CLIMATE APPROPRIATE COOLING FOR A GROCERY STORE: HYBRID UNITARY DOAS SYSTEM IN SAN RAMON, CA

ET Project Number: ET12PGE3102

Product Managers: Marshall Hunt, Keith Forsman, Peter Biermayer, and Chris Li

Project Manager: Phil Broaddus
Pacific Gas & Electric Company

Prepared By: Jonathan Woolley and Robert McMurry
Western Cooling Efficiency Center
University of California, Davis
215 Sage St
Davis, CA 95616
wcec.ucdavis.edu

Issued: June 1, 2015

Pacific Gas and Electric Company®

ACKNOWLEDGEMENTS

Pacific Gas and Electric Company's Emerging Technologies Program is responsible for this project. It was developed as part of Pacific Gas and Electric Company's Emerging Technology program under internal project number ET12PGE3102. The University of California, Davis Western Cooling Efficiency Center conducted this technology evaluation for Pacific Gas and Electric Company with overall guidance and management from PG&E including Marshall Hunt, Keith Forsman, Peter Biermayer, Chris Li and Philip Broaddus. For more information on this report, contact Peter Biermayer at PJBN@pge.com

The project core team is grateful for collaboration and support from Munters and Whole Foods Market, who were encouraging and accommodating in all aspects of the research, and who embraced the mutual learning that developed through pilot evaluation and assessment of the innovative efficiency measure.

LEGAL NOTICE

ABBREVIATIONS AND ACRONYMS

ATS	PG&E Applied Technology Services
AHRI	Air-Conditioning Heating & Refrigeration Institute
ANSI	American National Standards Institute
Btu	British Thermal Unit - the energy required to raise 1 pound of water by 1°F
CFM	Cubic Feet per Minute
DOAS	Dedicated Outside Air System
DX	Direct eXpansion, as a descriptor for vapor compression air conditioning
EER	Energy Efficiency Ratio (defined by ANSI/AHRI 340/360)
EPX	Munters' proprietary indirect evaporative heat exchanger
HVAC	Heating, Ventilation and Air Conditioning
IEC	Indirect Evaporative Cooling (or Cooler)
IEER	Integrated Energy Efficiency Ratio (defined by ANSI/AHRI 340/360)
PG&E	Pacific Gas and Electric Company
RTD	Resistance Temperature Detector or Resistance Thermometer
Therm	A unit of energy equal to 100,000 Btu
WBD	Wet-bulb depression
WBE	Wet-bulb effectiveness
WCEC	UC Davis Western Cooling Efficiency Center
inWC	Inches of Water Column

FIGURES

TABLES

CONTENTS

EXECUTIVE SUMMARY

An inordinate amount of energy is wasted each year as a result of the fact that packaged commercial air conditioners are designed as 'one-size fits-all' products that can function in any climate and are – as a result – inefficient in every climate. Climate appropriate solutions recognize unique opportunities for efficiency that arise from the climatic patterns and characteristics in particular regions. In California, these solutions may include:

1. System designs and controls that avoid unnecessary dehumidification
2. Strategies that benefit from large diurnal temperature swings to reduce or eliminate mechanical cooling
3. Technologies that use water evaporation strategically to for substantial gains in cooling efficiency

The research reported herein directly supports California Energy Efficiency Strategic Plan goals to accelerate marketplace penetration of climate appropriate air conditioning technologies. The report presents results from a field assessment of a dedicated outdoor air supply (DOAS) air hander that uses both indirect evaporative cooling and vapor compression to cool ventilation air for commercial buildings. This hybrid system was installed for an existing food store in San Ramon, California in combination with a whole building systems controls revision, and a closed door medium temperature refrigerated case lineup. In the year since installation, the project has demonstrated 20% whole building peak demand reduction, and 20% annual energy savings.

The research work was executed by the UC Davis Western Cooling Efficiency Center and the PG&E Emerging Technology Program in collaboration with the major grocery chain, and with technical support from the technology manufacturer, and the project engineering, installation, controls and commission teams.

PROJECT GOALS

The main objectives of this project were to:

1. Provide reliable energy and water performance data for the technology in real world operation
2. Facilitate, review, and document application and operation of the technology.
3. Develop practical design guidance and recommendations based on in-field experience with the technology.
4. Assess the whole building energy savings achieved by the project

TECHNOLOGY REVIEW & RESULTS

The DOAS technology employs an innovative design that combines indirect evaporative cooling with vapor compression in a packaged unitary air handler. The equipment can be configured to use stale room air as the process air for indirect evaporative cooling – an arrangement that can be thought of as evaporatively-enhanced heat recovery ventilation. The system achieves excellent cooling performance – at peak, it promises to reduce whole building electrical demand for cooling and ventilation by more than 20%. At part load conditions the system can eliminate the need for compressor cooling altogether while operating with sensible system EER >20. The suite of efficiency measures advanced in this project reduced whole building annual energy consumption by 20%.

However, the research team also observed a number of challenges with implementation of the technology. These issues were mostly associated with building controls, and proper coordination with other cooling and ventilation equipment. Even after addressing these challenges, the measured performance for this machine does not match exactly with the laboratory performance results – the differences are attributed to the fact that the system was configured differently for the laboratory test than it was installed for field operation.

RECOMMENDATIONS

The research team strongly recommends that the technology be adopted by utility energy efficiency programs. The strategy is especially compelling since such substantial savings can be achieved in an existing building with the addition of a single machine that can work together with existing equipment. As is the case for most air conditioning technologies, we believe that some applications will be more appropriate than others, and suggest that programs adopting this technology be designed to avoid scenarios where overall energy performance is more limited. A simulation study should be conducted to assess the savings potential in a variety of applications and configurations, and to explain the differences between laboratory results and the field results presented here. The thorough dataset available from this study should be compared against simulation results for the same application in order to clearly validate the modeling approaches utilized.

INTRODUCTION

This report documents a field evaluation of the Munters EPX 5000 installed at a grocery store in San Ramon, CA. The technology studied is an indirect evaporative–vapor compression hybrid dedicated outside air supply (DOAS) system. The study directly in supports PG&E Emerging Technologies efforts to advance the California Energy Efficiency Strategic Plan goals related to broad marketplace application of climate-appropriate HVAC technologies.

Cooling and ventilation account for more than 25 percent of the annual electricity consumption for commercial buildings in California. If natural gas use is also considered, heating, cooling and ventilation typically account for more than 35% of the annual primary energy footprint for a commercial building (EIA 2012). Further, HVAC accounts for more than 30% of the greenhouse gas emissions associated with commercial buildings in California, amounting to statewide emission of more than 23 MMT CO_2e (CEC 2006, CARB 2014). Efficiency for these systems must improve in order to reach strategic energy and environmental goals, and state policy in response to climate change.

Moreover, air conditioning is the largest single contributor to peak electrical demand. Rooftop units are usually the largest single connected load in a commercial building, and can account for more than 50% of the on-peak demand from commercial facilities. California's electric grid is especially stressed during summer periods when generation requirements can be twice as high as other seasons. On the hottest summer days, air conditioning alone accounts for more than 30% of the peak demand on the statewide electric network (EIA 2014, CEC 2006). Grid management is anticipated to become more challenging as a larger number of intermittent renewable generators are brought on to the network. Since air conditioning loads are such a singularly large fraction of statewide demand, these systems will play a key role on the newly emerging paradigm of dynamic grid management.

Packaged rooftop air conditioners (RTUs) are predominately responsible for heating and cooling in commercial buildings. Rooftop units utilize technology that has not evolved to keep pace with the efficiency improvements that have been achieved for other key end-use systems – such as lighting. DOAS is one strategy to reduce energy use from these systems by separating ventilation supply from the management of sensible room cooling. This design strategy allows each system to be optimized for a specific role. The system technology studied in this evaluation uses a sophisticated indirect evaporative cooling system, which substantially reduces the need for compressor operation. By dividing responsibilities, each system may be optimized for the specific role that it plays in a building. The DOAS approach offers a number of opportunities for efficiency improvements that were heretofore not easily achieved with conventional rooftop units:

1. Heat recovery between incoming ventilation air and room temp exhaust can occur at a central location.
2. Low energy mechanical systems that focus on sensible cooling can be applied to manage room conditioning loads. This includes radiant cooling, chilled beams, ductless split systems, or forced air cooling with high sensible heat ratios.
3. Advanced evaporative cooling strategies can be utilized for ventilation cooling.

In California climates, DOAS equipment is especially well suited to employ indirect evaporative cooling. Whereas cooling capacity and efficiency of vapor compression systems generally decreases at high outdoor air temperatures, the capacity and efficiency for indirect evaporative cooling components typically increases. Subsequently, the largest incremental savings can be achieved when indirect evaporative is applied to cooling hot ventilation air. The technology also provides some room cooling and substantially reduces the amount of cooling that is required from compressor systems. Similarly, in humid climates DOAS equipment can employ highly efficient dehumidification components to relieve rooftop units, or other room cooling components, from the need to generate latent cooling.

This study focuses on application of the climate-appropriate DOAS in a grocery. Food stores are a significant commercial building energy use sector. In California, these buildings account for almost 10% of the electrical use, and roughly 3 percent of the natural gas use for all commercial facilities. Among all commercial buildings, food stores have the highest annual electric energy use intensity (CEC 2006). These facilities have large refrigeration needs, large exhaust airflows, and large ventilation rates. Luckily, there are broad opportunities to improve efficiency for various systems in grocery stores. For example, enclosing refrigerated cases could reduce refrigeration energy use by more than 40 percent (Fricke 2010, Hirsch 2012). A combination of strategic measures, including the type of air handler studied here, could reduce energy use in standard food stores by more than 50% (Hale 2008, Hirsch 2012, ASHRAE 2015). The DOAS strategy studied here is an especially appealing measure because it offers a route to major energy savings that does not require complete replacement of all existing air conditioning equipment in a facility.

The core of this study documents characteristic performance of the Munters EPX 5000 machine in each mode of operation. However, it is important to recognize that the DOAS machine is only one part of a whole-building HVAC design strategy. Other aspects of the whole-building system, such as controls and building air distribution characteristics, will impact degree of savings realized by the measure. Therefore, the study also reviews the overall design and application of the system, and presents clear recommendations as guidelines to help future projects achieve the greatest possible energy savings. In order to explore the whole building impacts of the measure, analysis incorporates all available information from the building EMCS systems, and the building electric meter.

Report Outline

Section *Project Overview* provides some background explanation about the "Climate Appropriate Cooling" concept, especially as it relates to California policy measures, then provides a detailed technical explanation of the technology that was studied in this project.

Section *Design and Application* outlines the recommended design practices for integration of the DOAS solution into a whole building HVAC system, and gives special attention to some of the design considerations unique to grocery stores. Incidentally, the way that the whole building ventilation flow was implemented for this project is no longer permitted by California's *Building Energy Efficiency Standards*, so we present a recommended alternative design concept.

The research team joined the project several months after the DOAS equipment had been installed and originally commissioned. Our initial diagnostics and review of the installation revealed that the new DOAS system had not been integrated with the whole building HVAC systems in an effective way. Subsequently, the research team facilitated a number of revisions to improve operations. These lessons learned are documented in section *Challenges with Implementation*.

Technical Approach & Test Methodology describes some of the physical details, scope, and process of the field evaluation. The section documents that range of measurements that were made, and discusses the methods utilized for analysis of data.

The *Results* section presents several quantitative metrics to describe performance of the Munters EPX 5000, including cooling capacity, coefficient of performance, supply air temperature, system power draw, and water use as a function of ambient conditions. Results also explain operation of this machine in sequence with other HVAC equipment, and document the difference in whole building energy use before and after the retrofits associated with this this project. The meaning and implication of results are discussed in subsections for each performance metric evaluated.

Lastly, the *Discussion and Conclusions* section synthesizes the major findings from this study, and outlines technical and non-technical recommendations for how we might advance successful market adoption this climate appropriate cooling technology and other measures that help to transform HVAC in California, in accordance with Energy Efficiency Strategic Plan goals and related State policy.

Project Objectives

The main objectives of this project were to:

1. Provide reliable energy and water performance data for the technology in real world operation
2. Facilitate, review, and document application and operation of the technology.
3. Develop practical design guidance and recommendations based on in field experience with the technology.
4. Utilize available data to capture information about whole building energy savings achieved.

To accomplish these objectives, the research team collaborated closely with the manufacturer, the grocery retailer's facilities team, the grocery's mechanical engineer, and various mechanical contractors. The research team facilitated commissioning of the air handler in sequence with other air conditioners in the building, then installed instrumentation to monitor performance for the hybrid air handler.

PROJECT OVERVIEW

CLIMATE APPROPRIATE COOLING

California's *Long Term Energy Efficiency Strategic Plan* outlines four major programmatic initiatives, as *"Big Bold Energy Efficiency Strategies*" to facilitate broad energy savings for our built environment:

- All new residential construction will be zero net energy by 2020
- All new commercial construction will be zero net energy by 2030
- HVAC will be transformed to ensure that its energy performance is optimal for California's climate
- All low-income customers will have the opportunity to participate in energy efficiency programs by 2020

The third initiative targets a 50 percent efficiency improvement for HVAC by 2020, and a 75 percent improvement by 2030. The plan recognizes that cooling and ventilation is the single largest contributor to peak electrical demand in California, which results in "*enormous and costly impacts on generation, transmission, and distribution resources as well as a concurrent lowering of utility load factors.*" Strategic goals to transform the HVAC industry focus on:

1. Code compliance
2. Quality installation and maintenance
3. Whole-building integrated design practices, and
4. Development and accelerated implementation of new climate-appropriate equipment and controls

The efficiency measure studied in this project specifically targets the fourth goal: it advances the evaluation and application of climate appropriate systems and controls. Air conditioning equipment has traditionally been designed and rated according to a single number efficiency metric that does not accurately represent the performance of air conditioners in California climates. Optimizing for this metric, manufacturers have mainly sold a single type of air conditioner that functions reliably in any climate, but is also inefficient in every climate. Luckily, there are many climate appropriate technologies and system design strategies that use far less energy than the "one-size fits all" approach. Climate appropriate air conditioning systems and controls are designed and tuned specifically for local climate conditions, and occupant comfort needs; they provide an equal (or better) quality of service with less energy input. Some examples of cooling strategies appropriate for California climates include:

- Sensible-only cooling measures that do not waste energy on unnecessary dehumidification
- Indirect evaporative cooling (and other evaporative measures), when water is used efficiently
- Advanced economizer controls, natural ventilation cooling, nighttime ventilation pre-cooling, and other passive or semi-active systems that capitalize on large diurnal outdoor temperature swings to reduce the amount of active cooling required at other periods.
- Adaptive comfort controls, and predictive control strategies that conserve energy by allowing indoor conditions to drift across a wider range, in concert with dynamic human comfort considerations.
- Any technology that uses substantially less energy for cooling (especially at peak) than the industry standard "*one size fits all*" minimum efficiency equipment.

Climate appropriate cooling technologies have reliably demonstrated peak demand reduction of more than 40% (Woolley 2013). Some solutions have shown annual cooling energy savings beyond 65% (Harrington 2015).

One should also note that current single number industry standard rating methods are generally not appropriate for describing performance of climate appropriate technologies. The problem is not that the limited range of standard test conditions are not exactly representative of every application in California; the issue is that the standard methods of test can actually portray climate optimized products as less efficient than traditional air conditioners. These standards unintentionally disadvantage climate appropriate strategies by misrepresenting their performance in comparison to the status quo. In many circumstances climate appropriate strategies cannot even be tested by industry standard methods because they operate in configurations that are fundamentally different than the scenario for which current standards were designed. This shortcoming is especially true for whole building integrated design practices.

The project reported here builds on a body of research, evaluation, and pilot demonstrations recently advanced by PG&E and other California entities to advance the understanding and market introduction of climate appropriate HVAC solutions. The findings from this project should guide the development and implementation of programs and policies designed to accelerate the broad and successful uptake of these solutions for new and existing buildings.

TECHNOLOGY OVERVIEW

Munters' EPX 5000 is a Dedicated Outside Air System (DOAS) that utilizes an indirect evaporative heat exchanger, plus vapor compression to cool ventilation air. When heating is required, ventilation air is tempered with a gas furnace or heating hot water coil. The EPX 5000 supplies 100% fresh air, however in many instances return air can be used as the secondary air stream for the indirect evaporative heat exchanger. In either configuration, Munters refers to the secondary air stream as the "scavenger" air stream.

The equipment utilizes Munters' EPX indirect evaporative heat exchanger – a polymer construction, tube-in-flow design that is arranged in a cross-flow orientation. The secondary air stream flows upward across the outside of these tubes. During cooling hours, water is sprayed into the secondary air stream over the top of the tube array. As water flows downward over the tubes, some evaporates into the secondary air stream, and the air and water are cooled. This results in sensible heat transfer away from the primary air stream, which flows through the inside of the tube array. The heat exchanger can use outside air in the secondary air stream, or in applications where expired room air can be returned to the air handler, cooling efficiency can be improved by utilizing this lower enthalpy source as the secondary air stream. Previous laboratory testing indicated that cooling capacity increases by roughly 20% at peak conditions when room air is used for the secondary air stream; efficiency increases concurrently by 43% (Woolley 2013). In the heating season, the return air as scavenger configuration has the added benefit that it allows for exhaust air heat recovery when the heat exchanger remains dry. As illustrated in Figure 1, for this project the EPX 5000 was configured to use outside air as the secondary air stream. This design decision is discussed with further detail in section: *Design and Application*.

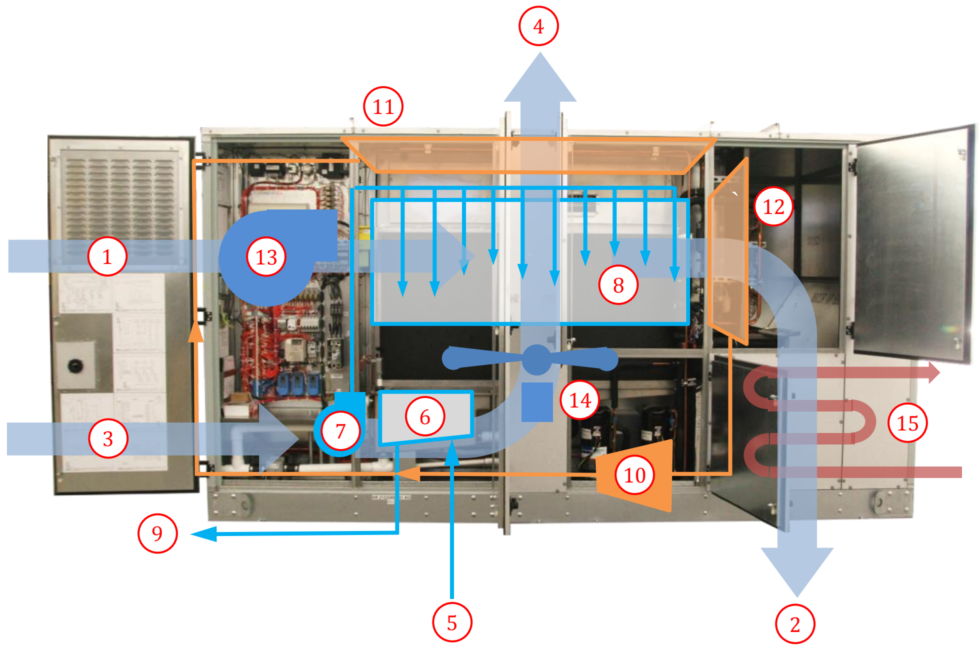
It is expected that indirect evaporative cooling mode will provide adequate cooling on its own during many hours of the year. However, when additional capacity is needed the EPX 5000 also includes two stages of vapor compression cooling. The condenser coil for the vapor compression system is located in the secondary air stream, downstream of the EPX heat exchanger. This location offers a thermodynamic benefit since the temperature of air at the secondary outlet can be significantly cooler than outdoors.

The EPX 5000 can be installed in a variety of different applications. For some buildings it may be desirable to supply room-neutral ventilation air, and to rely on parallel mechanical systems to condition sensible room loads. In other installations, it may be most efficient to allow the DOAS to cover a portion of the room loads, as long as the zone to which ventilation air is supplied is not overcooled. For example, in grocery stores where DOAS equipment is preferably installed above refrigerated cases, a low supply air temperature may overcool the local zone. For this and other reasons, the energy savings potential for this system depends on the configuration in which it is installed.

Generally, it is expected that the EPX5000 will not have adequate capacity to address all room cooling loads in a building. However, it does have significantly more cooling capacity than is needed to deliver room-neutral ventilation air. Previous laboratory testing showed that in western climate conditions, the EPX 5000 is usually more efficient at sensible room cooling than a conventional vapor compression system operating as recirculation only. Therefore, when room cooling capacity from the EPX 5000 can be used in place of less efficient cooling components in the building it will provide additional energy savings beyond its role as a ventilation cooling system.

When return air is used as the source for scavenger airflow, the equipment is intended to operate with a supply airflow rate of 5000 *cfm* and a secondary airflow rate of 4000 *cfm*. In applications where the EPX 5000 must rely on outside air for the scavenger airflow, the equipment is intended to operate with supply airflow rate of 5000 *cfm* and a secondary airflow rate of 6000 *cfm*. The added secondary airflow requirement in this configuration significantly reduces efficiency for indirect evaporative cooling.

The EPX 5000 is generally intended to provide continuous ventilation during all building operating hours, whether for heating or for cooling. When return air is used for scavenger flow, the secondary fan is intended to operate any time the supply blower operates, and both fans are intended to run at constant speed. However, both fans are variable speed components, so the EPX 5000 could be easily utilized for a demand controlled ventilation scheme, or in a scenario where fan speed is adjusted in response to room pressurization.



1	*Outside Air (Primary Inlet)*		6	*Sump*		11	*DX Condenser*	
2	*Supply Air*		7	*Circulation Pump*		12	*DX Evaporator*	
3	*Outside Air (Secondary Inlet)*		8	*EPX (Indirect Evap. Heat Exch.)*		13	*Blower (Primary Air)*	
4	*Exhaust Air*		9	*Drain*		14	*Fan (Secondary Air)*	
5	*Water Supply*		10	*Compressors (2 parallel circuits)*		15	*Gas Heat*	

FIGURE 1. CONCEPTUAL SCHEMATIC FOR MUNTERS EPX 5000

It is intended that the DOAS system should provide for all of the fresh air ventilation needs in a building. This approach is well suited to large retail stores and similarly scaled commercial facilities where displacement ventilation can be easily accomplished. However the approach can also be utilized in office buildings, schools, multi-family residential, or other buildings with large ventilation needs. In any scenario, the DOAS approach relieves other conditioning equipment from their role as ventilation systems. This means that other rooftop units should be set to operate as recirculation only, and that the fans on those systems need not operate unless there is a call for cooling from that zone. Furthermore, the DOAS approach also allows for the use of very efficient room cooling components, such as radiant cooling, chilled beams, or VRF systems.

It should be noted that there is no bypass for the EPX – both the primary and secondary airflows always pass through the EPX heat exchanger and across corresponding evaporator and condenser coils. This added resistance is a detriment to fan power during periods when ventilation does not require conditioning, or in instances when economizer-only cooling is desirable. Moreover, when room air is used as the scavenger air stream, although the unit provides heat recovery during heating periods, it also recovers heat when economizer cooling is desirable. The importance of these considerations will depend on the climate and application. It is anticipated that in most circumstances the energy savings for cooling will far outweigh these penalties.

Table 1 outlines the five modes of operation for the EPX 5000. In certain applications controls can be sequenced to shift between these modes in order to maintain a supply air set point temperature. For this project, the operating sequence was programmed to respond to room temperature. In either case, it is important that the whole building system be sequenced in a way that the most efficient cooling or heating sources are given priority. As discussed in section *Challenges with Implementation*, the control sequence initially deployed for this project failed to operate indirect evaporative cooling as the priority.

TABLE 1. MODES OF OPERATION

Mode	Primary Blower	Secondary Fan	Compressor 1	Compressor 2	Circulation Pump	Heat
Indirect Evaporative Cooling	ON	ON	OFF	OFF	ON	OFF
Indirect Evaporative & DX1	ON	ON	ON	OFF	ON	OFF
Indirect Evaporative & DX2	ON	ON	ON	ON	ON	OFF
Ventilation Only	ON	ON	OFF	OFF	OFF	OFF
Heating	ON	ON	OFF	OFF	OFF	ON

DESIGN AND APPLICATION

For the project reported here, the Munters EPX 5000 was installed to serve ventilation to the sales floor of a grocery. The system was installed to supply approximately 5,000 cfm of ventilation air at the back of the sales floor, which is intended to move by displacement across the sales floor and ultimately toward the kitchen exhaust. The DOAS system was added as a retrofit to an existing building with six existing rooftop units. Addition of the DOAS offset ventilation that was originally provided by the existing rooftop air conditioners. Since these rooftop units were not required for ventilation their outside air dampers were closed, and their supply fans – which typically run continuously for all operating hours – were switched to 'AUTO' mode. A significant amount of energy savings associated with this measure is captured from switching ordinary supply fans to 'AUTO' mode. If the fan runtime for all six rooftop units observed were reduced by 85%, this would save 52,000 kWh per year.

The food store is approximately 35,000 ft^2, with 18,000 ft^2 sales floor area, 4,500 ft^2 kitchen area, 9,000 ft^2 of unconditioned warehouse storage area, and 3,500 ft^2 administrative office space on the mezzanine. At a ventilation rate of 0.2 cfm/ft^2, the sales floor and office space require a total of 4,300 cfm outside air – well matched for the EPX 5000 which is designed to provide roughly 5000 cfm as DOAS.

However, like most supermarkets, this store has a large kitchen and bakery with several kitchen exhaust fans that operate continuously during operating hours. In total these exhaust fans draw 15,505 cfm. Air that is removed from the kitchen through these exhaust hoods must be replaced with an equal volume of outside air supplied to the building from elsewhere. Design guidelines for kitchens and restaurants regularly recommend the use of transfer air from the sales floor as makeup for kitchen exhaust. Maintaining a pressure gradient where air flows into the kitchen will avoid the migration of cooking effluent and kitchen smells into the occupied space (AEC 2002). For this project 5,886 cfm was supplied to the kitchen locally through a direct evaporative makeup air unit, and another 1,250 cfm of outside air was supplied through a conventional rooftop unit that supplied vapor compression cooling to the kitchen area. The remaining 8,377 cfm of makeup air was transferred from the sales floor. This outside air was supplied by the EPX 5000, two other rooftop units on the sales floor, and a rooftop unit serving the administrative office space. In order to provide enough makeup air for the kitchen exhaust, the EPX 5000 was not configured to use outside air as the secondary air stream instead of room air. Using room air improves the peak efficiency for indirect evaporative cooling by 40%, but requires that additional makeup air be provided elsewhere.

Figure 2 identifies the layout of HVAC equipment on the rooftop, and Table 2 documents the exhaust and ventilation air flow from each piece of equipment in the facility. Figure 3A illustrates the standard practice air balance design that was implemented for this project. By relying on transfer air from the sales floor as makeup air for kitchen exhaust, the design supplied roughly twice the code required minimum ventilation rate for the sales floor.

Incidentally, it should be noted that California's *2013 Building Energy Efficiency Standards* (CEC 2012) no longer allow for the makeup air strategy that was implemented for this project. Namely, the standards do not permit *increasing* the general ventilation supplied to a space *in order to* serve the makeup air requirements of a large kitchen exhaust system. Doing so introduces excess conditioning loads on the sales floor.

According to 140.9 (b) 2 - Kitchen Ventilation (CEC 2012):

- B. *A … facility having a total … kitchen hood exhaust airflow rate greater than 5,000 cfm shall have one of the following:*
    - a. *At least 50 percent of all replacement air is transfer air that would otherwise be exhausted; or*
    - b. *Demand ventilation system(s) on at least 75 percent of the exhaust air.*

      …

      …
    - c. *Listed energy recovery devices with a sensible heat recovery effectiveness of not less than 40 percent on at least 50 percent of the total exhaust airflow; and*
    - d. *A minimum of 75 percent of makeup air volume that is:*
        - i. *Unheated or heated to no more than 60°F; and*
        - ii. *Uncooled or cooled without the use of mechanical cooling.*

1. Each of six rooftop unit supply fans draw approximately 2 kW, and the scheduled operating hours for the store are 7:00 am – 9:00 pm, seven days each week. 85% annual fan runtime reduction is a rough estimate, and would vary by application. In the period of observation presented in this report, the rooftop units on the sales floor were practically never required.

For the grocery facility studied, only 28% of the replacement air is available transfer air that would otherwise be exhausted, all exhaust systems operate continuously, there is no heat recovery utilized, and only 38% of the makeup air volume is conditioned by a direct evaporative makeup air unit. Therefore, the approach demonstrated is not permissible for new construction. Going forward, we recommend an alternate design approach that would not increase the ventilation load for the sales floor for the sole purpose of balancing makeup air for the kitchen. Moreover, we recommend pursuing a building air balance design that:

1. Allows the Munters EPX 5000 to operate with room air as the secondary air stream.
2. Minimizes the amount of mechanically conditioned air that is exhausted through kitchen hoods.

The first strategy will increase peak efficiency for indirect evaporative cooling of ventilation air by 40%. The second strategy avoids excessive energy use for cooling outside air that is used as makeup air for kitchen exhaust.

With these design intents in mind, we suggest that about 20% of the ventilation air supplied to the sales floor should be used as transfer air to the kitchen, but that the rest should be returned to the DOAS equipment for exhaust heat recovery. Kitchen ventilation should rely on local makeup air sources, at least 75% of which should be unconditioned, or conditioned by direct evaporative makeup air systems. To the extent possible, makeup air sources should be passive. All makeup air sources should be configured in a way that:

1. Does not disrupt effluent capture effectiveness for exhaust hoods
2. Maintains comfort for kitchen staff
3. Minimizes air mixing between the kitchen area and sales floor

Best practices for design makeup air systems that meet these goals have been well documented elsewhere, including through the Food Service Technology Center's Commercial Kitchen Ventilation Design Guides (AEC 2002). Figure 3B illustrates the schematic air balance for an alternative recommended design. It should be noted that with this approach, both the sales floor systems, and the kitchen exhaust systems *could* operate as demand controlled ventilation without the complicated interactions normally associated with demand controlled ventilation when transfer air is used as makeup for kitchen exhaust.

FIGURE 2. HVAC EQUIPMENT LAYOUT (GOOGLE 2015)

TABLE 2. AIR FLOW RATES ACCORDING TO AIR BALANCE REPORT (A^2B 2014)

A

Minimum Required Ventilation Rates	
CA T24 120.1-A Required Ventilation Rate for Sales Floor (0.2 cfm/ft^2)	3,600
CA T24 120.1-A Required Ventilation Rate for Mezzanine	700
TOTAL	**4,300**

B

Actual Flow Rates	
Actual Continuous Exhaust from Kitchen	15,505
Actual Ventilation Rate on Sales Floor	8,377
Actual Makeup Air in Kitchen	7,133

C

Ventilation / Make-up Air (cfm)	
TOTAL	**15,510**
AC-29	1,350
AC-30	1,145
AC-31	0
AC-32	1,247
AC-33	0
AC-34	932
AC-35	0
MAU-1	5,886
AHU-1 (new)	4,950

D

Exhaust Air (cfm)	
TOTAL	**15,505**
EF1	1,668
EF2	1,606
EF3	2,066
EF4	1,815
EF5	918
EF6	1673
EF7	953
EF8	1,359
EF9	1,259
EF10	NA
EF11	221
EF12	NA
EF13	1,744
EF14	0
EF15	222
EF16	0

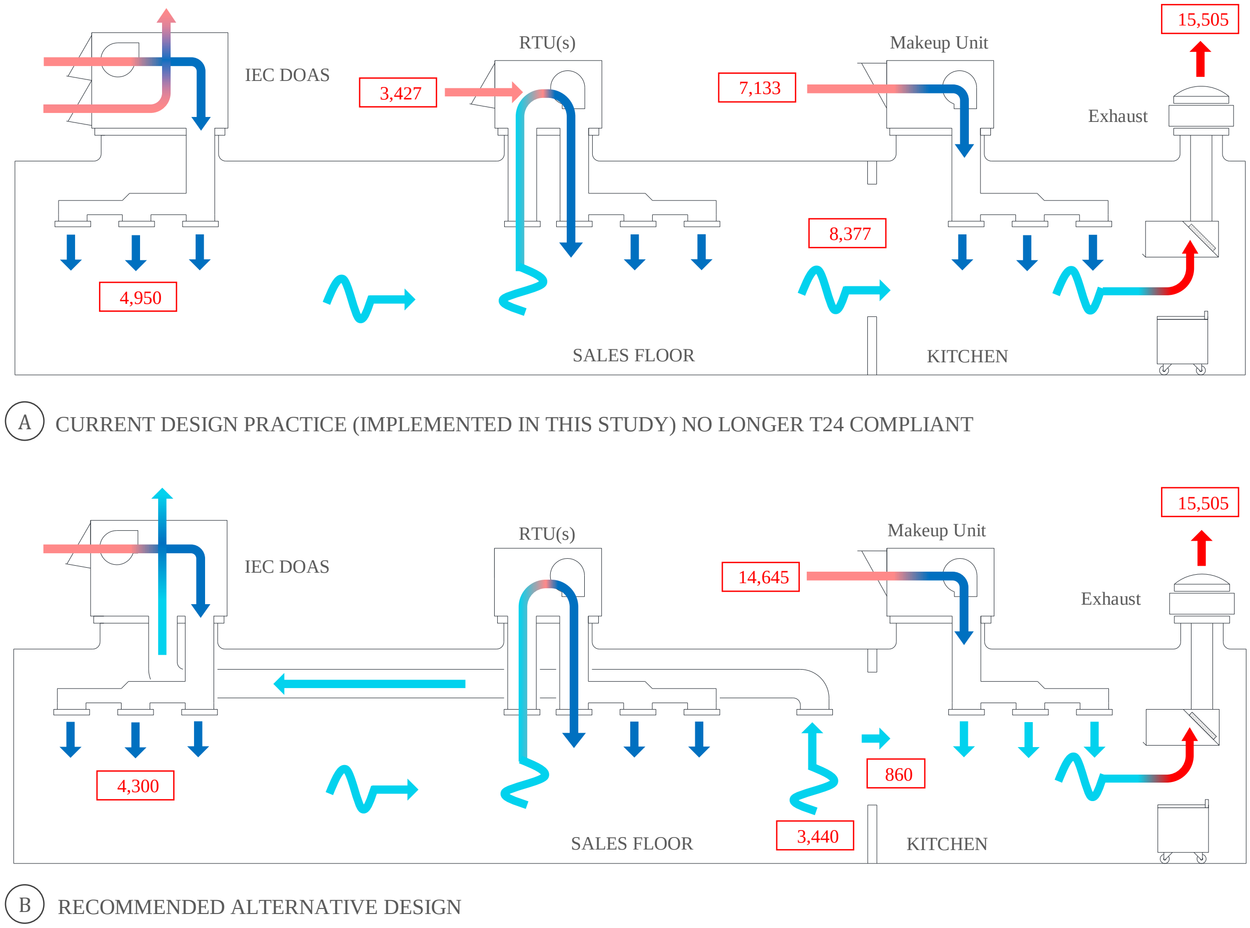


FIGURE 3. AIR BALANCE SCHEMATIC FOR (A) CURRENT DESIGN PRACTICE , AND (B) RECOMMENDED ALTERNATIVE DESIGN

CHALLENGES WITH IMPLEMENTATION

One intention for this project was to develop real world experience with the energy efficiency measure, and to highlight some the challenges experienced so that future installations might avoid likely hang-ups, so that the technology can be improved further, and so that programs might be designed in a way to facilitate broad and successful application of this and other climate appropriate measures.

There were a number of challenges associated with this installation. The equipment was installed and commissioned in January 2014 and operated for several months before measurement and evaluation began. When the study began in March 2014, the research team immediately found a number of problems with the way that the equipment had been setup. The challenges discussed here should not be considered as failures for the technology – per se; rather, these lessons learned should be thought of as guidelines for how to apply the strategy most effectively in future projects. After all, the measure is an emerging technology, and although it is technically mature, industry professionals are not yet well acquainted with its design, application, and upkeep.

Notwithstanding the fact that most emerging technologies suffer from similar challenges, the authors must stress that the energy and demand savings achieved by this measure will depend substantially on the quality of installation, commissioning, and ongoing service. Were it not for the persistent post-installation review and feedback from the research team to guide proper setup of the machine, the installation observed would not have resulted in any energy savings whatsoever. There were several significant lessons learned through the course of the project, including:

1. *A high efficiency cooling system will only save energy if it operates in place of less efficient equipment.*

   Although the unit had been scheduled to operate during all normal open hours for the store, initial observations indicated that the system only ever functioned as ventilation only, and never operated in any cooling mode. Instead, all cooling for the facility was carried by the conventional rooftop units. This was even true during hot afternoons, at which time the DOAS unit delivered unconditioned outside air to the floor and other units worked overtime to keep the store cool. This failure arose mainly from two factors:

   - The room temperature sensor used to control staging of the DOAS unit was not properly located. The sensor was originally located above the meat counter and in the vicinity of refrigerated cases. Since it was already cooler in this area, the controls never requested cooling from the EPX 5000, and other rooftop units on the store responded to keep other zones cool.

     Since the EPX 5000 is designed to move ventilation across the entire sales floor by displacement, it should not be treated the same as zone conditioning equipment. Cooling capacity supplied by the unit is carried over into downstream zones. The preferred approach is to measure temperature at the opposite end of the store. In theory this approach could result in overcooling of the areas into which the DOAS supplies air, especially if this ventilation air is supplied to areas with refrigerated cases, but no such problem was observed for this project.

   - The building controls did not prioritize cooling from the EPX 5000 ahead of cooling from other air conditioning equipment. As a result, a substantial portion of the building cooling load was carried by conventional cooling systems, and not by the most efficient means.

     In order to ensure that the EPX 5000 operates in place of less efficient systems, it must be controlled in a way that it operates as the first priority cooling source. There are various ways to give one system priority over others; we recommend a tiered set point schedule where the more efficient cooling mode is given a set point that is 3°F lower than the conventional equipment. This approach mirrors the staging methods that are normally used to stage cooling capacity for multi-compressor systems. There are also more sophisticated approaches if the application requires tighter set point control.

     The recommended practices were implemented, and the problems discussed here were resolved.

2. *Ventilation supplied by the DOAS should reduce conditioned ventilation provided by other equipment*

   While the Munters EPX 5000 had been scheduled to supply ventilation air to the sales floor during all open hours, no effort had been made to adjust the outside air fraction, or fan operation for any of the original air conditioners on the store. Instead, an antiquated direct evaporative makeup air unit in the kitchen had been shut off in order to maintain a rough balance between the overall building ventilation supply and exhaust. This approach resulted in two major consequences:

   - Since the ventilation rate from each rooftop unit remained the same, addition of the DOAS equipment actually increased the overall conditioning load for sales floor, and increased the amount of mechanically conditioned air exhausted through kitchen hoods. Since the building controls never requested cooling from the DOAS unit, the excess ventilation resulted in additional runtime for the original vapour-compression equipment

     Any retrofit project to install DOAS equipment should be coupled with an appropriately engineered whole building ventilation plan, and a complete test and balance of all ventilation and exhaust systems. The retrofit addition of DOAS equipment is necessarily a whole building system project, and not merely replacement of a single rooftop air conditioner. Special attention should be given to the role of transfer air for kitchen ventilation systems, as discussed previously in section *Design and Application*.

   - Even after a test and balance effort adjusted the ventilation rate for some rooftop units to zero, building controls continued to call for continuous fan operation from these systems during all occupied hours. Continued operation of these fans is not necessary, and represents more than 40,000 kWh annual waste.

While the Munters EPX 5000 provides obvious benefit for cooling efficiency, an important portion of the annual savings opportunity for this measure results from reducing the fan energy consumed by conventional rooftop air conditioners. In order to capture these savings, retrofit addition of a DOAS air handler requires revision to the whole building sequence of operations so that the fans in non-ventilation units only operate when heating or cooling is needed in the corresponding zone.

Ultimately, the recommended practices were implemented, and the problems discussed here were resolved.

3. *Active dehumidification is not needed for most commercial cooling applications in California.*

   Humidity management can be important for grocery stores, but when humidity control is deployed unnecessarily, or in an inappropriate way, it can result in substantial energy waste. In most California climates, outdoor humidity is always low enough that active dehumidification is not needed. In fact, although the vapour compression stages for the Munters EPX 5000 provide some dehumidification, the machine is not intended for use in an active humidity management role. This is an important departure from the role played by DOAS air handlers in more humid climates, where systems are designed and controlled primarily for humidity management.

   Although the manufacturer did not intend for the equipment to provide humidity management, the sequence of operations initially deployed for this project included a dehumidification mode which called for cooling and heating simultaneously. The control scheme responded to an indoor dew point temperature set point, and requested simultaneous cooling and heating to dehumidify air when room cooling wasn't required. This occurred on most mornings, and sometimes persisted for up to two hours. In some instances, the system was caught in a circle of inappropriate operation where it would:

   1. Determine that dehumidification is needed but that cooling is not needed
   2. Operate indirect evaporative cooling and heat simultaneously
      (This does not provide dehumidification, but does heat the space unnecessarily)
   3. Determine that dehumidification is needed and that cooling is now also needed
   4. Operate as indirect evaporative only until the indoor dry bulb set point is satisfied

   RETURN TO 1

   At other times, staging in response to the dew point measurement would ultimately call for vapour compression cooling concurrent with heating. This did provide some dehumidification, but only at great energy expense.

   Although San Ramon is considered part of a hot dry inland climate, it does experience more marine influence than other regions within the same climate zone, and many mornings in the summer can be warm with relatively high humidity. It is not unreasonable to expect that some dehumidification might be needed in this application – in fact the indoor humidity often rose above the 50°F dew point set point programmed for humidity controls. However, prior to addition of the DOAS equipment, the grocery store did not have controls for active humidity management, and indoor humidity had never been a problem.

   The Munters EPX 5000 is intended to condition ventilation air in climates and applications that do not require active humidity control. Although the system has some latent capacity, it is not optimized for dehumidification, and should not be used for that purpose. If active humidity control is necessary, there are more efficient systems for the job. After these facts were reviewed with the manufacturer, engineer, facility manager, and controls contractor, the recommended practices were implemented and the problem with simultaneous cooling and heating was resolved. No indoor humidity concerns have been observed, and the max indoor dew point observed since then was 60°F.

4. *Quality installation and maintenance practices impact energy performance for the whole building system.*

   Many studies have demonstrated that installation and maintenance practices for conventional rooftop units can have major implications to the overall HVAC energy use for commercial buildings. Although conventional rooftop units operate far less often with addition of the Munters DOAS system; airflow issues, economizer controls, refrigerant charge, and other 'normal' faults still impact overall HVAC system energy consumption. There were two main challenges encountered for this installation:

- Since DOAS equipment moves 100% outside air, air filters for this equipment can become soiled more quickly than filters for traditional systems that supply ventilation as a fraction of the supply air stream. The proper filter change schedule will depend on application, and the number of daily operating hours, but Munters generally recommends monthly filter replacement. In the initial months of operation, the service contractor responsible for filter changes did not maintain this schedule, and after approximately three months of operation, the filters were so heavily soiled that airflow declined and the supply fan faulted on a low static pressure alarm. In fact, when the research team first inspected the installation the unit was not operating as a result of this fault.

- Although each of the existing rooftop air conditioners on this store had outside air damper systems and economizer controls, none of the economizers were functional. Some of the motor and damper systems were seized, while other economizers had been physically disconnected. There are a many operating hours in San Ramon when economizer operation would serve as a valuable cooling mode in a grocery, and the value of economizers in grocery stores is confirmed by a variety of literature (Wang 2011, Hirsch 2012). In fact, when outside air temperature is low enough for economizer cooling, economizer is even more efficient than indirect evaporative cooling. Ideally, economizers ought to be utilized where possible – their value increases when refrigerated cases are enclosed and the building is not overcooled by refrigeration.

  To remedy this challenge, the research team facilitated the installation and commissioning of new differential-enthalpy integrated economizer controls for all six conventional air conditioners on the grocery store. However, despite this effort, there was not much opportunity for these controls to provide value because the hybrid DOAS equipment was typically able to cover all cooling needs on the sales floor for periods when outside air temperature is cool enough for economizer operation. Since economizer cooling is the most efficient option for cooling, the optimal approach would be to have integrated systems control scheme that utilizes economizer cooling as the priority when outside temperature is below room temperature, and then, when outside temperature is above room temperature, utilizes indirect evaporative and indirect evaporative + vapour compression as the first priority for cooling.

In summary, while the Munters EPX 5000 demonstrates admirable performance, the degree of energy and demand savings realized for real world projects that deploy climate appropriate DOAS strategies like this will depend largely on the design, installation, commissioning and maintenance of the equipment. The hybrid DOAS strategy serves an important role as part of a smartly integrated whole building system, but must be integrated properly. This project experienced a number of challenges that would have resulted in zero savings – or even increased energy use for the building – were it not for persistent attention to ongoing system commissioning. These challenges underscore the need for clearer design guidelines, broader familiarity with the measure, and better procedures and processes to facilitate proper application of the technology. Specific methods to achieve these things might include increased manufacturer involvement in the whole system integration efforts, standardization of proper sequences of operation, development of best practice design guidelines, or improved professional training resources surrounding climate appropriate cooling strategies.

TECHNICAL APPROACH & TEST METHODOLOGY

FIELD TESTING OF TECHNOLOGY

The overarching intent of this pilot was to explore, evaluate, and advance the application of an innovative hybrid unitary DOAS product that utilizes indirect evaporative cooling and vapor compression to treat ventilation air. The technology was laboratory tested by PG&E and UC Davis in 2013 (Woolley 2014). This field installation is a direct follow on from that laboratory study. The field study focused on evaluation of characteristic performance for the hybrid machine in all modes of operation, explored the interactions with other equipment on the building, and utilized interval metering data to describe whole building energy use before and after the suite of measures that were installed in association with this study.

The quantitative results presented here provide a clear map of performance for the Munters EPX 5000 in all modes of operation. This characterization should serve as the basis for building energy simulations and other analysis designed to project savings in various applications and climates. Results from this field evaluation generally corroborate conclusions from laboratory tests which estimate the measure could reduce electric demand from cooling and ventilation by as much as 20% (Woolley 2014). However, there are significant differences in performance that appear to result from differences in system configuration. Whole building electricity data was used to develop a pre-post estimate of annual energy savings, however there were other major efficiency measures installed at the same time, which makes it difficult to identify the portion of annual energy savings that can fairly be attributed to the hybrid DOAS system.

PG&E considered a variety of field demonstration sites, and selected the food store based on the willingness of all parties to cooperate in application and study of the technology. Food stores represent a unique energy end user; they incorporate a variety of energy systems from refrigeration to specialty lighting, and have long operating hours compared to other commercial buildings. As a result, among all commercial building types, food stores have the highest annual electric energy use intensity (CEC 2006). Grocery stores also have unique design considerations for which a DOAS ventilation approach can be well suited.

The food store is approximately 35,000 ft^2, with 18,000 ft^2 floor sales floor area, 4,500 ft^2 kitchen, 9,000 ft^2 warehouse storage area, and 3,500 ft^2 mezzanine office space. Figure 4 provides an overhead view of the site prior to installation of the DOAS. Figure 5 identifies the site location on a regional climate zone map. San Ramon is located in California Climate Zone 12, in the San Ramon Valley on the inland side of the Contra Costa Range and just immediately southwest of Mt Diablo. The site experiences much warmer summers than locations only a few miles further west. However, the site is also subject to some marine influence, especially as a result of its location at the eastern mouth of Crown Canyon which can funnel fog from the San Francisco Bay. As a result, summer mornings are often cool and foggy, but warm rapidly. Average daily high temperature from June – September is 85–90 °F, though daily high temperature in the mid-nineties occurs regularly, and record highs are above 110 °F (NOAA 2015). The 0.4% design condition for the region is 98.8 °F and 69.2 °F mean coincident wet bulb (ASHRAE 2009). Daily low temperature during the summer months is typically 50–55°F, with high relative humidity. The region rarely freezes; temperature December–February normally ranges between 35–60 °F.

To execute installation of the measure, PG&E facilitated cooperation between the manufacturer, the customer, the customer's design–build mechanical contractor, and the customer's controls team. UC Davis provided initial design consultation to introduce the customer and mechanical team to the technology, then later facilitated a commissioning and controls revision effort – the results of this effort are documented in section *Challenges with Implementation*. In May 2014, UC Davis installed a thorough suite of instrumentation to monitor thermodynamic performance of the unit, then cooperated with the customer to access data from two building EMCS systems and the facility electric meter. The later data streams allowed for some analysis of the whole-building impacts of the measure.

Data for the Munters EPX 5000 was collected on one-minute intervals starting the first week in June 2014. Data collection persisted through the conclusion of the project. Data from the building EMCS and electric meter was recorded on 15 minute intervals. The results for characteristic system performance focus on data from October 2014 because some of the system controls challenges discussed in section *Challenges with Implementation* persisted until September 2014. Plenty of characteristic performance data for particular modes of operation is available from the entire monitoring period from June 2014 – March 2014. The annual savings estimates and peak demand reduction values are based on two years of pre-retrofit data, and one full year of post retrofit data.

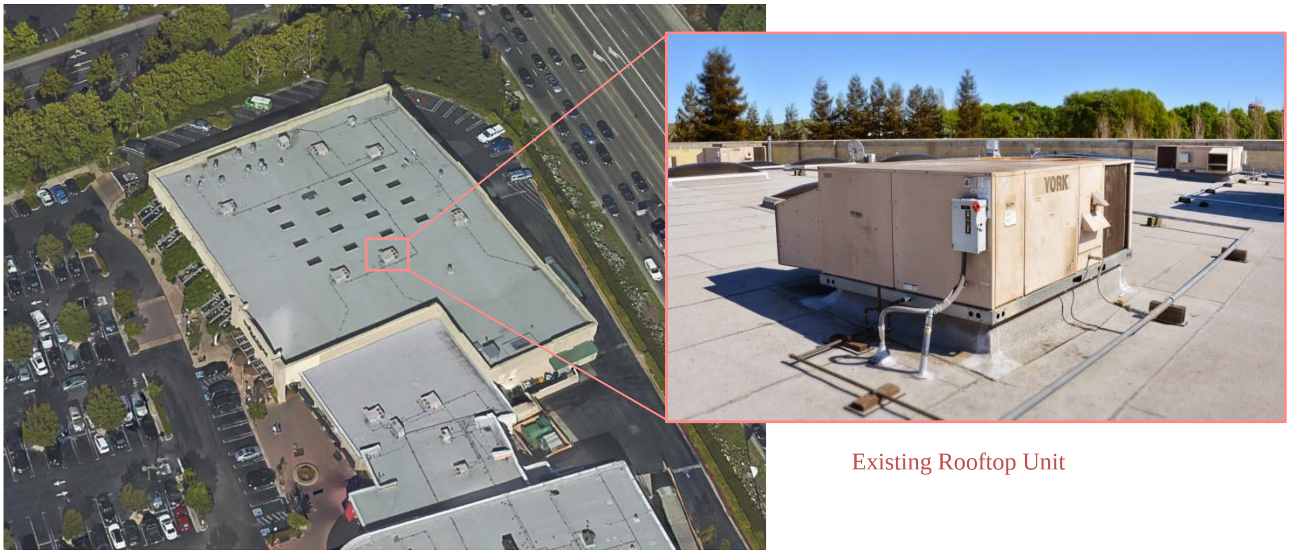


FIGURE 4. SITE OVERHEAD AND PHOTO OF EXISTING ROOFTOP UNITS (GOOGLE)

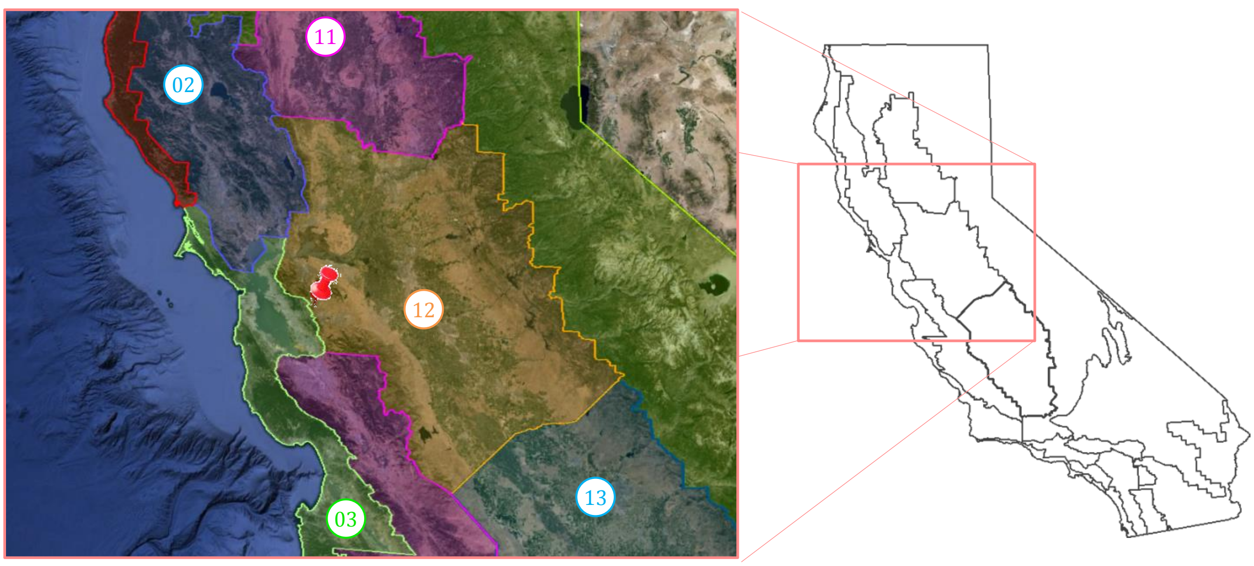


FIGURE 5. FIELD TEST SITE LOCATED IN CALIFORNIA CLIMATE ZONE 12 (GOOGLE)

MONITORING PLAN

The research team developed a monitoring plan that allowed for (1) assessment of overall performance for system inputs and outputs (2) evaluation of sub-component performance characteristics. The study also collected data about other rooftop units on the building, and about whole building energy consumption, in order to characterize building scale effects. The monitoring scheme utilized for the study is illustrated schematically in

Figure 6. Figure 7 identifies several key measurements that were taken from the building energy management and control system. Table 3 provides a simple description of each measurement marked in the instrumentation schematic, and documents the performance specifications for the sensors utilized for each corresponding measurement. Some important attributes about the monitoring approach include.

- Current transducers listed in the monitoring plan are used mainly for sensing component operations to determine system mode, not for determining power consumption.
- System amperage, line voltage, and power factor are recorded to accurately determine the total power draw for the unit in each minute of operation.
- All temperature and humidity measurements identified in the schematic represent single-point measurements and not space averages.
- Outside air temperature is measured in a radiation shield, and immediately next to the outside air intake.
- Room air temperature and humidity was measured approximately 15' above the floor, at the back of the sales floor, immediately beneath the EPX 5000.
- All refrigerant temperature measurements use surface mounted thermistors beneath >1" insulation.
- All refrigerant pressure measurements use Schrader valve pressure transducers placed on the system's existing service ports.
- ΔP_{FAN} measures the differential pressure across the fan inlet ring. This measurement is mapped against tracer gas supply airflow rate measurements to determine supply airflow rate in every minute – as discussed in section *Supply Fan Airflow and Power Measurements*.
- The supply fan speed is manually set by a potentiometer. AO_{F1} measures the analog signal from this potentiometer, in order to ensure that it did not change throughout the study.
- Measurements collected from the building energy management and control system were recorded through a separate data acquisition system, on different times intervals, and were communicated to the research team by email.

Analog and digital measurements from the unit were collected by a data acquisition module located on board the equipment. The data acquisition connects wirelessly to the EDGE cellular network and communicates records to an SFTP sever hosted by UC Davis. One minute interval data was collected from the system over the course of the study, with minor gaps during any period when the equipment was shut down for service, or for the purposes of diagnostic measurements. The minute interval data was stored on board the data acquisition module for 24 hours, then automatically uploaded over the EDGE cellular network to the SFTP server. Data for each unit is collected on this server as a separate CSV file each day.

Raw day-by-day datasets for each unit were concatenated into larger datasets that group minute interval data into month long time series sets. These month-long files were then used as manageable chunks for further data analysis and visualization. One minute increment data was collected from June 2014 through until the conclusion of this study. As discussed previously, results to describe characteristic performance of the unit are presented for October 2014. Results for demand savings and whole building energy savings consider all months of the year.

Data analysis and visualization was conducted using custom developed algorithm for data processing in Python (Rossum). The tools and functions available in Python are especially well-suited for manipulation and analysis of large time series datasets. This analysis separates data by distinct operating modes and filters data to extract performance results for periods of steady state operations. The research team also developed a library of psychrometric functions for Python, as well as an array of calculators for common analysis metrics such as cooling capacity, energy and water use efficiency.

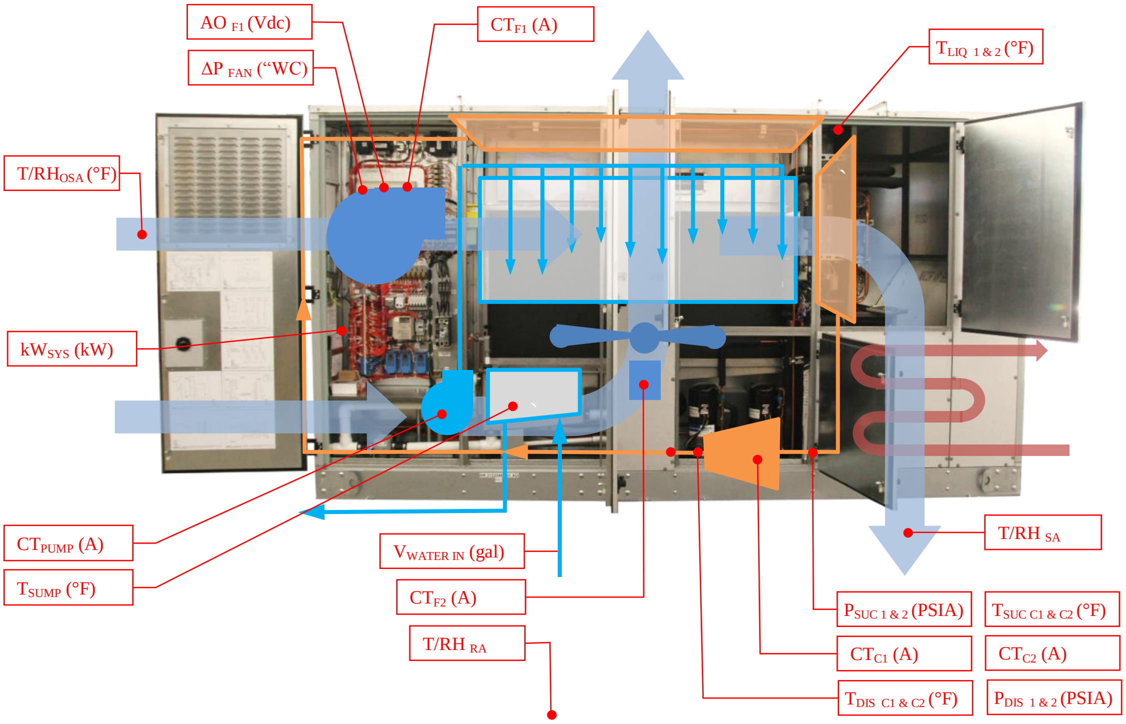


FIGURE 6. INSTRUMENTATION SCHEMATIC FOR MONITORING OF HYBRID DOAS

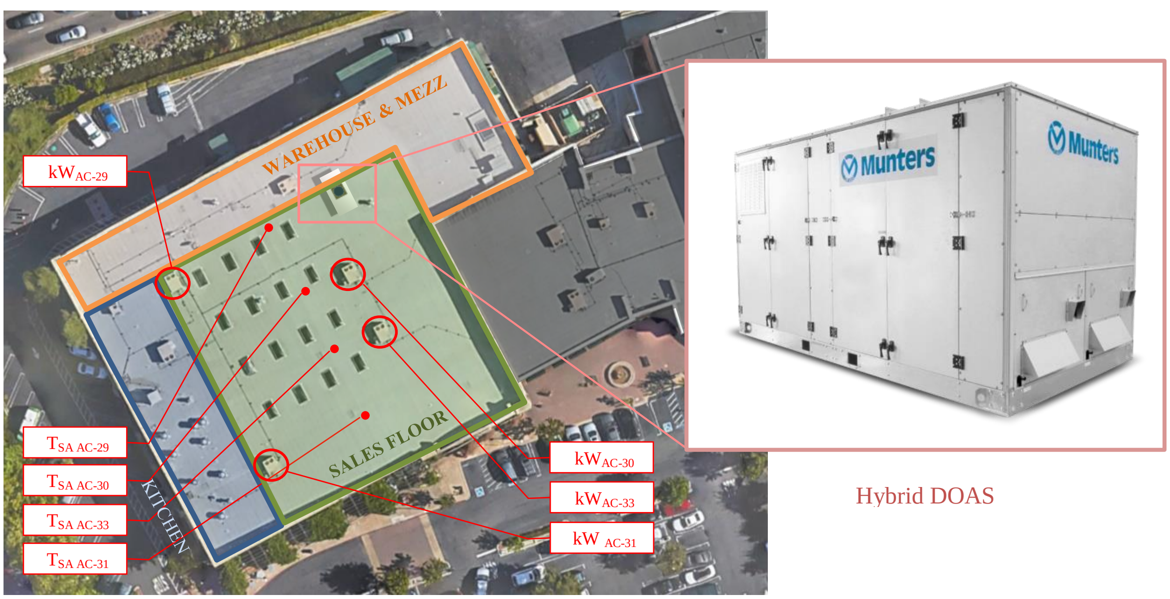


FIGURE 7. RELEVANT MEASUREMENTS COLLECTED FROM BUILDING EMCS AND PHOTO OF HYBRID DOAS

Table 3. Instrumentation Schedule for Hybrid DOAS

Measurement	*Description*	*Device*	*Details*	*Uncertainty*
T_{OSA}	Outside Air Temperature	Vaisala HUMICAP HMP110	-40°F ≤ T_{OSA} ≤ 176°F	± 0.36 °F
RH_{OSA}	Outside Air Relative Humidity	Vaisala HUMICAP HMP110	0% ≤ RH_{OSA} ≤ 100%	± 1.7% RH
T_{RA}	Room Air Temperature	Vaisala HUMICAP HMP110	-40°F ≤ T_{OSA} ≤ 176°F	± 0.36 °F
RH_{RA}	Room Air Relative Humidity	Vaisala HUMICAP HMP110	0% ≤ RH_{OSA} ≤ 100%	± 1.7% RH
T_{SA}	Supply Air Temperature	Vaisala HUMICAP HMP110	-40°F ≤ T_{OSA} ≤ 176°F	± 0.36 °F
RH_{SA}	Supply Air Relative Humidity	Vaisala HUMICAP HMP110	0% ≤ RH_{OSA} ≤ 100%	± 1.7% RH
$\dot{V}_{WATER\ IN}$	Water Consumption	Badger PFT 420-25 RCDL	5/8" , 198.4 PPG	± 0.5% msmt
AO_{F1}	Primary Blower Speed Signal	Analog Output	0–10 V_{DC}	± 0.1V
CT_{F1}	Primary Blower Current	NK AT1-005-000-SP	0-10,0-20,0-50 A_{AC} to 0-5 V_{DC}	± 0.1 A
CT_{F2}	Secondary Fan Current	NK AT1-005-000-SP	0-10,0-20,0-50 A_{AC} to 0-5 V_{DC}	± 0.1 A
CT_{C1}	Compressor 1 Current	NK AT1-005-000-SP	0-10,0-20,0-50 A_{AC} to 0-5 V_{DC}	± 0.1 A
CT_{C2}	Compressor 2 Current	NK AT1-005-000-SP	0-10,0-20,0-50 A_{AC} to 0-5 V_{DC}	± 0.1 A
CT_{PUMP}	Pump Current	NK AT1-005-000-SP	0-10,0-20,0-50 A_{AC} to 0-5 V_{DC}	± 0.1 A
T_{SUMP}	Sump Water Temperature	Omega HSTH-44031-120	Sealed 10kΩ Th. Herm. Seal	± 0.36 °F
KW_{SYS}	System Power Draw	Dent Powerscout 3	RS485	± 1%
$T_{SUC\ 1}$	Suction Temperature (Ckt 1)	Omega SA1-TH-44006-120-T	10kΩ Th. Surface Mount	± 0.36 °F
$T_{SUC\ 2}$	Suction Temperature (Ckt 2)	Omega SA1-TH-44006-120-T	10kΩ Th. Surface Mount	± 0.36 °F
$T_{DIS\ 1}$	Discharge Temperature (Ckt 1)	Omega SA1-TH-44006-120-T	10kΩ Th. Surface Mount	± 0.36 °F
$T_{DIS\ 2}$	Discharge Temperature (Ckt 2)	Omega SA1-TH-44006-120-T	10kΩ Th. Surface Mount	± 0.36 °F
$T_{LIQ\ 1}$	Liquid Temperature (Ckt 1)	Omega SA1-TH-44006-120-T	10kΩ Th. Surface Mount	± 0.36 °F
$T_{LIQ\ 2}$	Liquid Temperature (Ckt 2)	Omega SA1-TH-44006-120-T	10kΩ Th. Surface Mount	± 0.36 °F
$P_{SUC\ 1}$	Suction Pressure (Ckt 1)	ClimaCheck, 35 *bar*	0-35 *bar* to 1-5 V_{DC}	± 0.35 *bar*
$P_{SUC\ 2}$	Suction Pressure (Ckt 2)	ClimaCheck, 35 *bar*	0-35 *bar* to 1-5 V_{DC}	± 0.35 *bar*
$P_{DIS\ 1}$	Discharge Pressure (Ckt 1)	ClimaCheck, 50 *bar*	0-50 *bar* to 1-5 V_{DC}	± 0.50 *bar*
$P_{DIS\ 2}$	Discharge Pressure (Ckt 2)	ClimaCheck, 50 *bar*	0-50 *bar* to 1-5 V_{DC}	± 0.50 *bar*
ΔP_{SF}	Fan Inlet Pressure	Dwyer Series 668-4 0-2.5"WC	0-2.5" WC	± 1%

Data Analysis

Airflow Measurements

Supply airflow rate across a range of fan speeds was determined using a tracer gas airflow measurement, conducted according to ASTM E2029 *Standard Test Method for Volumetric and Mass Flow Rate Measurement in a Duct Using Tracer Gas Dilution* (ASTM 2011). This method mixes a measured mass flow rate of CO_2 into the supply air stream then measures the corresponding rise in CO_2 concentration downstream. The volume flow of air into which the tracer is mixed can be calculated by the following relation:

$$\dot{V}_{Airflow} = \frac{\dot{V}_{CO_2}}{C_{CO_2\,downstream} - C_{CO_2\,background}} \tag{1}$$

This method has many advantages compared to conventional air balance techniques, the most significant of which is accuracy. The tracer gas airflow tool used can operate with a calculated uncertainty of less than ±2%.

Calculating Cooling Capacity

The system-cooling-capacity for the air handler is determined at any operating condition according to the supply airflow rate and the specific enthalpy difference between the outside air stream entering the system and the supply air stream, as described by Equation 2. This is the net cooling produced by the equipment, including what is lost due to fan heat.

$$\dot{H}_{system} = \dot{m}_{SA} \cdot \left(h_{OSA} - h_{SA}\right) \tag{2}$$

The assessment presented here focuses on the system's ability to produce sensible cooling, and discounts the value of any dehumidification. As discussed in section *Challenges with Implementation*, dehumidification is not necessary for most commercial cooling applications in California. Although grocery stores have tighter humidity tolerances than other commercial buildings, active humidity management was not required for the grocery store in this study. Since the thermostat controls only respond to temperature and do not control for humidity, it is not appropriate to give value to any latent cooling in analysis of performance for the DOAS machine. The energy associated with any latent cooling is completely wasted. The net sensible system cooling capacity is determined according to Equation 3:

$$\dot{H}^{sensible}_{system} = \dot{m}_{SA} \cdot C_p \cdot \left(T_{OSA} - T_{SA}\right) \tag{3}$$

Concomitantly, the latent system cooling is determined as:

$$\dot{H}^{latent}_{system} = \dot{H}_{system} - \dot{H}^{sensible}_{system} \tag{4}$$

This study also presents results for sensible room cooling capacity. This metric describes the equipment's net contribution to thermal energy in the room. In some scenarios DOAS equipment might supply room-neutral air, in which case the room cooling capacity would be zero, even though the system cooling capacity may be substantial.

$$\dot{H}^{sensible}_{room} = \dot{m}_{SA} \cdot C_p \cdot \left(T_{RA} - T_{SA}\right) \tag{5}$$

Understanding the room cooling capacity characteristics for a machine is important fort thinking about the role that the system plays in the whole building in different instances. Generally, in mild periods a system may provide enough room cooling to cover all loads, while during warmer periods – even though the system cooling capacity is larger – it will have a smaller contribution to room cooling.

Calculating Coefficient of Performance

Energy efficiency at any given operating condition is expressed as the dimensionless ratio of useful thermal capacity delivered to electrical power consumed by the system – the Coefficient of Performance:

$$COP = \frac{Thermal\ Energy\ Delivered}{Electrical\ Energy\ Consumed} = \frac{\dot{H}}{\dot{E}_{system}} \tag{6}$$

Analysis in this report focuses on the sensible cooling generated by the equipment. This approach discounts the enthalpy associated with reduced humidity, and accounts for the net cooling energy generated by the system, which is larger than the amount of cooling actually delivered to the room. The Sensible System Coefficient of Performance can be expressed as:

$$COP_{system}^{sensible} = \frac{\dot{H}_{system}^{sensible}}{\dot{E}_{system}} \tag{7}$$

Similarly, the sensible room coefficient of performance is defined as:

$$COP_{room}^{sensible} = \frac{\dot{H}_{room}^{sensible}}{\dot{E}_{system}} \tag{8}$$

Calculating Wet Bulb Effectiveness

Wet bulb effectiveness (WBE) measures the extent to which an evaporative system is able to cool toward the wet bulb temperature of the inlet air. For simple direct evaporative systems, this metric tends to remain steady for a given system configuration even while meteorological conditions and system cooling capacity vary. WBE is the most common metric to describe performance of evaporative systems and is used as an input for models of evaporative cooling systems in most building energy simulation tools.

$$WBE = \frac{T_{DB\ 1st\ inlet} - T_{DB\ 1st\ out}}{WBD_{1st\ inlet}} = \frac{T_{DB\ 1st\ inlet} - T_{DB\ 1st\ out}}{T_{DB\ 1st\ inlet} - T_{WB\ 1st\ inlet}} \tag{9}$$

The metric has traditionally been used to describe performance of direct evaporative coolers, but it can also be applied to indirect evaporative equipment. Since indirect evaporative heat exchangers use a secondary air stream that can have an inlet wet bulb temperature that is lower than that of the primary inlet, it is possible to achieve better than 100% effectiveness.

Describing performance in terms of wet bulb effectiveness offers good conceptual comparison against conventional evaporative coolers, but since the metric does not align with the physical heat transfer mechanisms active in the indirect evaporative heat exchanger, it does not provide a useful empirical correlation for system and building modeling. Alternative metrics use wet bulb temperature of the secondary air stream, or the dew point temperature of the primary air stream as the theoretical potential for an effectiveness ratio.

Here, performance of the indirect evaporative system is described by two different metrics. The indirect wet bulb effectiveness accounts for the wet bulb potential in the secondary air stream, this metric is given by:

$$Ind.WBE = \frac{T_{DB\ 1st\ inlet} - T_{DB\ 1st\ out}}{Ind.WBD} = \frac{T_{DB\ 1st\ inlet} - T_{DB\ 1st\ out}}{T_{DB\ 1st\ inlet} - T_{WB\ 2nd\ inlet}} \tag{10}$$

Performance is also described by the indirect wet bulb approach, which is given by:

$$Ind.WBA = T_{WB\ 2nd\ inlet} - T_{DB\ 1st\ out} \tag{11}$$

WEATHER NORMALIZATION FOR PRE-POST ASSESSMENT OF WHOLE BUILDING ENERGY CONSUMPTION

Weather normalization is important to eliminate the influence from year to year weather differences clouding the distinction between the energy consumption of the pre and post retrofit setup for a building. This was accomplished using methods described by ASHRAE Guideline 14: *Measurement of Energy, Demand and Water Savings*. To weather normalize the data, the research team used historical kW smart meter data from the San Ramon grocery store host site as well as historical outdoor air temperature data from PG&E's Applied Technology Services Laboratory in San Ramon to build a linear regression model of the pre retrofit state of the grocery store building.

$$kW_i = \beta_i \cdot T_{DB,OSA} + \varepsilon_i \qquad 12$$

In analyzing the relationship between the kW and outside temperature the research team decided to divide the regression into three parts based on hour of the day characterizing an Open, Closed, and Transitional operation state for the building. A separate model was prepared for each operating state.

Using data from 2012 and 2013 to train the three part regression, it was then applied to the 2014 outside air temperature data to develop a projected baseline scenario to for comparison with the actual energy consumption measured in 2014.

To predict peak energy draw a temperature binned statistical analysis of the open hour training regression data was used to recreate expected variance. This variance was then added back into to the linear 2014 predicted energy usage data to also predict expected peak conditions for the pre retrofit predicted data set. A simple linear regression model would have only predicted expected averages of energy use in a linear fashion from given outside air temperature values, drastically underestimating peak energy use.

RESULTS

SUPPLY FAN AIRFLOW AND POWER MEASUREMENTS

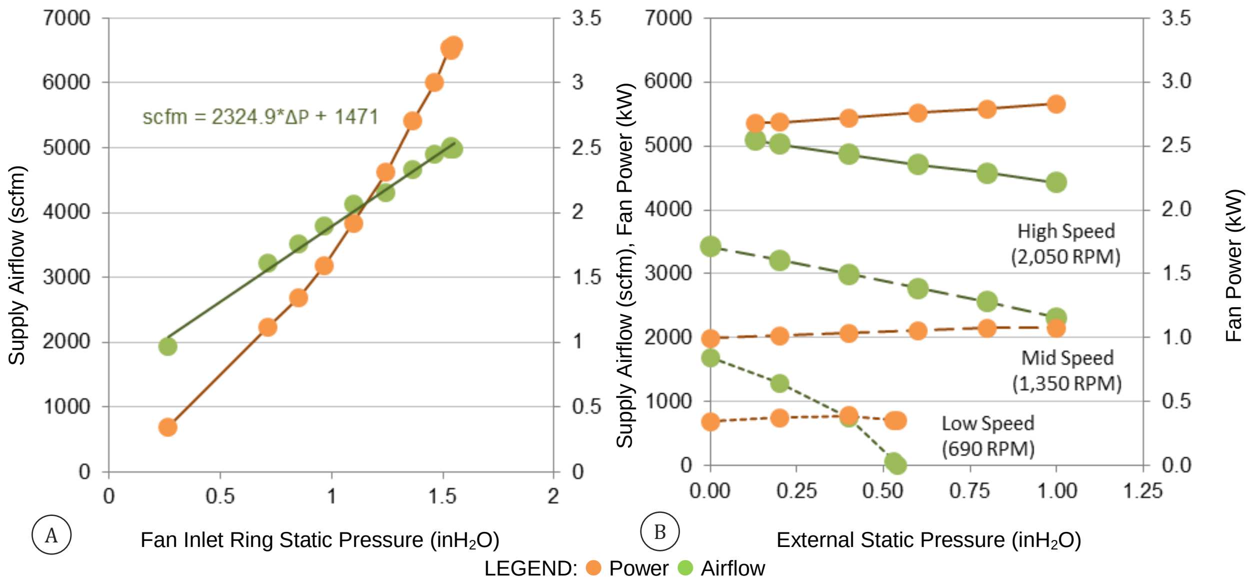


FIGURE 8: SUPPLY AIRFLOW AND SYSTEM POWER (A) FOR FIELD TEST AS A FUNCTION OF FAN INLET RING STATIC PRESSURE AND (B) FOR LABORATORY TEST AS A FUNCTION OF EXTERNAL STATIC PRESSURE

Figure 8A plots the supply airflow rate and fan power for the primary blower, as measured in the field for a range of fan speeds. The measured airflow rate and fan power draw are plotted against the static pressure measured across the inlet ring of the primary fan. Differential static pressure at this location has a reliable relationship to airflow regardless of the system resistance or fan speed. Airflow measurements were conducted using a tracer gas system as described in section Data Analysis and Equation 1. The range of airflow rates was captured by adjusting the fan speed setting to 10 different points. The resulting map illustrated in Figure 8A was used to determine airflow rate at all instances throughought the period of observation.

Figure 8B plots supply airflow rate and fan power measured in a previous laboratory test of the same system (Woolley 2014). The results are plotted against external static pressure, and present operation across a range of different fan speeds and external resistances. The corresponding fan inlet ring pressure was not measured.

The unit evaluated in the field used a somewhat more powerful fan than the system laboratory tested, and the consequences of the change to a new fan are apparent by comparison of the two maps in Figure 8. While the system supplied 5,000 cfm at 2.7 kW in the laboratory, 3.3 kW was required in the field. Moreover, it appears that fan power reported by laboratory tests for system performance at each climate condition was roughly 2 kW. Some of the difference between the laboratory test and field evaluation can be attributed to the more powerful fan, and part of the difference is associated with real world airflow resistance. This increase highlights the importance of appropriately handling fan power consumption in simulations.

Coefficient of Performance

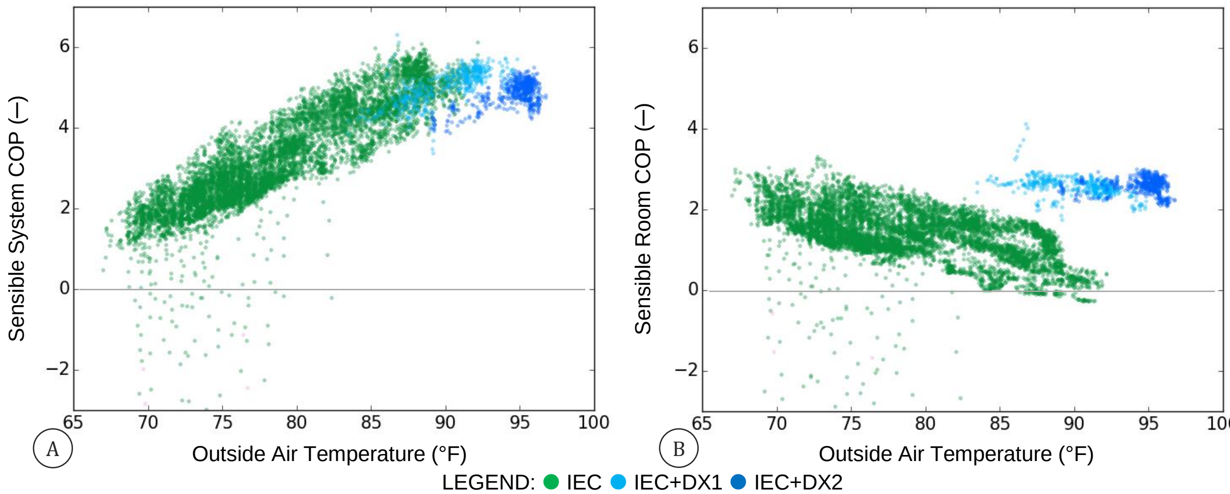


Figure 9. Coefficient of Performance as a function of outside air temperature for (A) Sensible System Cooling, and (B) Sensible Room Cooling

Figure 9A plots the sensible system coefficient of performance in each cooling mode as a function of outside air temperature, and Figure 9B plots the sensible room coefficient of performance for the same instances. These metrics are described by equations 3, 5, 7 and 8. The sensible system coefficient of performance considers the net sensible cooling capacity generated by the machine, while the sensible room coefficient of performance only considers the thermal energy balance for the conditioned environment.

Most importantly, sensible system COP in every mode increases as outside air temperature increases. This behavior is a major advantage compared to conventional air conditioning equipment, for which efficiency typically declines as outside temperature and cooling load increase. It is also noteworthy that at 90°F, when compressor operation is added on top of indirect evaporative cooling, the sensible system COP only declines by about 15% while cooling capacity increases by 133%.

Despite the advantages, practitioners should be reminded that the sensible room COP in indirect evaporative cooling mode is lower at high outside air temperatures because cooling capacity decreases as outside temperature rises. The indirect evaporative cooler may offer its most substantial value at this point, but additional capacity may be required from the on board vapour compression component, or from other systems in the building. Cooling from the on-board vapor compression comes at a much lower energy cost than compressor cooling from conventional equipment. For example, at 90°F, where the indirect evaporative cooler delivers room neutral air, the addition of first stage cooling adds 75 but/hr while electric power only increases by 3.5 kW. This addition effectively delivers room cooling with a sensible room EER of 21.4. At the same condition, a modern high efficiency rooftop unit would operate with sensible room EER of 8.5, and the antiquated systems on the grocery studied here likely operate with a sensible room EER of 6 or less. Therefore, when additional room cooling capacity is needed, the on-board vapor compression capacity should be utilized to every extent possible. Building system design and control systems should be developed in such a way to accommodate this opportunity, and to use capacity available from the hybrid DOAS for needed room cooling to whatever extent possible.

The general trends observed here are consistent with records from a previous laboratory study (Woolley 2014), however there are some significant differences that should be pointed out. First, the difference in efficiency between modes is smaller in this study than what was measured in laboratory testing. Second, the sensible system COP at all conditions is 5–15% lower than what was recorded in laboratory tests. The reason for these differences are explored in later sections through analysis of supply air temperature, cooling capacity, and electric power draw in each mode. These observations underscore the performance differences that can arise because of differences in configuration for each application.

COOLING CAPACITY

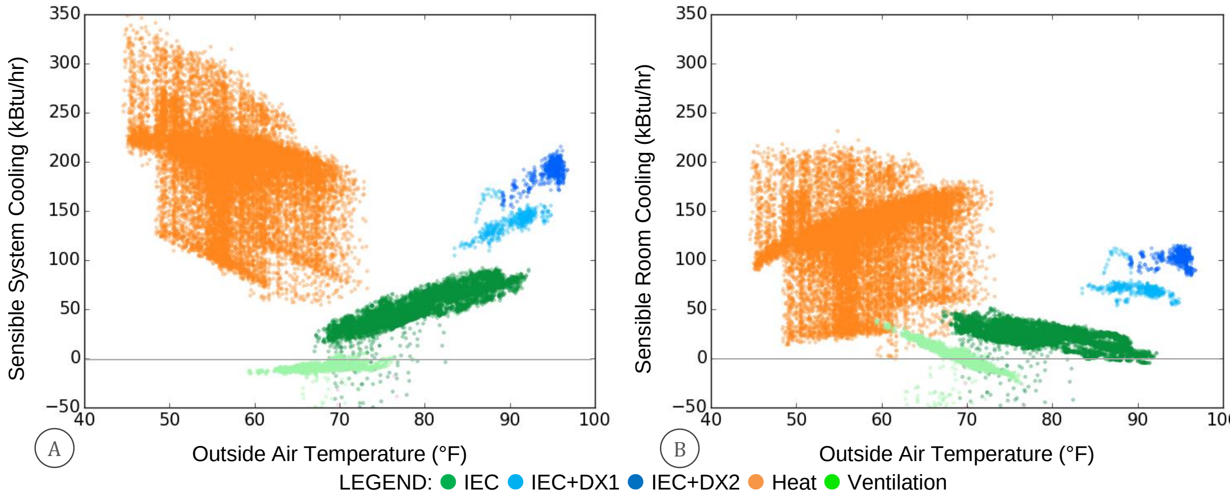


FIGURE 10. COOLING CAPACITY AS A FUNCTION OF OUTSIDE AIR TEMPERATURE FOR (A) SENSIBLE SYSTEM COOLING, AND (B) SENSIBLE ROOM COOLING

Figure 10 plots the sensible cooling capacity in each mode of operation as a function of outside air temperature. Each point in the plot represents a one minute interval of operation. Figure 10 A plots the system cooling capacity, described by equation 3, and Figure 10 B plots the sensible room cooling capacity, described by equation 5. Cooling is plotted as positive values and heating is plotted as negative values.

The performance recorded here corresponds well with measurements from a previous laboratory evaluation of the same machine (Woolley 2014). At 90°F outside air temperature and in mode "IEC+DX2", laboratory measurements indicated the system should generate 182 kBtu/hr sensible system cooling capacity. The field measurements presented in Figure 10 record almost exactly the same value. For operation in mode "IEC" the laboratory tests measured 73.4 kBtu/h sensible system cooling capacity and field observations indicated roughly 75 kBtu/hr.

Notably, the system sensible cooling capacity increases as outside air temperature increases – this is true in all cooling modes. The trend is significant advantage over conventional vapour compression equipment for which sensible cooling capacity tends to decrease when outside temperature increases. Even while the system cooling capacity increases, the room cooling effect decreases as outside air temperature rises because the supply air temperature also rises. At around 90°F, indirect evaporative cooling delivers room-neutral air, so the sensible room cooling capacity is approximately zero. At higher outside air temperature the indirect evaporative mode would result in a sensible gain to the room. Even while the room cooling capacity is zero, the mode still has great benefit at this point because it offsets roughly 75 kBtu/hr of sensible load in the ventilation air and substantially reduces the amount of vapour compression cooling that is needed. While it may seem somewhat incongruous, the instances where indirect evaporative cooling has the largest efficiency benefit correspond to periods when the mode does not have enough cooling capacity to cover all room cooling needs on its own.

Compressor operation is restricted to periods above about 85°F, and the second compressor only operates above about 90°F. When the first and second stage compressors do operate, they add roughly 65 kBtu/hr and 100 kBtu/hr sensible cooling capacity, respectively, on top of what is generated by indirect evaporative cooling. As a result, at 95°F in mode "IEC+DX2", roughly half of the system sensible cooling capacity is generated by indirect evaporative cooling and half is generated by vapour compression cooling. In this way, the system is able to increase the room cooling capacity as room cooling loads increase.

In heating mode, capacity ranges from about 75 kBtu/hr to 350 kBtu/hr, depending on heating stage and outside air temperature. Heating mode almost never operates in a steady state. Instead, gas combustion cycles on and off in pulses of 1-5 minutes with breaks. As a result, the supply air temperature during heating periods varies dramatically, which results in large minute-to-minute variation in the heating capacity. It can be difficult to model the energy performance of these patterns since almost all of the heating capacity generated occurs in transient periods.

Runtime in Each Mode of Operation

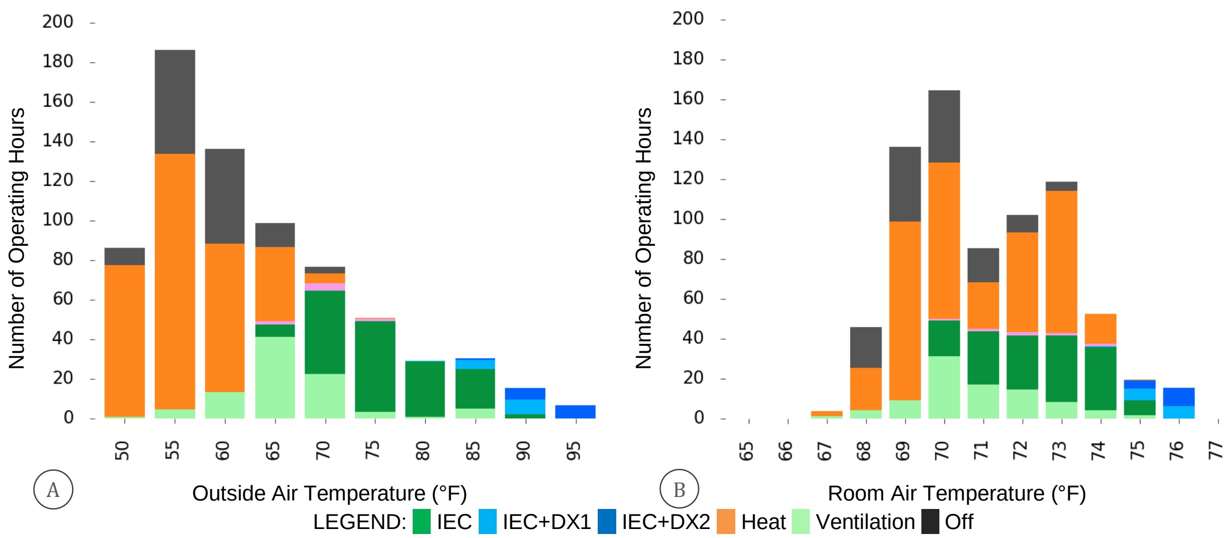


Figure 11. Cumulative runtime in each mode of operation as a function of outside air temperature

Figure 11 plots distribution of time spent in each operating mode over the course of the study period. Figure 11 A presents the amount of time spent in each mode across a range of outside temperatures and Figure 11 B plots the same data across the corresponding range of room temperatures observed.

First of all, for the period of observation presented, heating accounts for the largest number of operating hours. For this application, heating is requested for some periods with outside temperature as warm as 72°F. Many buildings would not require heating at such a warm temperature, but this can occur for a grocery store because there is so much refrigeration in the store. The trend is also likely influenced by the fact that this store operates with excess ventilation. Heating is not always needed as such warm temperatures, in fact, some periods as low as 60°F operate in ventilation only, and cooling operates as low as 65°F. The question of whether heating or cooling is required during these mild temperature periods depends on a variety of other variables including time of day and store operations. However, the research team consistently observed a very rapid changeover from heating to cooling each morning. It appears that the sequence of operations programmed may not have given enough deadband between heating and cooling. A substantial amount of energy is wasted by switching rapidly between heating and cooling. More appropriate control strategies would adopt a dynamic definition of comfort that allows a building to drift in temperature during these mild temperatures. Instead, there is really no outside temperature range in which the system not required to operate in either heating or cooling.

Even so, there are a significant number of ventilation only hours, when the system does not provide heating or cooling. These likely occur in between heating and cooling cycles, once the set point has been met. Interestingly, Figure 11 B shows that ventilation operates across a wide range of room air temperatures from 67–75°F and that room temperature ranges across an even wider range. Heating operates when the room temperature is as warm as 74°F and cooling operates when the room temperature is as low as 70°F. This strange occurrence arises in part from the fact that temperature is not evenly distributed across the entire store. Operating mode for the DOAS is cued by room temperature measured at the front of the sales floor and the room temperature reported in Figure 11 B is measured nearer the back of the sales floor. Therefore, the most valuable point to be grasped from Figure 11 B is that it can be difficult to maintain tight temperature distribution with the DOAS system – especially in a grocery.

Indirect evaporative cooling operates as low as 65°F, and covers all cooling requirements to about 85°F outside air temperature. Compressor operation is only needed when outside air temperature is above this point, and is sequestered tightly in those periods when room temperature is above 75°F. Lastly, we observe that compressor operation is only required for less than 15% of the total cooling hours observed. Since indirect evaporative cooling accounts for 50-75% of the capacity generated in modes with compressor operation, vapour compression cooling only accounts for about 20% of the total cooling generated.

Supply Air Temperature

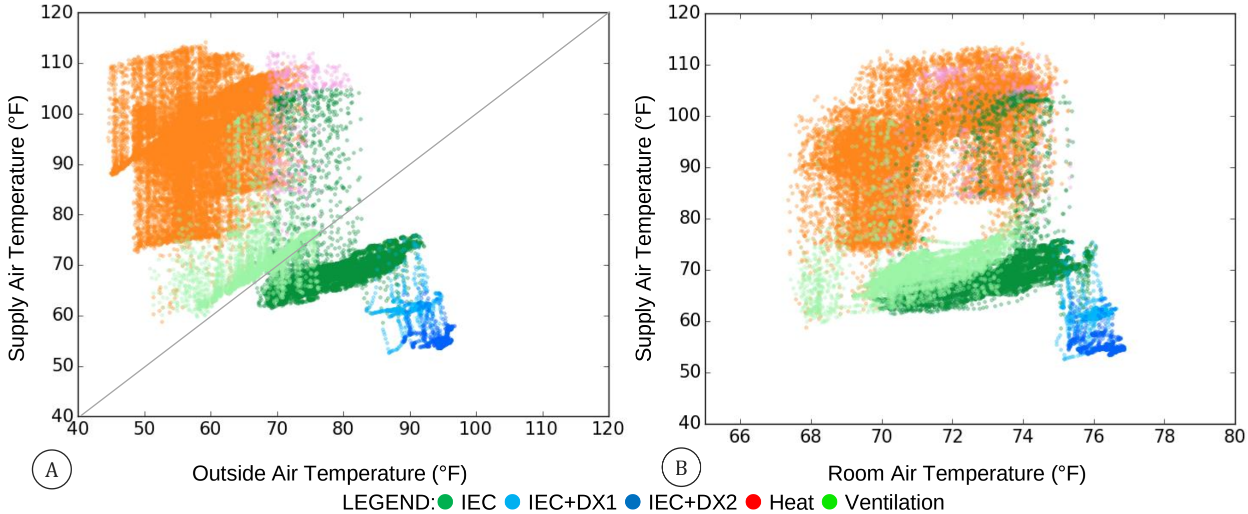


Figure 12. Supply air temperature in each mode of operation as a function of (A) outside air temperature (B) room air temperature

Figure 12A plots supply air temperature for each mode of operation as a function of outside air temperature. Then, Figure 12B plots the same data as a function of room air temperature. These charts include all instances observed, whether or not the machine had reached steady state operating conditions. As a result, the plots include large data variation, but equally, they offer some insight into the intermodal dynamics for the system.

The temperatures observed at steady state in all cooling modes are 2–3°F cooler than what was observed for the same outside air temperatures in recent laboratory tests (Woolley 2014). However, despite delivering supply air that is consistently cooler than the laboratory results, the cooling capacity observed is roughly equal, and the resulting efficiency is noticeably lower than laboratory results. Although the two tests evaluate the same machine there are significant differences between the configuration that was laboratory tested, and the configuration that is evaluated in the field. Some of the differences that could account for the seemingly incongruous results include:

- The outside air humidity was often somewhat lower than the conditions used for laboratory testing
- The laboratory test used room air as scavenger airflow, and the field test used outside air
- The field test used a larger scavenger airflow rate than the laboratory tests
- The field test utilized a more powerful primary fan capable of operating at higher static pressure

The differences are significant – efficiency for the field test is consistently15-20% lower on account of the way that the system was configured. It is currently very difficult for engineers, customers, and efficiency practitioners to estimate savings that will be achieved in a real world applications because they are not able to easily translate published laboratory performance numbers into models that predict application-specific performance. The differences observed here underscore the need for simple and flexible modeling tools that bridge this gap.

Figure 12B also illustrates that indirect evaporative cooling operates for periods when room temperature is above 70°F, and that compressor modes only operate once room temperature reaches 75°F. Controls for the system use a series of simple room temperature thresholds to cue the operation of each mode. As a result, the room temperature inevitably drifts before higher capacity modes are activated. This may have some efficiency advantage, however stacking several alternative modes in this way can result in a wider operating range for the room temperature than might be preferred. The remaining conventional rooftop units should be controlled to operate as the last and least efficient option, but this might require that the room temperature drift more than 6°F. If they are controlled in a way that keeps a tight set point, the predicted efficiency potential for the climate appropriate DOAS system may not be achieved. This all highlights that it can be difficult to balance efficiency and precision control with complex integrated systems. Future work ought to advance design strategies and controls innovations that could satisfy comfort and indoor environmental quality with the least amount of energy use.

CUMULATIVE COOLING IN EACH MODE OF OPERATION

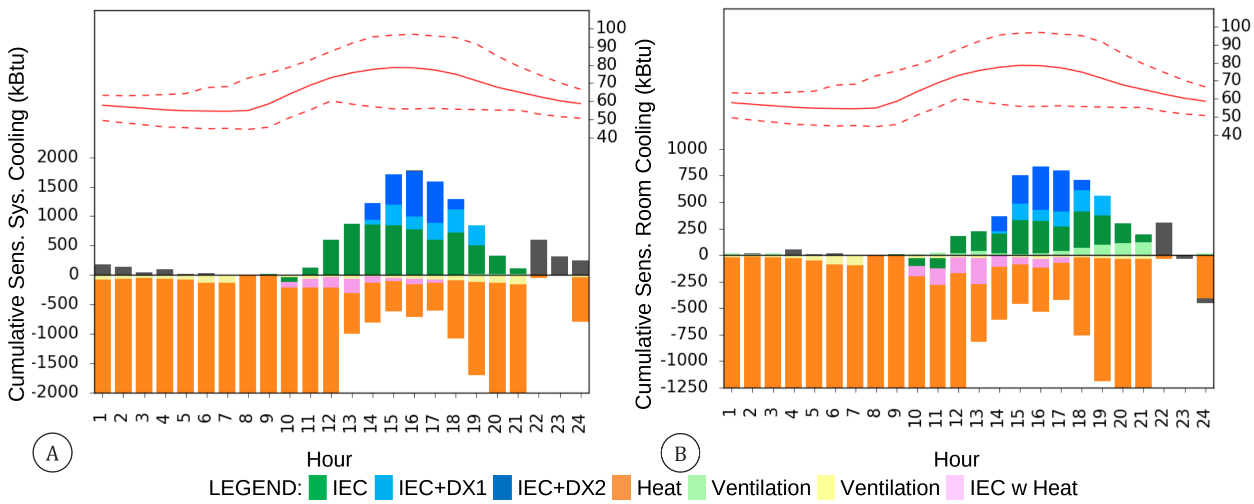


FIGURE 13. CUMULATIVE SENSIBLE COOLING CAPACITY IN EACH MODE OF OPERATION AS A FUNCTION OF HOUR OF THE DAY (A) SENSIBLE SYSTEM COOLING, AND (B) SENSIBLE ROOM COOLING

Figure 13 charts the cumulative amount of cooling and heating in each mode of operation at each hour of the day over the course of the study period. This plot illustrates how the cooling load is generally distributed over the course of the day, and the extent to which each mode is responsible for cooling. Figure 13A shows the cumulative system cooling capacity, and Figure 13B shows the room cooling capacity. The orange bars plot heat as a negative value, and the scale is cut off at -2,000 kBtu to focus on the cooling load distribution. The amount of heating energy delivered during overnight periods far outweighs the amount of cooling energy.

There are several significant observations that arise from this analysis. First, the largest amount of conditioning energy is associated with heat, and heating can be required in almost any hour of the day, depending mostly on the outside air temperature. The largest amount of heating energy is consumed when the store is closed. Ventilation is not required overnight, so quite a bit of energy is wasted by using the DOAS equipment to heat the space. If heating is really necessary at these times, an enormous amount of natural gas energy could be saved by adjusting the sequence of operations and set point temperatures during vacant hours. In October, the energy associated with heating unnecessary ventilation air during non-occupied hours accounted for 18,000 kBtu, which equates to roughly 226 therms natural gas consumption.

Next, the setpoints between heating and cooling appear to be too close. The unit regularly switches directly from heat to indirect evaporative cooling, which indicates that the sequence of operations is attempting to control to a fixed temperature. As a result, there are many periods where indirect evaporative cooling mode actually delivers a supply air temperature well above the incoming outside air temperature. The pink bars show the periods when the unit operates in indirect evaporative cooling mode, but the supply air temperature is still above 110°F. As illustrated in Figure 12 there are other instances where indirect evaporative cooling yields a temperature that is warmer than the incoming outside air. These observations occur in transition periods, but as illustrated in Figure 13, the total amount of energy delivered is significant.

Importantly, indirect evaporative carries more than half of the cumulative sensible cooling loads without compressor operation over the course of the period observed. Considering the fact that between 40-60% of the sensible cooling capacity is generated by indirect evaporative in modes where vapor compression is active, indirect evaporative cooling is responsible for more than 80% of the cumulative sensible cooling generated.

Lastly, given the fact that so much energy use is associated with heating, practitioners should consider that the design practices recommended in section *Design and Application* would have substantial benefit for heat recovery during occupied hours, where the DOAS could recovery approximately 40% of the sensible heat in the ventilation exhaust. The energy savings associated with heat recovery in this application would outweigh the benefits added to cooling efficiency by using stale return air as the secondary air stream for indirect evaporative cooling.

PSYCHROMETRIC PERFORMANCE

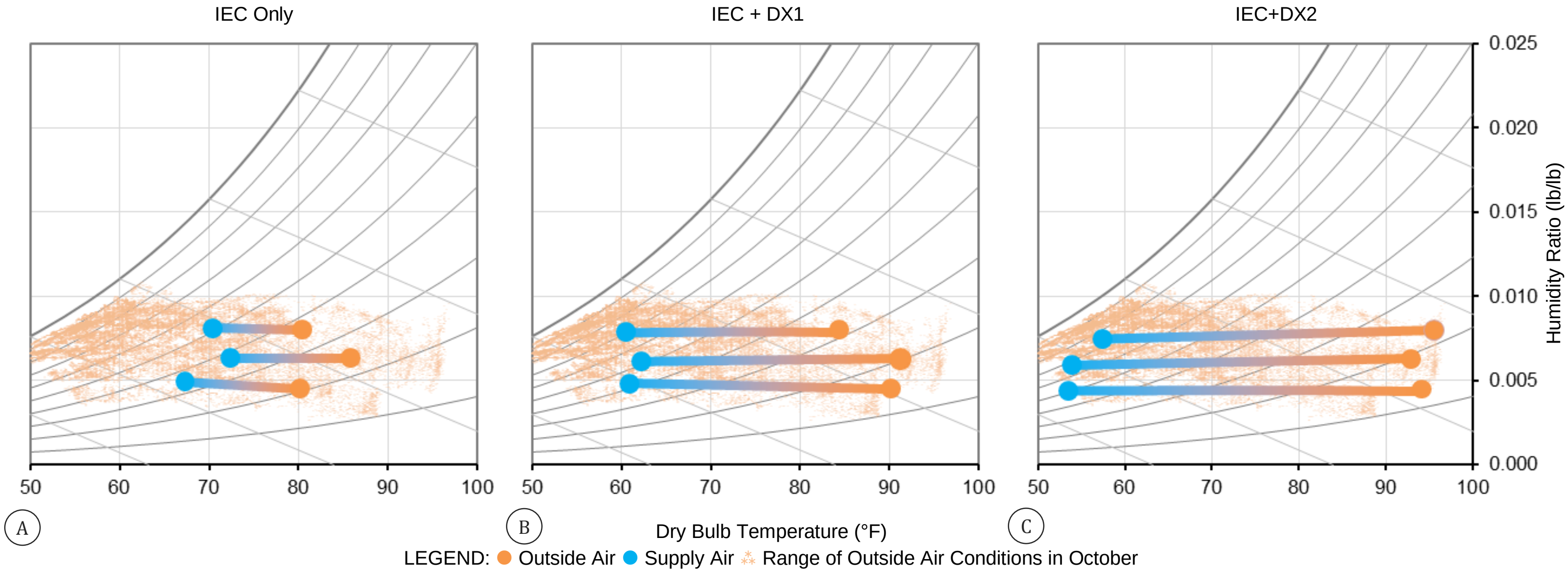


FIGURE 14. PSYCHROMETRIC CHART – PERFORMANCE IN EACH COOLING MODE FOR SEVERAL OUTSIDE AIR CONDITIONS

Figure 14 plots the outside air conditions and supply air conditions for operation in each cooling mode and at various outdoor humidity ratios. Figure 14 A illustrates operation in mode: "IEC Only", while Figure 14 B and C illustrates operation in modes "IEC+DX1" and "IEC+DX2" respectively. These plots clearly indicate the result of operation in each mode. When loads are low indirect evaporative cooling can maintain comfort with supply air temperature at roughly 65–70°F. Operation in mode "IEC+DX1" supplies air closer to 60°F, and mode "IEC+DX2" supplies air closer to 55°F.

Most importantly, the amount of latent cooling in each mode is almost negligible on account of the fact that the outdoor humidity never exceeded 0.010 lb/lb, and because the supply air temperature was never so cold as to result in an appreciable amount of dehumidification. This characteristic is an excellent benefit for California climates because the thermal energy associated with dehumidification is generally of no practical use. In more humid climates, the hybrid DOAS studied here would generate some dehumidification, but the unit is not intended to provide active dehumidification by design, so if active dehumidification is really necessary there are other solutions that are more appropriate for the job.

Wet Bulb Effectiveness

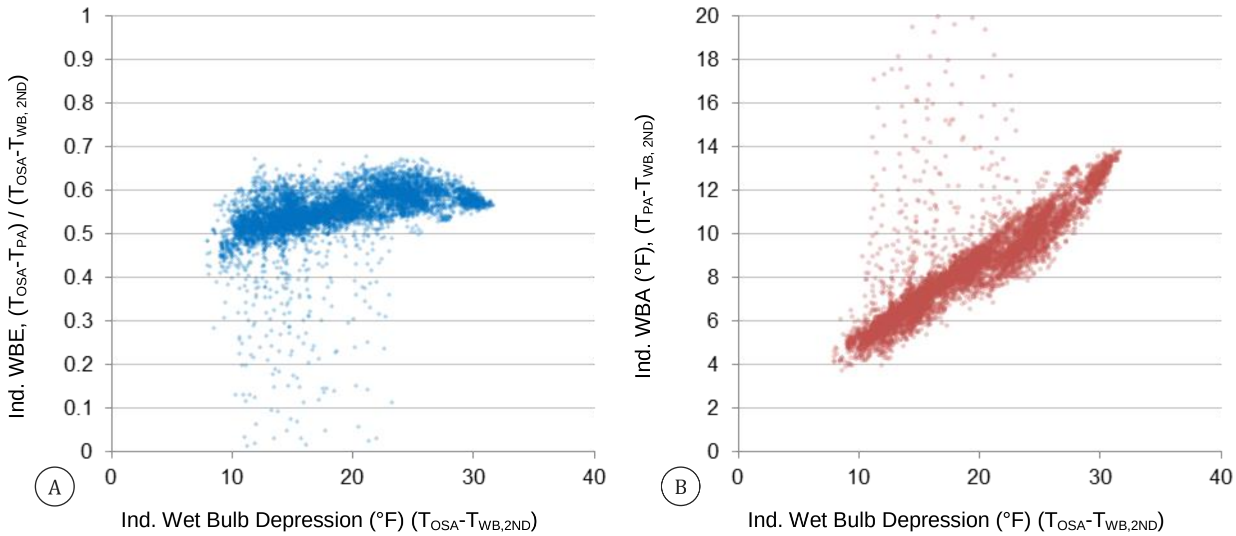


Figure 15. Indirect evaporative cooling performance as a function of wet bulb depression (A) Wet bulb effectiveness and (B) Wet bulb approach for all instances in indirect evaporative mode

Figure 15 describes the evaporative performance of the indirect evaporative cooler as a function of wet bulb depression. The indirect wet bulb effectiveness presented in Figure 15 A is described by Equation 10. The indirect wet bulb approach presented in in Figure 15 B is described by Equation 11. Previous laboratory evaluation of this system showed that evaporative performance trends most closely with the difference between the dry bulb temperature at the primary inlet, and the wet bulb temperature at the secondary inlet – since those values establish the driving potential for cooling. In the application studied here, outside air is used as the source for the secondary air stream, so the "indirect wet bulb effectiveness" is equivalent to "wet bulb effectiveness". However, in applications where return air can be used, practitioners should be careful to apply these results appropriately, according to Equation 10 and Equation 11.

The indirect evaporative cooler achieves wet bulb effectiveness between 0.4 and 0.7, depending on the operating conditions. There are some instances in Figure 15 with lower effectiveness, but these occur during transition periods, especially immediately after changeover from heating. Effectiveness decreases somewhat when wet bulb depression is lower. 75% of points measured achieve wet bulb effectiveness between 0.5 and 0.6. The results correspond very closely with the conclusions from laboratory testing. In fact laboratory tests indicate that wet bulb effectiveness remains steady near 0.6 for wet bulb depression above 20°F.

Indirect wet bulb approach also agrees exceptionally well with laboratory observations. This metric forms a very linear relationship with wet bulb depression, even when the wet bulb depression is small. Therefore, the research team recommends the use of wet bulb approach for the purposes of modeling performance for this system.

Wet bulb effectiveness of 0.6 is somewhat lower than other modern indirect evaporative air conditioners, however, when return air is used as the secondary air stream for indirect evaporative cooling this machine can essentially increase performance to wet bulb effectiveness near 0.8. Competing indirect evaporative systems do not have this unique capability for return air heat recovery.

SYSTEM POWER DRAW

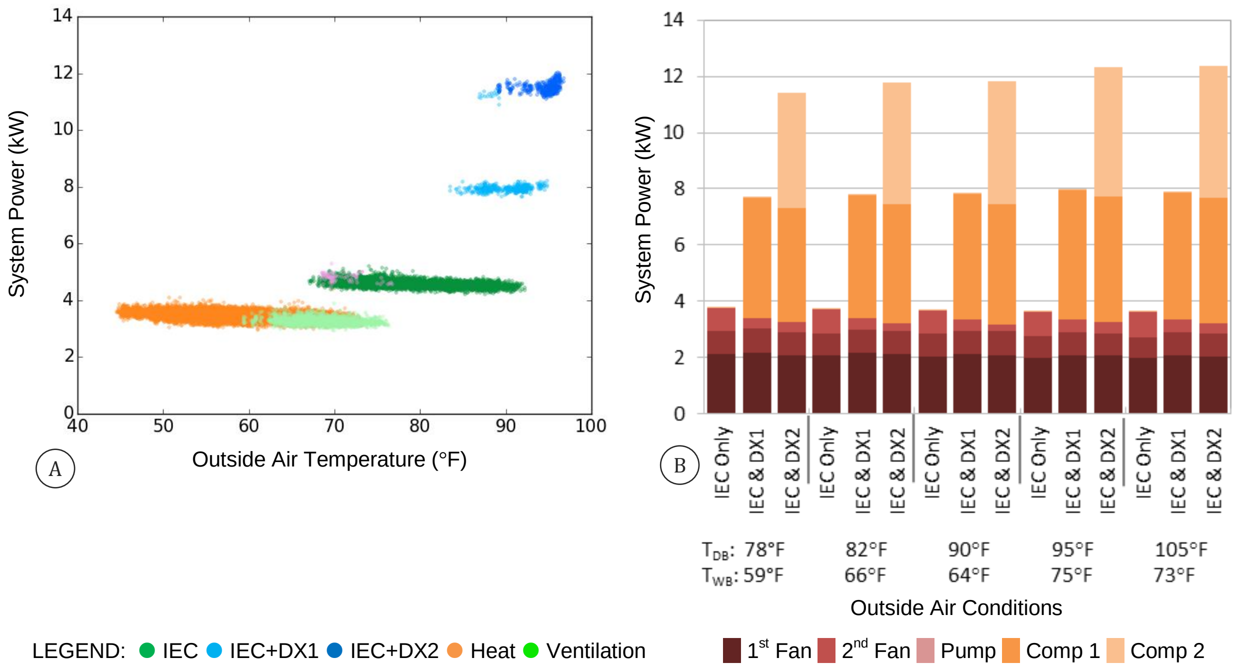


FIGURE 16. SYSTEM POWER DRAW (A) IN EACH MODE OF OPERATION AS A FUNCTION OF OUTSIDE AIR TEMPERATURE , AND (B) FOR VARIOUS MODES AND CONDITIONS MEASURED IN PREVIOUS LABORATORY TESTING

Figure 16 A plots the electric power draw by the DOAS system in each mode of operation across the range of outside air temperature observed. Figure 16 B charts the power draw of each sub-component in the system as measured in a previous laboratory study.

Notably, the power draw in indirect evaporative cooling mode is less than half the power draw in mode "IEC+DX1", and less than one-third the power draw in mode "IEC+DX2". Indirect evaporative cooling only requires electricity for the primary fan, the secondary fan, the water pump, and controls. The power consumption for indirect evaporative cooling remains consistent, regardless of the climate conditions to which it is subjected. Also, power draw for vapour compression cooling does not change much as outside temperature changes on account of the fact that the vapour compression condenser is located in the indirect evaporative exhaust air stream.

There are a few important differences between the power draw recorded for laboratory tests, and the power draw measured from in field application. Most importantly, power draw for the primary fan was about 1 kW larger for the in the field, and power for the secondary was roughly 0.25 kW larger. At the same time, whole system power draw in modes "IEC+DX1" and "IEC+DX2" is almost exactly the same as what was measured in the laboratory – this indicates that the compressors actually used somewhat less electricity than the laboratory tests.

This all corresponds to previously discussed differences between the laboratory configuration and the configuration tested here. Primary fan energy use was higher because the primary fan used in the field is more powerful than the one tested in the laboratory. Secondary fan power was larger because the secondary airflow rate was larger. Power draw for the compressors was reduced because the condenser airflow rate was larger, and because the conditions encountered in the field had a somewhat lower wet bulb temperature than the most comparable laboratory tests – the resulting condenser inlet temperature would tend to reduce compressor power draw.

WATER CONSUMPTION AND EFFICIENCY

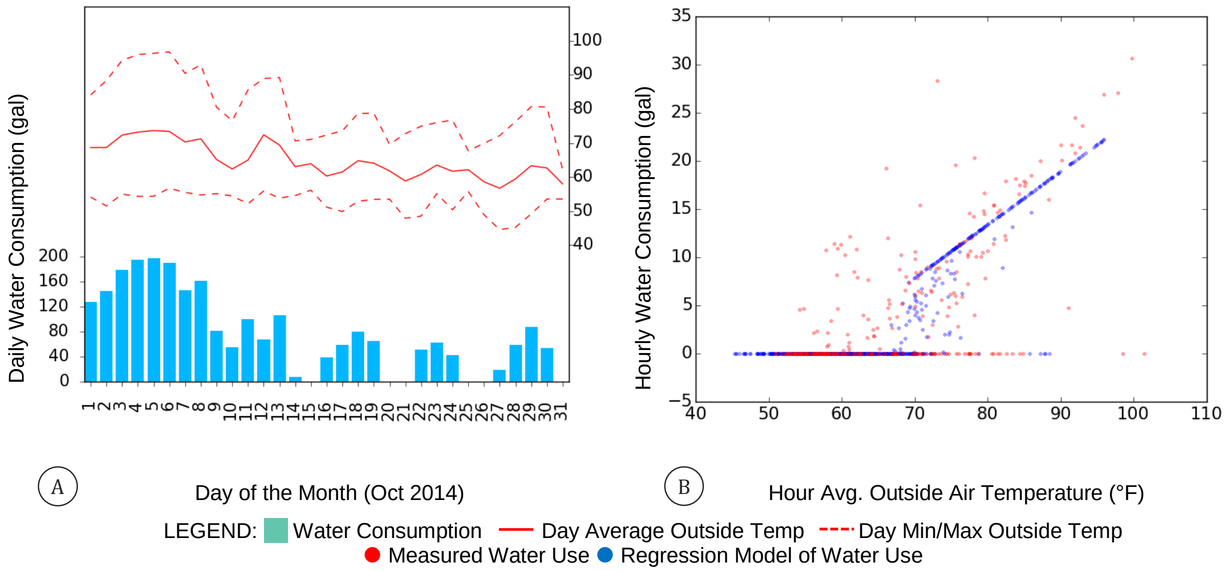


FIGURE 17. SYSTEM POWER DRAW IN EACH MODE OF OPERATION AS A FUNCTION OF OUTSIDE AIR TEMPERATURE

Figure 17 describes water consumption characteristics for the air handler. Figure 17 A charts the cumulative water consumption for each day along with the corresponding daily average, minimum, and maximum outside air temperatures. Warmer days result in larger water consumption in part because the instantaneous evaporation rate increases with outside air temperature, and in part because the number of cooling hours increases on hotter days. Some days in October were not warm enough to require cooling.

Figure 17 B plots measured hourly water consumption data as a function of the average outside air temperature in each hour. The figure also presents the predicted output from a regression model of water consumption that uses outside air temperature and mode of operation as independent predictors. Figure 17 B illustrates that there is a strong correlation between water consumption and outside air temperature, but that water use at any given outside condition can be vary significantly. Other factors that influence water consumption include water quality, outside air humidity, the difference between wet bulb temperature of the secondary air stream and dry bulb temperature for the primary air stream, as well as the circumstances of timing for sump fill and drain sequences.

At peak conditions, the hybrid air handler consumes 25–30 gallons of water per hour. Given that the sensible system coefficient of performance at this point is approximately 5.2, and that use of this system would reduce whole building electrical demand for cooling and ventilation by 9.75 kW, we estimate that the ratio of water use to electrical energy savings at peak is approximately 2.5- 3.1 gallon/kWh savings. At milder temperatures the magnitude of electrical savings is somewhat lower and the ratio of water use to energy savings rises to 6 gallons per ton hour.

The water consumption associated with this measure may be of some concern, especially while California is in the midst of a severe drought, and building owners face tighter water use regulations. However, on site water use is offset by upstream water savings associated with reduced electricity generation. Also, the amount of water used by this system could be easily offset by water efficiency measures for other key end uses.

For context, estimates of the water use intensity for electricity generation range by more than an order of magnitude, depending on the source mix for electricity generation. However, the most well founded research estimates a water use intensity of 1.41 gal/kWh for California's grid mix, on average including evaporative losses from reservoirs for hydroelectric generation (Pistochini 2011, Torcellini 2003, Larson 2007). This me ans, from a statewide water use perspective, the local water consumption for these retrofits would be partially offset by the water savings associated with reduced electrical generation.

WEATHER NORMALIZED WHOLE BUILDING ENERGY SAVINGS

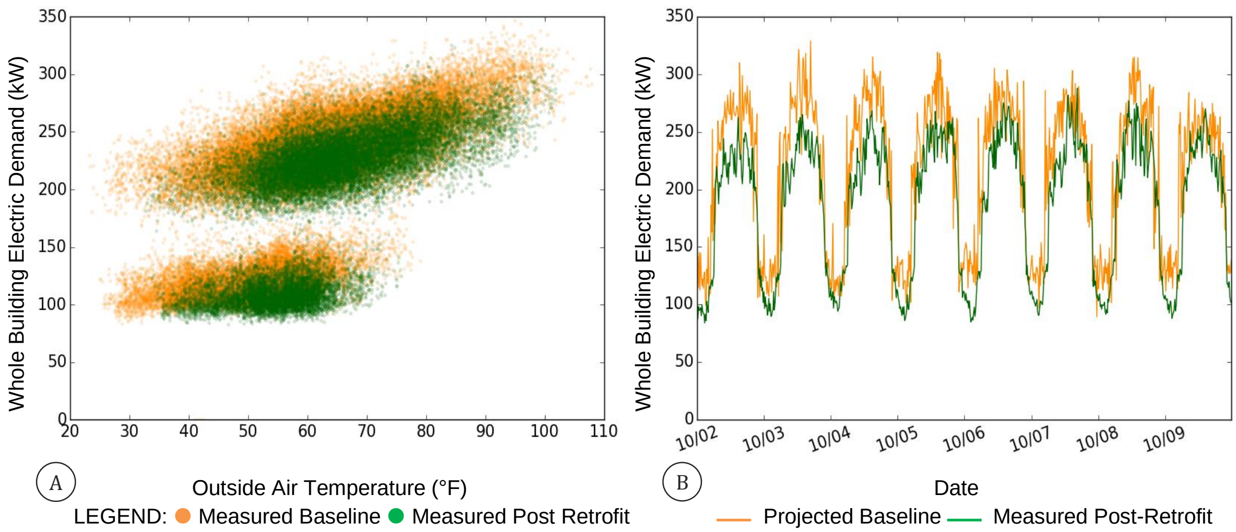


FIGURE 18. (A) PRE-POST COMPARISON OF WHOLE BUILDING ELECTRIC DEMAND AS A FUNCTION OF OUTSIDE AIR TEMPERATURE (B) TIME SERIES COMPARISON OF PROJECTED BASELINE AND MEASURED POST RETROFIT ELECTRIC DEMAND

Electric meter for the facility was analyzed to assess the whole building energy savings associated with the retrofit. Energy use from 2012 and 2013, before the retrofit, was compared against energy use 2014 to establish an estimate of the savings achieved. The comparison was normalized for differences in weather conditions according to methods described in ASHRAE Guideline 14 (ASHRAE 2002). Pre-retrofit whole building electricity consumption data was used to develop a single-break-point linear regression model as a function of outside temperature. Separate models were developed for open hours, closed hours, and transition periods. Then, the resulting map of baseline electricity consumption was utilized to estimate a projected baseline energy consumption profile that would have resulted for the weather conditions observed in the post retrofit measurement period.

Figure 18A plots the measured baseline data and measured post retrofit data as a function of outside temperature for open hours, for closed hours, and for transition hours that were not reliably in either the open or closed groups. Significant savings is readily apparent in each group. Overall, energy use during closed hours is not strongly dependent on outside air temperature, but energy use during operating hours is. For the baseline period, site electric demand usually ranged from 175 – 300 kW during open hours, and dropped back to 100-175 kW during closed hours. Electric demand during open hours in the post-retrofit period never exceeded 300 kW, and energy use during closed hours was typically 25 kW lower than the pre-retrofit period.

Figure 18A plots a time series comparison of the measured post retrofit electricity consumption and the projected baseline electricity consumption for the same operating conditions. This assessment clearly indicates that that weather normalized electricity savings is substantial. During closed hours savings is consistently around 25 kW, while savings during open hours averages at 50 kW but changes some with outside temperature.

The results are surprising because: the hybrid DOAS should not provide significant savings during closed hours, and because 50kW peak demand savings is roughly four times larger than the anticipated reduction.

However, the climate appropriate hybrid evaluated here was only one of three retrofits installed concurrently.

1. The store also installed a new energy management and control system to integrate control of lighting, HVAC, and refrigeration.
2. A lineup of open medium-temp refrigerated cases was replaced with a new case equipped with doors.

As a result, the whole building peak electrical demand was reduced by more than 20% (almost 25 kW), and the store reduced annual energy consumption by 124,373 kWh. Of this, we estimate that the DOAS system contributed 10-20 kW demand reduction at peak, 60,0000 kWh savings per year. The methodology for this calaculation is described in section: *Weather Normalization for Pre-Post Assessment of Whole Building Energy Consumption.*

DISCUSSION AND CONCLUSIONS

The observations presented in this report form a clear and reliable map of performance for the Munters EPX 5000 in real world operation. There are some significant differences between performance observed for this study, and what was reported from laboratory testing. The apparently incongruity can be attributed mostly to differences in system configuration and setup. The field evaluation also describes some of the dynamic characteristics associated with intermodal operations. These transient performance periods can form a significant portion of the overall operating time, so accounting for transient performance appropriately is important. These observations underscore the need for modeling and simulation tools that are capable of capturing application–specific performance characteristics.

Aside from mapping performance quantitatively, this study also explored the opportunities and challenges associated with real world application and operation of the technology. The lessons learned are presented throughout this report as recommendations and design guidance that should help practitioners apply the technology appropriately. The most compelling recommendations and conclusions are summarized in the following paragraphs.

Proper setup and commissioning for this technology is imperative. Initially, the whole building controls were not set up appropriately for the application studied here. This would have resulted in zero energy savings for the project. The technology should be thought of as the enabling element in a "whole building" efficiency measure, and not as a standalone solution. The technology requires proper integration with building controls, needs coordination with other systems and operating modes, and must be accompanied by a re-balancing and controls reprogramming for other rooftop units on a facility.

Annual energy savings will depend on whole building airflow interactions, and so setup of the dedicated outside air scheme must be engineered in a thoughtful way. For example, if the mode of operation is controlled on temperature in a refrigerated section of the store, the high efficiency cooling system may never operate. Also, part of the savings for this measure is achieved by shutting off the supply fans on other rooftop air conditioners when cooling is not needed in the associated zones.

Proper ongoing service is necessary; the most important element observed through this project is the need for regular filter changes. Do not underestimate that this is a critical need. If not serviced, the equipment will turn off on a fault when airflow restriction is too great. This could easily go unnoticed since the store would be ventilated by kitchen exhaust and conditioning requirements would be handled by conventional rooftop air conditioners. This exact scenario occurred during the course of this study. In a real world application, all savings benefits would be lost.

Whenever possible, system design should strive to use return air as the secondary air stream. This will improve efficiency for indirect evaporative cooling by 43%, and will allow for heat recovery ventilation when heating is required. This may not be accomplished straightforwardly when installed in a grocery or other application with large kitchen exhaust flows, but it should be done if possible.

The technology has major efficiency advantages when compared to conventional rooftop air conditioners. We estimate that whole building HVAC electrical demand could be reduced by 20% at peak, and the measured whole building annual energy use confirms that there are substantial annual savings associated with the measure. However, proper application is esstial. The research team recommends that any program and efforts to advance the technology should take careful steps to avoid possible points of risk. Some options include:

a. Require ongoing verification to ensure that the target energy savings is achieved
b. Incorporate prescriptive requirements to ensure proper whole system design and commissioning, and (importantly) incorporate some mechanism to ensure that this actually occurs.
c. Advance resources for professional training, develop more demonstrations, and develop publically available resources to document best practices (as well as common challenges)
d. Combine the DOAS retrofit with removal of some rooftop air conditioners. This would help to ensure that the high efficiency solution is not disabled in the future because it would be integral to maintaining comfort in the building

Ultimately, we strongly recommend broader application of the technology as a climate appropriate air conditioning solution. The solution offers substantial savings, and performs reliably in the field. We also recommend that additional efforts are needed to accurately establish the annual savings from this measure, separate from other efficiency measures that were installed in parallel during this project.

REFERENCES

A2B. Accurate Air Balancing LLC. Test and Balance Report. Grocery store in San Ramon, CA. April 24, 2014.

ARB. California Greenhouse Gas Emission Inventory. Air Resources Board. California Environmental Protection Agency. 2014. Online. http://www.arb.ca.gov/cc/inventory/inventory_current.htm

Architectural Energy Corporation, and Fisher Nickel Inc. Commercial Kitchen Ventilation Design Guides. Improving Commercial Kitchen Ventilation System Performance. Updated March 2004. Appendix to report: Makeup Air Effects on the Performance of Kitchen Exhaust Systems. CEC #500-98-031. December 2002. Online: http://www.fishnick.com/ventilation/designguides/

ASHRAE. 2009. ASHRAE Handbook – Fundamentals. IP Edition. Amercian Society of Heating and Air-Conditioning Engineers, Inc. Atlanta GA. 2009.

ASHRAE. Advanced Energy Design Guide for Grocery Stores – Achieving 50% Energy Savings - Toward a Zero Net Energy Building. American Society of Heating Refrigeration and Air Conditioning Engineers. 2015. Online. https://www.ashrae.org/standards-research--technology/advanced-energy-design-guides

California Energy Commission. 2013 Building Energy Efficiency Standards for Residential and Nonresidential Buildings. Title 24, Part 6 and Associated Administrative Regulations in Part 1. CEC-400-2012-004-CMF-REV2. Edmund G. Brown Jr., Governor. California Energy Commission. May 2012. Online. http://www.energy.ca.gov/2012publications/CEC-400-2012-004/CEC-400-2012-004-CMF-REV2.pdf

California Energy Commission. California Commercial End-Use Survey. CEC 400-2006-005. March 2006. Online. http://www.energy.ca.gov/ceus/

Energy Information Administration. Commercial Buildings Energy Consumption Survey. 2012. Online. http://www.eia.gov/consumption/commercial/

Fricke, B. Becker, B. Energy Use of Doored and Open Vertical Refrigerated Display Cases. Purdue University. International Refrigeration and Air Conditioning Conference. 2010. Online. http://docs.lib.purdue.edu/iracc

Google. Imagery ©2015 Goolge, Map data ©Google. Online. https://www.google.com/maps

Hale, E.T. Macumber, D.L. Long, N.L. Griffith, B.T. Benne, K.S. Pless, S.D. Torcellini, P.A. Technical Support Document: Development of the Advanced Energy Design Guide for Grocery Stores – 50% Energy Savings. National Renewable Energy Laboratory. Technical Report NREL/TP-550-42849. September 2008.

Harrington, C. Woolley, J. Davis, R. Laboratory Performance Results: Indirect Evaporative Air Conditioning & Condenser Pre Cooling as Climate Appropriate Retrofits for Packaged Rooftop Units. Southern California Edison. Customer Programs and Services. 2015. Online. http://www.etcc-ca.com/ (submission pending publication)

Hirsch, A. Hale, E. Leach, M. Strategies to Save 50% Site Energy in Grocery and General Merchandise Stores. ACEEE Summer Study, 2012.

NOAA. Quality Controlled Local Climatological Data. National Climatic Data Center. National Oceanic and Atmospheric Administration. 2015. Online. http://www.ncdc.noaa.gov/data-access/land-based-station-data/land-based-datasets/quality-controlled-local-climatological-data-qclcd.

Wang, W. Katipamula, S. Huang, Y. Brambley MR. Energy Savings and Economics of Advanced Control Strategies for Packaged Air Conditioning Units with Gas Heat. U.S. Department of Energy. DE-AC05-76RL01830. Pacific Northwest National Laboratory. PNNL-20995. December 2011. http://www.pnnl.gov/main/publications/external/technical_reports/PNNL-20955.pdf

Woolley, J. Davis, R. Western Cooling Challenge Laboratory Performance Results: Munters EPX 5000 Hybrid DOAS. ET12PGE3101. Pacific Gas & Electric Company. 2013. Online. http://www.etcc-ca.com/

Woolley, J. Modera, M. Western Cooling Challenge Laboratory Results: Trane Voyager DC Hybrid Rooftop Unit. Southern California Edison, Design and Engineering Services. HT.12.SCE.017. 2012. Online. http://www.etcc-ca.com/