage. After being regenerated, the concentrated liquid-desiccant solution flowed to the storage tank to repeat the cycle.

At the absorber, the liquid-desiccant film interacted with the air flowing in cross-flow configuration. The air speed could be adjusted with a simple speed controller connected to the fan. Heat and mass were exchanged at the interface between the air and the liquid-desiccant film. Since the liquid-desiccant solution is mildly corrosive, the absorber, regenerators, and IHX were built by AIL Research using non-metallic materials.

It is noted that since the ambient conditions at California's San Joaquin Valley are hot and dry during several months of the year, only ambient air was used to regenerate the liquid desiccant so that no external source of heat was used. Two regenerating cores with similar design as the absorber were used with the purpose of having a large regeneration surface. During the night when the ambient relative humidity increased, the control unit shut down the flow to the regenerators and only the concentrated liquid-desiccant solution from the storage tank was used for the absorption process at the dehumidifier.

The project lasted for two and a half years. The liquid-desiccant system effectively operated for twelve months that included the end of season 2007, season 2008, and the start of season 2009. Data was collected at a sampling rate of twenty six channels per minute. The data have been carefully examined and it has been concluded that the operation of the system can be summarized by analyzing representative information for summer and fall conditions.

Table 1 – Design operating conditions inside refrigerated warehouse.

Design parameter	Range	Units
Indoor temperature	0–1	°C
Relative humidity	85–100	%
Evaporating temperature	−6.7 to −3.9	°C
Evaporator air-flow rate	7–40	$m^3\ s^{-1}$
Ammonia flow rate	0.605	$kg\ s^{-1}$
Cooling capacity	351	kW
Condensation temperature	29–32	°C

3. Operating conditions inside the refrigerated warehouse

The air inside the cold storage is required to have a high relative humidity to avoid lowering the quality of the produce. The design operating conditions inside the refrigerated warehouse, as well as, the ambient conditions are listed in Tables 1 and 2, respectively. In particular, pre-cooling rooms have bins with produce that are taken in and out of the cold storage facility allowing hot ambient air into the room. This increases the load on the evaporators and augments the amount of moisture that condensates on the surface of the cooling coils resulting in the formation of ice. The defrosting cycle consists on spraying water on the evaporator until the ice has melted. The spraying of water also creates a mist inside the cold storage that helps keeping a high moisture level (above 75%) inside the room. A typical defrosting cycle is shown in Fig. 2 where the surface temperature of the evaporator is shown together with the temperature and flow rate of the water sprayed to melt the ice. The air pressure drop measured across the evaporator is also displayed. The cycle starts by turning the fans off as indicated by the reduction in air pressure drop from approximately 53 Pa to zero. At 5:16 pm the flow rate of water is turned on. Water at 22 °C was sprayed on the surface of the evaporators for 17 min, approximately. It is observed that the temperature of the cooling coil increases during this process and remains at around 8 °C even after the flow of water has ceased. The defrosting cycle ends when the fans and refrigerant flow are turned back on and the surface temperature of the evaporator goes down to its design value. The cycle is repeated one to three times a day depending on the time of the year and type of produce being cooled. It can be observed that the air pressure drop across the evaporator has a lower value after the defrosting cycle indicating the removal of ice. During the peak of the season, the air pressure drop can reach values near 800 Pa therefore needing a significant amount of water to melt the ice. Integration of the flow rate

Table 2 – Historical ambient conditions in San Joaquin Valley, California.

	Range	Units
Summer temperature	16–36	°C
Fall temperature	6–20	°C
Summer RH	23–65	%
Fall RH	30–80	%

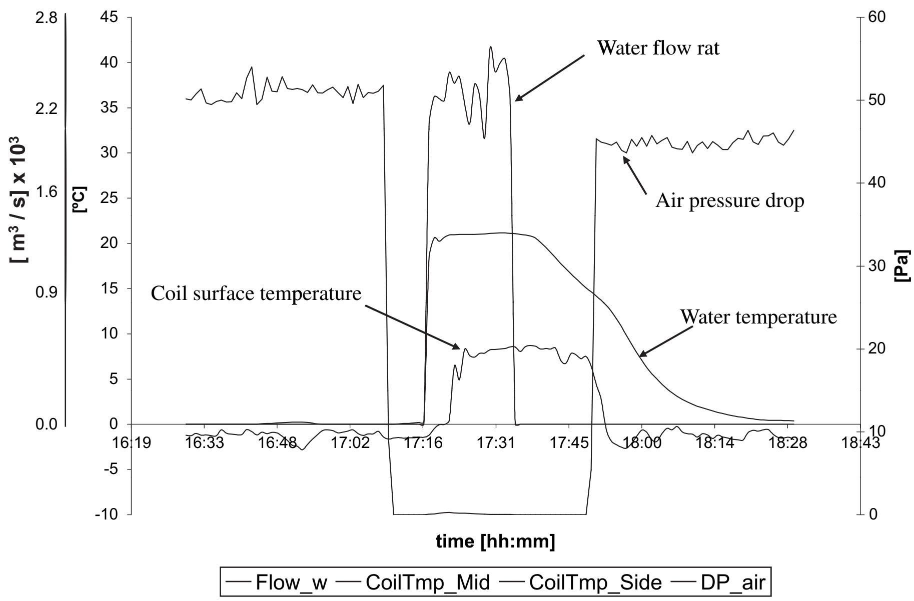

Fig. 2 – Defrosting cycle sequence of steps.

with respect to time results in 2.3 m^3 of water being used for each defrosting cycle.

3.1. *Operation of adiabatic absorber and evaporator inside the cold storage*

Liquid-desiccant systems have received significant attention recently as a way to reduce latent loads on air conditioning systems (Fumo and Goswami, 2002). Most systems have been installed to dehumidify ambient air before it reaches the evaporator. As the air humidity is absorbed at the dehumidifier, the temperature of the liquid desiccant increases due to the addition of heat from the enthalpy of condensation of water vapor. Thus, the design of a liquid-desiccant absorber includes the flow of a cooling fluid inside its walls that removes heat from the liquid desiccant.

A novel application of liquid-desiccant systems corresponds to the localized removal of moisture from the air inside refrigerated warehouses for the food industry which tends to operate with relatively high levels of indoor air humidity. This high indoor air humidity translates into ice formation at the evaporator. The localized removal of moisture by the liquid-

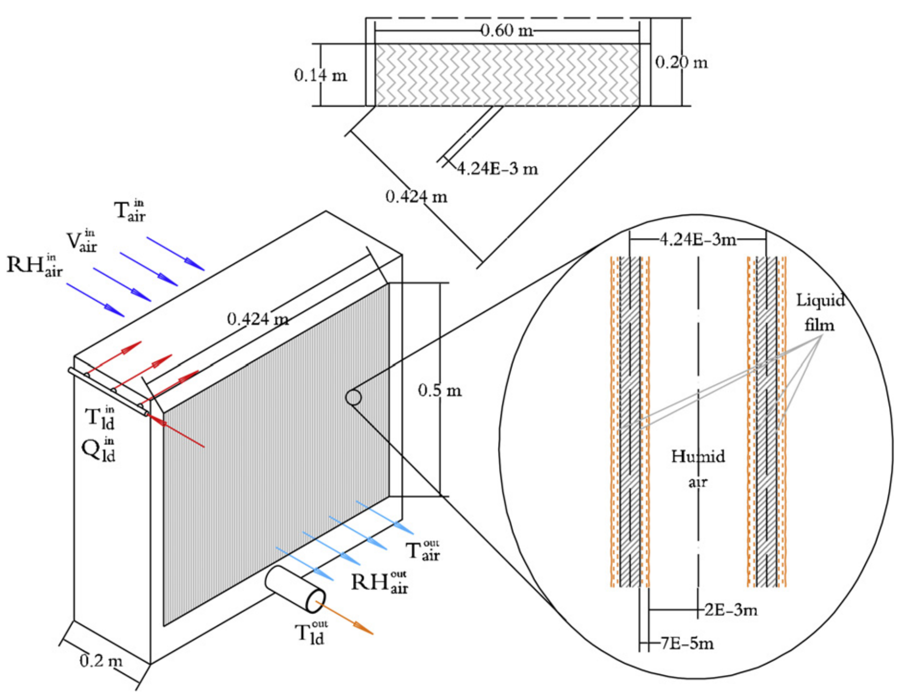

Fig. 3 – Schematic of adiabatic absorber.

Table 3 – Absorber operating conditions and geometric specifications.

	Parameter	Range	Units
Liquid desiccant (calcium chloride)	Concentration	33–36	% (by weight)
	Inlet temperature to IHX	18–25	°C
	Volumetric flow rate	1.33×10^{-5}	$m^3 s^{-1}$
Air	Inlet RH	75–100	%
	Inlet temperature	0–15	°C
	Face velocity	1.6–1.8	$m s^{-1}$
Geometry	Face area	0.3	m^2
	Contact surface area	20	m^2
	Plate spacing	4.2×10^{-3}	m
	Core depth	0.20	m

desiccant absorber upstream of the evaporator can substantially reduce the formation of ice. Due to the low temperature of the air inside these rooms, no cooling fluid is necessary for the removal of heat from the liquid desiccant. Thus, the absorber can be designed as an adiabatic core in cross-flow configuration, as seen in Fig. 3. In this work, non-metallic pleated plates were used to form zig-zag-shaped channels with liquid desiccant entering at the top of the absorber and flowing down the walls as a film that was in contact with humid air in cross-flow configuration. Table 3 indicates the operating conditions and geometric parameters of the absorber. Inlet and outlet air temperature and relative humidity at the absorber were measured using Vaisala sensors model HMT330 with a range −40 to 60 °C with accuracy of ±0.2 °C and 0–100% relative humidity with accuracy of ±1%. Based on these measurements, the upstream and downstream air dew point temperatures at the absorber were calculated. The core had a 53% humidity effectiveness as defined in Fumo and Goswami (2002). Space and operational constraints at the cold storage prevented the air downstream of the absorber to be sent to the inlet of the evaporator. However, as a means of comparison, the surface temperature of the cooling coil, subject to similar air inlet conditions as the absorber, was measured using a Veris RTD model AA10F1 sensor with a range −50 to 50 °C with an

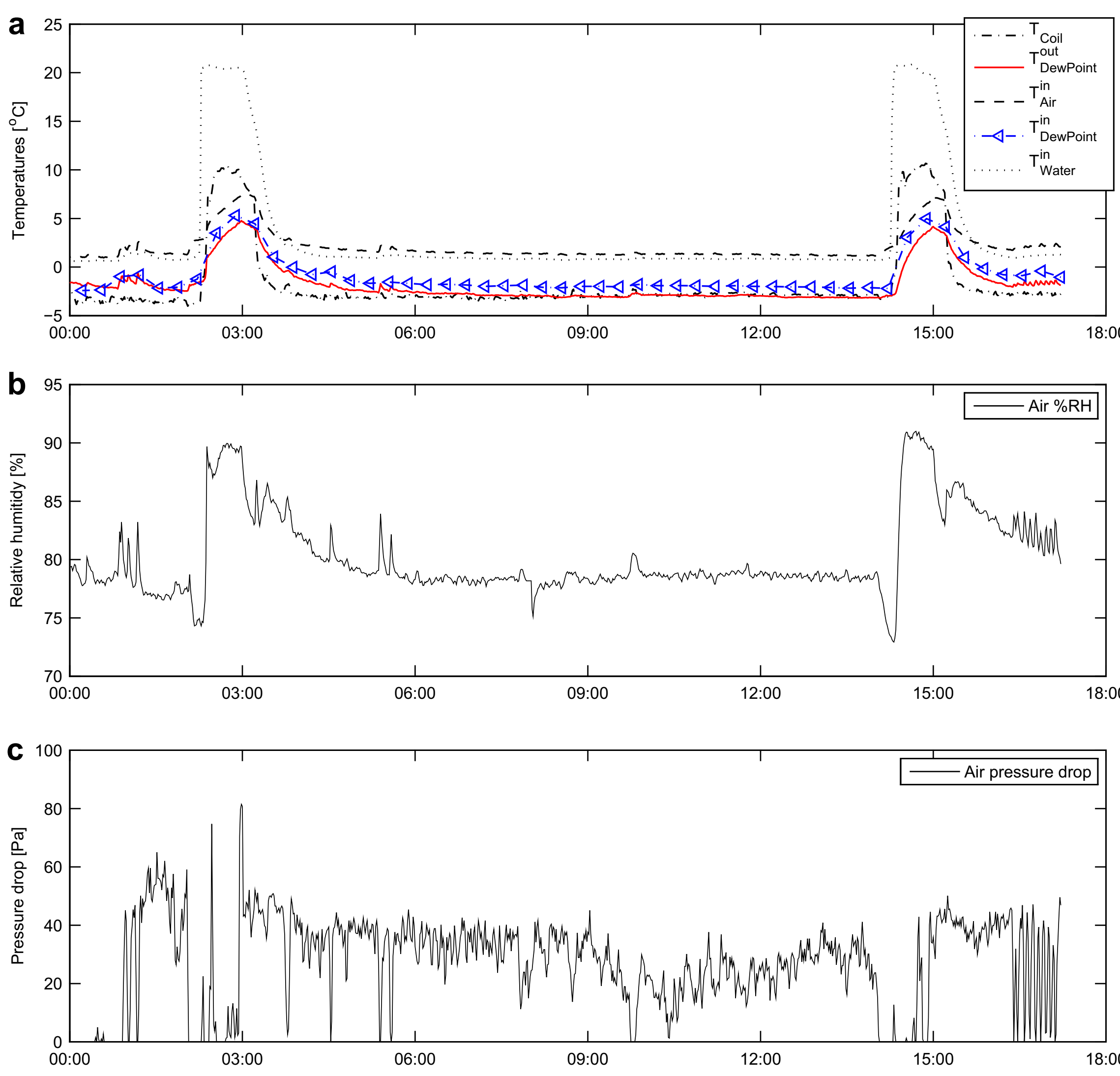

Fig. 4 – Operating conditions of the refrigerated warehouse under low air speed across the evaporator. The dew point temperature of the air upstream and downstream of the absorber is also shown.

accuracy of ±0.3 °C. Similar RTDs were used to measure the temperature of the water used for the defrosting cycle.

Operating conditions at the cold storage vary significantly throughout the year. For instance in November, the ambient temperature is low so the load on the evaporator due to infiltration of outdoor air is less significant than during the summer. Therefore, not all the fans inside the facility are used and the air flow across the evaporator is low. The air-flow rate across the evaporator varies from 7 $m^3 s^{-1}$ during late-fall conditions to 40 $m^3 s^{-1}$ during summer. Fig. 4(a) shows the performance of the absorber under fall operating conditions. Two defrosting cycles are depicted where the air inlet temperature, T^{in}_{Air}, to the absorber is shown together with the inlet temperature of the water used for the defrosting cycle, T^{in}_{Water}, and the cooling coil surface temperature, T_{Coil}. The calculated dew point temperature upstream, $T^{in}_{Dew\ point}$, and downstream, $T^{out}_{Dew\ point}$, of the absorber are also shown. It is seen that the inlet dew point temperature is, in general, above the cooling coil surface temperature leading to the formation of ice on the evaporator. The absorber removes moisture from the inlet air and lowers the dew point temperature to a value comparable with the evaporator surface temperature. Therefore, even if the formation of ice is not completely eliminated, this technology has the potential to delay frost formation significantly. Fig. 4(b) shows that the relative humidity inside the cold storage remains near 80% with spikes occurring during the defrosting cycles. The change in air pressure drop before and after the defrosting cycle, shown in Fig. 4(c), is not significant indicating a low rate of ice formation.

A very different pattern is obtained by analyzing data from summer. Fig. 5(a–c) displays the same variables shown in Fig. 4(a–c) but for a much higher air-flow rate across the evaporator and for ambient conditions that vary between 39 °C with 15% relative humidity during the day and 18 °C with 73% RH during the night. Four defrosting cycles are observed with a much more variable overall behavior. Ice formation can be inferred from the increase in air pressure drop across the cooling coil that reached a maximum of 800 Pa, as observed in Fig. 5(c). It is seen that once a significant amount of ice has been formed, the evaporator does not condition the air effectively and the temperature inside the room starts to rise. Therefore the calculated inlet and outlet dew point

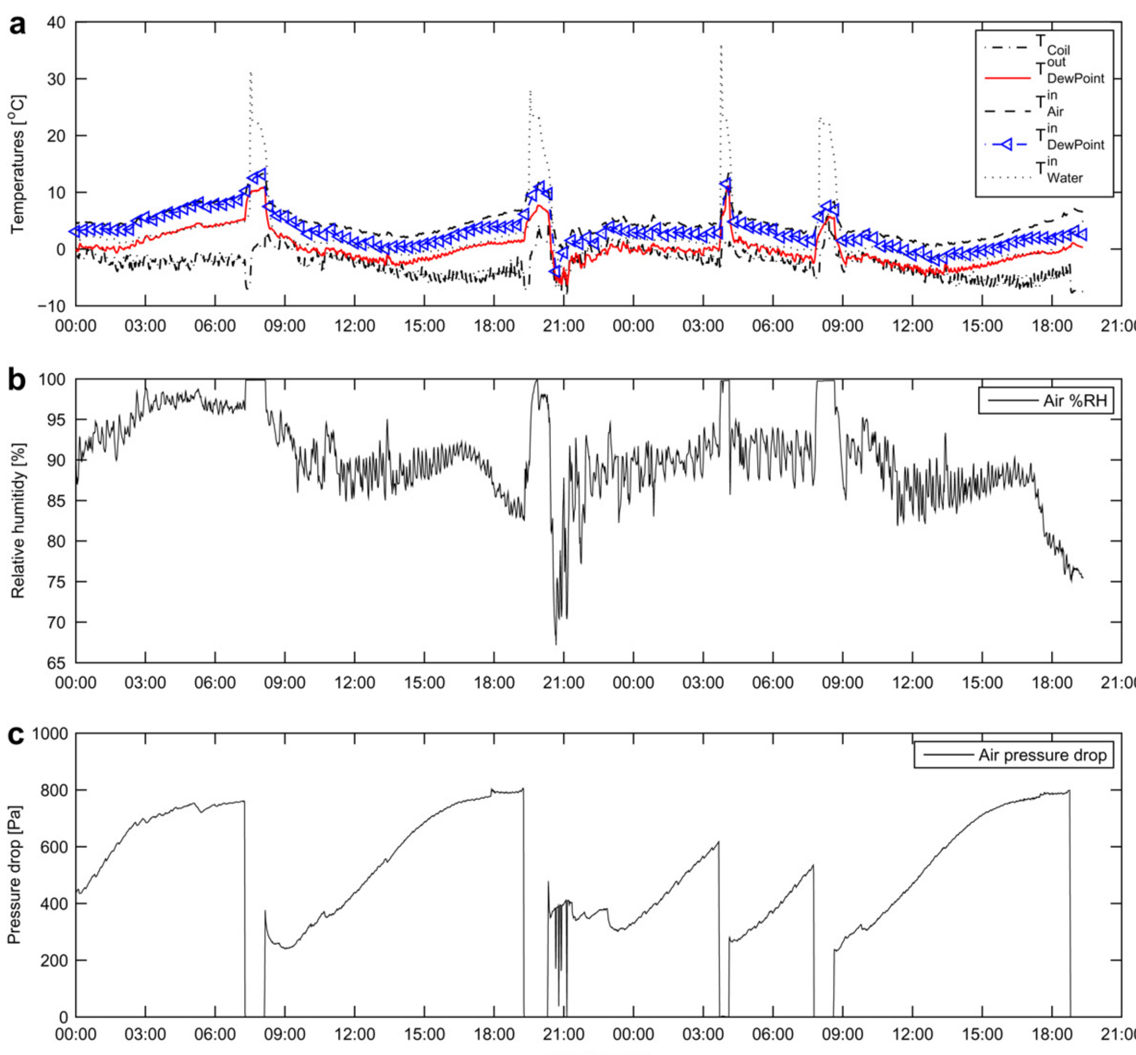

Fig. 5 – Operating conditions of the refrigerated warehouse under high air speed across the evaporator. The dew point temperature of the air upstream and downstream of the absorber is also shown.

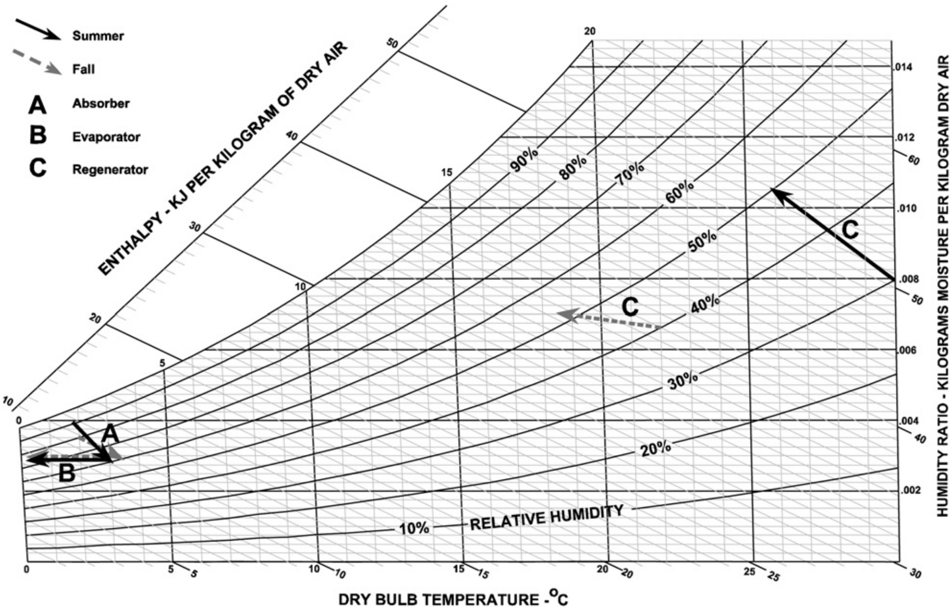

Fig. 6 – Dehumidification process across the absorber and cooling process across evaporator. Calcium chloride enters the absorber with a concentration of 35%, approximately. *A* = absorber, *B* = evaporator, solid line = summer conditions, dashed line = fall conditions.

temperatures become higher than the temperature measured at the surface of the coil. The absorber produces a downstream dew point temperature that is approximately 3.5 °C below the inlet air dew point temperature. This implies that under these conditions a layer of ice separates the temperature sensor attached to the coil from the air passing across the evaporator. It is noted that the absorber maintains the downstream air dew point temperature close to the evaporator surface temperature for air pressure drops below 600 Pa, as seen in Fig. 5(a). Fig. 5(b) shows that the relative humidity of the inlet air varies from 70 to 100% inside the room. A schematic of the dehumidification process, *A*, across the liquid-desiccant absorber is presented on a psychrometric chart in Fig. 6. The process across an evaporator receiving the outlet air from the absorber is labeled with a letter “B”. The regeneration process is shown with a letter “C”. Typical summer and fall conditions inside the cold storage are shown. Tables 4 and 5 list the parameters of the prototype absorber and regenerator, respectively. Summer and fall operating conditions are included.

3.2. *Control logic for the liquid-desiccant system*

The operation of the prototype liquid-desiccant system required a controller to choose the optimal operating parameters based on measurements of the status of the system. A SBC65EC single-board programmable computer from Modtronix Engineering was utilized for controlling purposes. Inputs included ambient relative humidity and gage pressure from the liquid-desiccant level inside the storage tank. The value of the pressure was translated to the level of liquid inside the tank. The level of liquid in the storage tank was allowed to vary between 0.96 m and 1.16 m that corresponded to a liquid-desiccant concentration between 38.1% and 32.7%, respectively. Liquid levels consistently outside of this range would indicate either a leak or a malfunctioning of the regenerators and the system would be shut down. For liquid levels within the operating range, the controller would turn off the liquid-desiccant flow to the regenerators when the liquid level reached the lower limit of the band. On the other

Table 4 – Representative summer and fall operation of prototype absorber.

	Summer	Fall	Units
T_{ld}^{in}	4.5	4.4	°C
Concentration (by weight)	35	35	%
Volumetric flow rate (liq. des.)	2.15×10^{-5}	2.1×10^{-5}	$m^3\ s^{-1}$
T_{Air}^{in}	1.9	2	°C
Inlet air RH	91	80	%
T_{Air}^{out}	2.95	3.34	°C
Outlet air RH	62	63	%
T_{ld}^{out}	2.8	3.24	°C

Table 5 – Representative summer and fall operation of prototype regenerator.

	Summer	Fall	Units
T_{ld}^{in}	25	18	°C
Concentration (by weight)	33	32	%
Volumetric flow rate (liq. des.)	2.15×10^{-5}	2.1×10^{-5}	$m^3\ s^{-1}$
T_{Air}^{in}	30	8	°C
Inlet air RH	30	50	%
T_{Air}^{out}	26	17	°C
Outlet air RH	50	53	%
T_{ld}^{out}	25	17.9	°C

hand, if the tank level reached the upper limit then the flow of liquid desiccant to the absorber would be turned off. The normal operation of the absorber and regenerators was not resumed until the level inside the tank had reached the middle of the operating range. Since the regeneration of the liquid desiccant was done using ambient air without any source of external heat, the controller turned on the flow to the regenerators when the ambient relative humidity was below 50%. For higher values of ambient relative humidity, the regenerators were kept off. This limit was set considering the vapor pressure of calcium chloride at the inlet temperature and concentration of the liquid desiccant to the regenerators.

It is noted that high-humidity weather conditions would not regenerate the liquid desiccant to the required levels of concentration. Therefore, external heat sources in the form of propane or natural gas heaters, double-effect boilers, or solar collectors would need to be implemented.

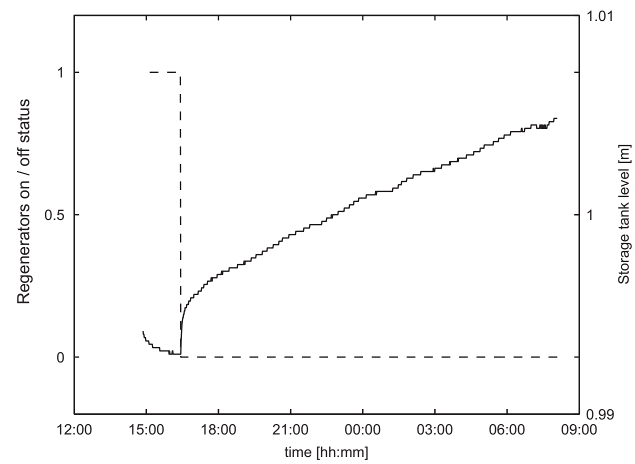

Fig. 8 – Moisture absorption rate.

3.3. *Rate of water absorption*

Space and operational constraints at the industrial facility prevented an accurate measurement of the air-flow rate across the absorber. Thus, to measure the rate of water absorption at the dehumidifier, the level of liquid desiccant at the storage tank was measured. The change in the level of the tank was obtained using a Setra pressure transducer model 264 with a range of ±6.23 kPa and an accuracy of ±1% on the full scale. The regenerators use ambient air to reject water from the liquid desiccant. However, during the night, the ambient air reached high levels of relative humidity that impeded the regeneration process. The control unit turned the regenerators on when the ambient relative humidity was below 50%. Fig. 7 shows the number of hours per day that the regenerators could operate during the season. The figure shows that during the summer, the regenerators can operate close to 20 h in a day. During fall ambient conditions, the number of hours of

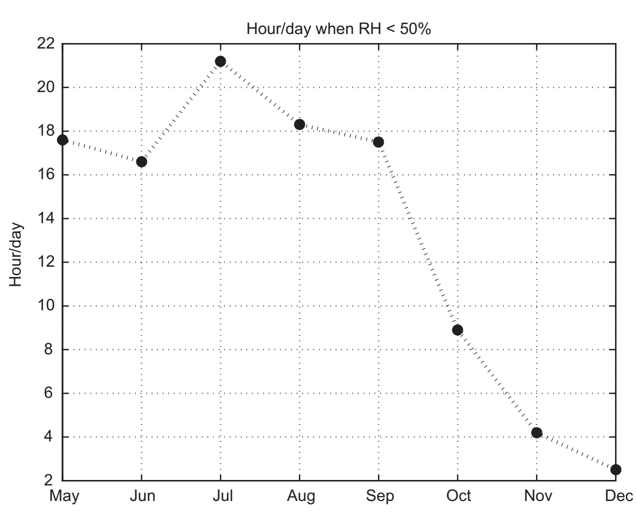

Fig. 7 – Hours per day suitable for liquid desiccant regeneration with ambient air at California's Central Valley. Regenerators are turned on when ambient RH < 50%.

regeneration is significantly reduced. When absorber and regenerators are operating, the level of liquid desiccant at the tank varies depending on the rate of water absorbed at the dehumidifier and rejected at the regenerator. Thus, the absolute rate of moisture absorption at the absorber was measured at a time when the regenerators were not running. Fig. 8 shows with a dashed line the status of the regenerators, i.e. on or off, together with the liquid-desiccant level at the storage tank. It is observed that as soon as the regenerators are turned off the level of the liquid desiccant shows a sharp increase due to the fact that the suction of the pump for the regenerators is no longer present. Then the level of the tank increases almost linearly with time. Based on the slope of the curve, it is estimated that the dehumidifier can absorb 0.008 kg min^{-1} of water, approximately for a liquid-desiccant concentration at the inlet of the absorber of 35% with a flow rate of 0.8 L per minute. At indoor relative humidities near 100% and colder liquid-desiccant inlet temperatures, the rate of water absorption has been observed to reach 0.017 kg min^{-1}. Therefore, the small-scale absorber processes an air-flow rate of 0.2381 $m^3\,s^{-1}$ and removes an average of 0.013 kg min^{-1} of water, approximately. Assuming a linear scale for a cold storage processing 40 $m^3\,s^{-1}$, the absorber would remove 3182 kg of water per day.

3.4. *Internal heat exchanger operation*

The absorption properties of the liquid desiccant improve as its temperature is lowered. The vapor pressure of the liquid desiccant tends to zero at low temperatures which provides a benefit in terms of increased water absorption rate and reduction in the potential for droplet carry over. Since the regeneration process is performed with ambient air, the inlet temperature of the liquid desiccant to the IHX remains between 18 and 25 °C for typical summer operating conditions, as seen in Fig. 9. Therefore, the temperature of the concentrated liquid desiccant, T^{in}_{IHX}, was lowered using the outlet stream of diluted liquid desiccant exiting the absorber at T^{out}_{Abs}. Fig. 9 shows that for ambient ranging between 20 and 38 °C, the conditions inside the cold storage generate an inlet air temperature to the absorber below 5 °C except during the defrosting cycles. Under these operating conditions, the IHX

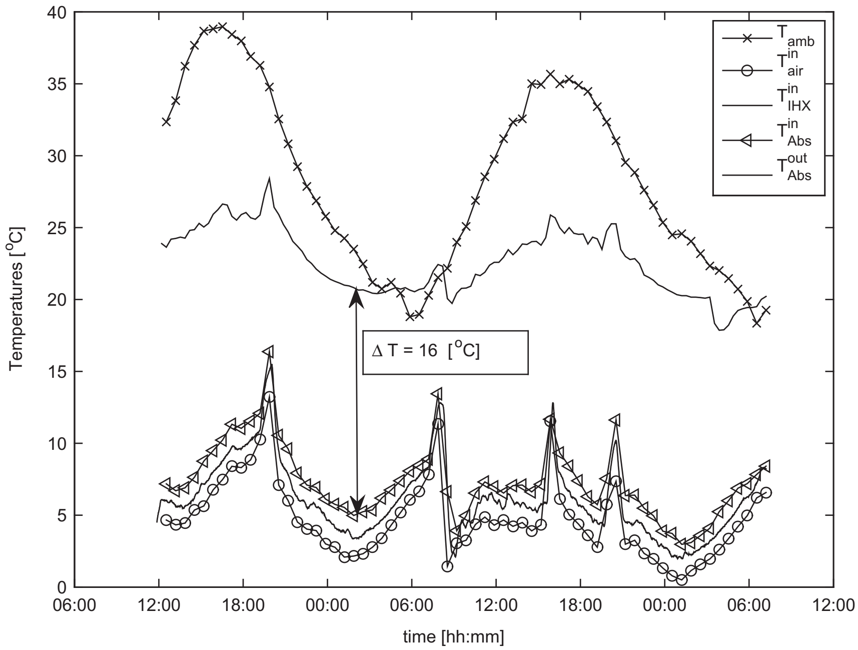

Fig. 9 – Internal heat exchanger performance.

lowers the temperature of the concentrated liquid-desiccant solution by 16 °C, approximately. Even though the liquid desiccant absorbs energy from the condensation of water vapor from the humid air, the low temperature conditions inside the cold storage imply that the outlet liquid-desiccant temperature from the absorber, T_{Abs}^{out}, is below T_{Abs}^{in} and therefore this stream can be used to lower T_{IHX}^{in}. An energy balance performed at the core shows that

$$Q = \dot{m}_{ld} Cp_{ld} \Delta T_{ld} = \dot{m}_a \left(Cp_a \Delta T_a + h_{fg} \Delta\omega \right). \tag{1}$$

In general, the results show that ΔT_{ld} and $\Delta\omega$ are negative and ΔT_a is positive. The uncertainty in the value of the heat transfer rate was ±15.4% and ±24.5% for the air and liquid desiccant sides, respectively. The liquid-desiccant uncertainty is larger mainly due to the ±0.3 °C of accuracy from the RTDs. This situation changes during transients where the thermal inertia of the absorber walls affects the energy balance such as during the start and end of defrosting cycles.

4. Potential water and energy savings

As seen in Fig. 2, each defrosting cycle uses 2.3 m^3 of water in this facility. The amount of water used is higher in cold storage rooms with larger evaporator coils. During the season of operation of a cold storage, that lasts between late May to early December, the number of defrosting cycles per day varies between 1 and 3. Taking an assumption of two defrosting cycles a day for six months of operation results in 828 m^3 of water used for one pre-cooling room. It is noted that according to California Energy Commission data, there are 246 refrigerated warehouses in the state many of which have more than one cold storage room in their facilities. Thus, the potential of reduction in water consumption is significant. However, it was mentioned that the conditions inside the refrigerated warehouse require high levels of air humidity so that if the defrosting cycles are significantly reduced then a water spray system would need to be added to the facility to conserve the quality of the produce.

The energy savings for a scaled-up system can be mainly associated with two effects related to eliminating or significantly delaying the formation of ice. After the defrosting cycle, the cooling coil surface temperature is higher than the design operating conditions of the evaporator so compressor power is required to lower its temperature after each defrosting cycle. Also, during periods where there is significant ice formation on the surface of the coil, the temperature inside the cold storage starts to rise, as seen in Fig. 5. The compressors tend to run more frequently as a reaction to the increasing value of the indoor temperature. The evaporating temperature decreases raising the power used by the compressor. Fig. 10 shows the coil temperature, T_{Coil}, the air inlet temperature, T_{Air}^{in}, and measurement of the temperature at the inlet manifold of the evaporator, $T_{Manifold}$. This temperature is not subject to the flow of air so it is closer to the refrigerant temperature at that location. The oscillations are due to the opening and closing of the valve that controls the refrigerant flow to the evaporator.

The commercial cold storage room has a total air-flow rate through the evaporator of 40 $m^3 s^{-1}$ with inlet air conditions of 1 °C and 85% relative humidity, and a design evaporating temperature of −4 °C. It is rated at 351 kW of cooling capacity operating with a COP of 2.2, approximately. Therefore, it utilizes 160 kW of compressor power and 21.9 kW of fan power under no-frost conditions. An absorber operating for 24 h a day can absorb 3182 kg of water. The extra air pressure drop due to the absorber is compensated by the lower pressure drop across the frost-free cooling coils. Ambient air would be used to regenerate the liquid desiccant. Considering ambient

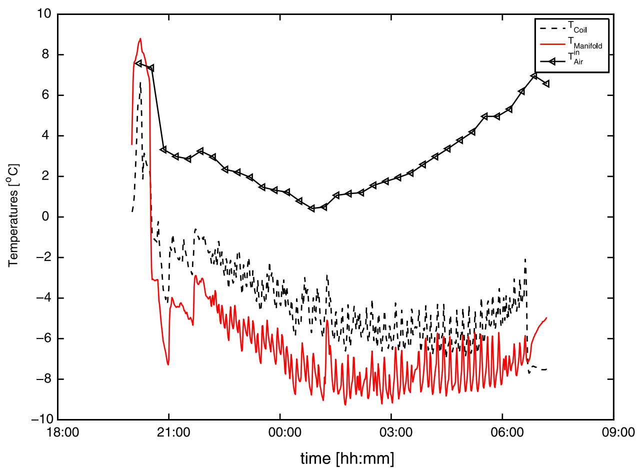

Fig. 10 – Inlet manifold, cooling coil, and inlet air temperatures inside the cold storage.

conditions of 35 °C and 20% RH, the regenerators are assumed to work for 6 h per day. In order to remove the amount of water required, a 0.15 m-deep regenerator with a face velocity of 2.1 m s^{-1} would require an air-flow rate of 72 m^3 s^{-1}. The fan power required to move the air would be 10.8 kW where the pumping power has been neglected. Fig. 10 shows that under frost conditions, the evaporating temperature of the refrigerant decreases and oscillates between −8 °C and −9 °C. Considering a constant condensing pressure of 1238 kPa and an isentropic compression efficiency of 75%, the reduction in evaporating temperature increases the compressor power by 16%. This increase in compressor power does not translate into a higher cooling capacity since the layer of ice acts as an insulator and the temperature inside the cold storage begins to rise, as observed in Fig. 10. The energy for lowering the coil temperature after each defrosting cycle based on the mass of the evaporator is estimated at 4.6 kWh. Thus, the net energy savings by not having frost formation on the surface evaporator are estimated at 16.3%. A more conservative 15% reduction in energy consumption and a rate of 13.54 cents kWh^{-1} imply that the cost savings during the year reach 11% of the installation cost for the liquid-desiccant system. Therefore, retrofitting of an existing cold storage would probably be not economically attractive, although more analysis is needed under real-time power pricing or at locations with higher power-generation costs. However, the application of this system to a new facility would significantly reduce the size of the evaporator by being able to reduce its fin spacing. The lower cost of the evaporator would offset the cost of the liquid-desiccant system.

5. Conclusions

The performance of a small-scale liquid-desiccant absorber subject to realistic operating conditions inside a commercial refrigerated warehouse has been studied. It is observed that the dew point temperature of the air downstream of the absorber was near or even below the measured surface temperature of the evaporator. This implies that a significant reduction in ice formation could be achieved with such a system. However, during operating conditions where a thick layer of ice was already formed on the cooling coil, the temperature of the evaporator surface was below the dew point temperature of the dehumidified air. The performance of an internal heat exchanger to lower the liquid-desiccant temperature has also been examined and it is noted that the regeneration process can be achieved only with ambient air. Analysis of the system indicates that a significant reduction of water consumption can be accomplished but the energy savings suggest that the system would be feasible for new installations instead of retro fitted ones.

Acknowledgments

This work has been supported by the California Energy Commission and the California Institute for Energy and Environment contract number MR-07-04. The authors would like to thank Aslan Cold Storage, LLC and Del Monte Foods for allowing access to their facilities to obtain measurements of realistic refrigerated warehouse conditions.

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