Energy-Efficient Supermarket Heating, Ventilation, and Air Conditioning in Humid Climates in the United States

J. Clark
National Renewable Energy Laboratory

NREL is a national laboratory of the U.S. Department of Energy
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Technical Report
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List of Acronyms

ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
CIBSE	Chartered Institution of Building Services Engineers
CBP	DOE Commercial Buildings Partnership
cfm	cubic feet per minute—unit often used to quantify volumetric airflow rates
COP	coefficient of performance = cooling energy delivered/power input
DBT	dry-bulb temperature
DCKV	demand-controlled kitchen ventilation—strategy for ventilating kitchens that modulates exhaust rates based on equipment in use
DOE	U.S. Department of Energy
DOAS	dedicated outdoor air system—an HVAC strategy that conditions outdoor air in a separate system from recirculating air
DPT	dew point temperature—temperature at which water in air condenses for given water content in air
DX	direct expansion—means of cooling air through refrigeration cycle
EA	exhaust air
EEM	energy efficiency measure
EER	energy efficiency ratio = cooling output of a device/electrical power input (Btu/Wh)
HEH	high-efficiency hood—a kitchen hood that captures exhaust air more efficiently
HVAC	heating, ventilation and air conditioning
LD	liquid desiccant—hygroscopic material that dehumidifies air with which it is in contact
LT	low temperature—refers to cooling devices with suction DPT < 10ºF
MAS	mixed air system
MAU	make-up air unit—device used to provide air to counteract exhaust
MBL	Modelica Buildings Library—library of building-related Modelica models created by Lawrence Berkeley National Laboratory
MSL	Modelica Standard Library—standard library included with all software packages that implement the Modelica language
MT	medium temperature—refers to cooling devices with suction DPT > 10ºF
MX	abbreviation used to designate mixed air condition on psychrometric charts
NREL	National Renewable Energy Laboratory
OA	outdoor air
RA	return air
RH	relative humidity—ratio of moisture content to saturated moisture content
RTU	rooftop unit
SD	solid desiccant—hygroscopic porous material that absorbs moisture from airstream
SHR	sensible heat ratio—ratio of sensible cooling provided by a cooling device or needed in a space to total cooling provided or needed
SP	set point—desired conditions in a space on which control strategies are based

WADW	wrap-around desiccant wheel- a cooling device that operates by passing air through one side of a desiccant wheel, then a cooling coil, then the other side of the desiccant wheel, also called the Cromer Cycle
WAHP	wrap-around heat pipe—a cooling device that operates by passing air through one side of a heat pipe heat exchanger, then a cooling coil, then the other side of the heat pipe heat exchanger

Executive Summary

Supermarkets are energy-intensive buildings that consume the greatest amount of electricity per square foot of building of any building type in the United States and represent 5% of total U.S. commercial building primary energy use (EIA 2005). Refrigeration and heating, ventilation, and air-conditioning (HVAC) systems are responsible for a large proportion of supermarkets' total energy use. These two systems sometimes work together and sometimes compete, but the performance of one system always affects the performance of the other.

A wide variety of solutions are currently available that meet the HVAC needs of supermarkets, but the effects of these solutions on energy use and costs are not fully understood. For example, there are a number of energy-saving options for dehumidification in supermarkets. There may also be new HVAC approaches that can offset energy use resulting from larger food preparation areas and the proliferation of superstores that combine general merchandise and food sales.

To better understand these challenges and opportunities, the Commercial Buildings team at the National Renewable Energy Laboratory investigated several of the most promising strategies for providing energy-efficient HVAC for supermarkets and quantified the resulting energy use and costs using detailed simulations. This research effort was conducted on behalf of the U.S. Department of Energy (DOE) Commercial Building Partnerships (CBP) (Baechler et al. 2012; Parrish et al. 2013; Antonopoulos et al. 2014; Hirsch et al. 2014). The goal of CBP was to reduce energy use in the commercial building sector by creating, testing, and validating design concepts on the pathway to net zero energy commercial buildings. Several CBP partners owned or operated buildings containing supermarkets and were interested in optimizing the energy efficiency of supermarket HVAC systems in hot-humid climates. These partners included Walmart, Target, Whole Foods Market, SUPERVALU, and the Defense Commissary Agency.

Simulations were performed for six climate zones in the United States with large humidity loads, where advanced HVAC strategies are likely to show the greatest benefit. Accordingly, the research focused on advanced approaches for energy-efficient dehumidification. This report targets the supermarket design team trying to lower operating costs through efficient HVAC design. It specifically targets design teams that use energy modeling to understand the energy impacts of different design decisions and the interactions between the HVAC and refrigeration systems. Building owners will also be interested in the high-level results comparing different approaches. This report can be thought of as an extension to the energy modeling done for the DOE-funded *ASHRAE 50% Advanced Energy Design Guide for Grocery Stores* (www.ashrae.org/freeaedg), which provides guidance to achieve 50% whole building energy savings through proven efficiency measures across all building systems.

Methods

The results in this report were generated using the most advanced publicly available building simulation tools. These include co-simulation of DOE's EnergyPlus software with an external program, Dymola, which allowed NREL to model systems not yet included in EnergyPlus. See Appendix C to learn about a procedure for conducting these simulations, along with sample input files that may be used and adapted to conduct such simulations.

With these expanded capabilities, NREL simulated 23 different HVAC solutions for each of the six climates, including demand-controlled kitchen ventilation (DCKV); improved kitchen hoods and outdoor air (OA) delivery; dedicated outdoor air systems (DOAS); and enhanced dehumidification and air-conditioning equipment.

Objectives

For this research, NREL pursued five objectives:

- Demonstrate the use of recently enhanced building-integrated HVAC modeling capabilities and explain their benefit for modeling building energy use.
- Evaluate and assess the energy impacts of using advanced exhaust equipment in supermarket kitchens.
- Quantify and compare the energy impacts of various methods for delivering and conditioning OA.
- Determine the advantages and disadvantages of using novel HVAC systems to condition supermarkets, with a special emphasis on dehumidification.
- Understand and quantify the interactions between the various energy efficiency measures (EEMs), such as the combined effects of reducing humidity in the refrigerated aisles, adding a DOAS unit, and using high-efficiency hoods (HEHs).

Scope

NREL investigated 23 HVAC solutions, which are combinations of three EEMs: reduced kitchen exhaust, improved delivery of OA, and advanced dehumidification equipment. The three EEMs are explained in detail in Chapter 3. The EEMs and candidate solutions included:

- Reduced ventilation requirements:
    - A DCKV strategy that uses sensors to detect activity in the kitchen and then automatically adjusts the exhaust flow rates to better meet occupants' needs
    - An approach that combines DCKV with HEHs to reduce the required exhaust flow to less than that of the DCKV strategy alone.
- Improved OA delivery and conditioning strategies:
    - A make-up air unit (MAU) in the kitchen that provides air to counteract the exhaust flow rates from the kitchen
    - Interior dehumidifiers working in conjunction with a reduced-capacity main rooftop unit (RTU) that brings in unconditioned air and provides some dehumidification before supplying it to the space
    - A DOAS unit pretreating OA before supplying it to the return of a main RTU
    - A DOAS unit in full dual-path configuration that delivers conditioned OA directly to the refrigerated section of the sales floor.

- Advanced dehumidification and air-conditioning systems:
    - Variable capacity direct-expansion (DX) systems
    - Liquid desiccant systems
    - Solid desiccant systems
    - Adaptable DX-based systems with advanced controls.

Results

In general, analysis results demonstrate that significant energy savings are possible with advanced HVAC strategies in supermarkets. The best performing systems demonstrated savings of more than 50% of HVAC energy and energy costs across climates, as well as more than 30% of combined HVAC and refrigeration energy and costs. Different types of systems and strategies with a range of associated initial costs were modeled and shown to save energy and operating costs in many instances. NREL identified three promising EEMs: reduction of exhaust requirements (EEM 1), improvement of OA delivery method (EEM 2), and improved dehumidification systems (EEM 3). See details about the results in Chapter 4.

Preliminary investigations showed that inclusion of an MAU is highly recommended in all climates studied. Simulations showed a 13%–46% savings in HVAC site energy (energy consumed at the building) for the Sales and Service Zones in the climates studied and a 33%–46% savings in HVAC energy costs.

Reduction of Exhaust Requirements (Energy Efficiency Measure 1)

EEM 1 investigations showed that inclusion of a DCKV system in conjunction with HEHs can result in 11%–16% savings over the baseline MAU strategy in HVAC site energy for the Sales and Service Zones in the climates studied and 11%–13% savings in HVAC energy costs. Figure i shows the savings calculated for refrigeration and HVAC site energy and energy costs using a DCKV strategy alone or in conjunction with HEHs. In all cases, an MAU was used to deliver air to the kitchen to counteract exhaust flows. The combined strategy was found to save from 5% of combined refrigeration and HVAC energy in hot-humid climates to 10% in cold-humid climates. HEHs made more of an impact in temperate-humid climates than in very hot or very cold climates.

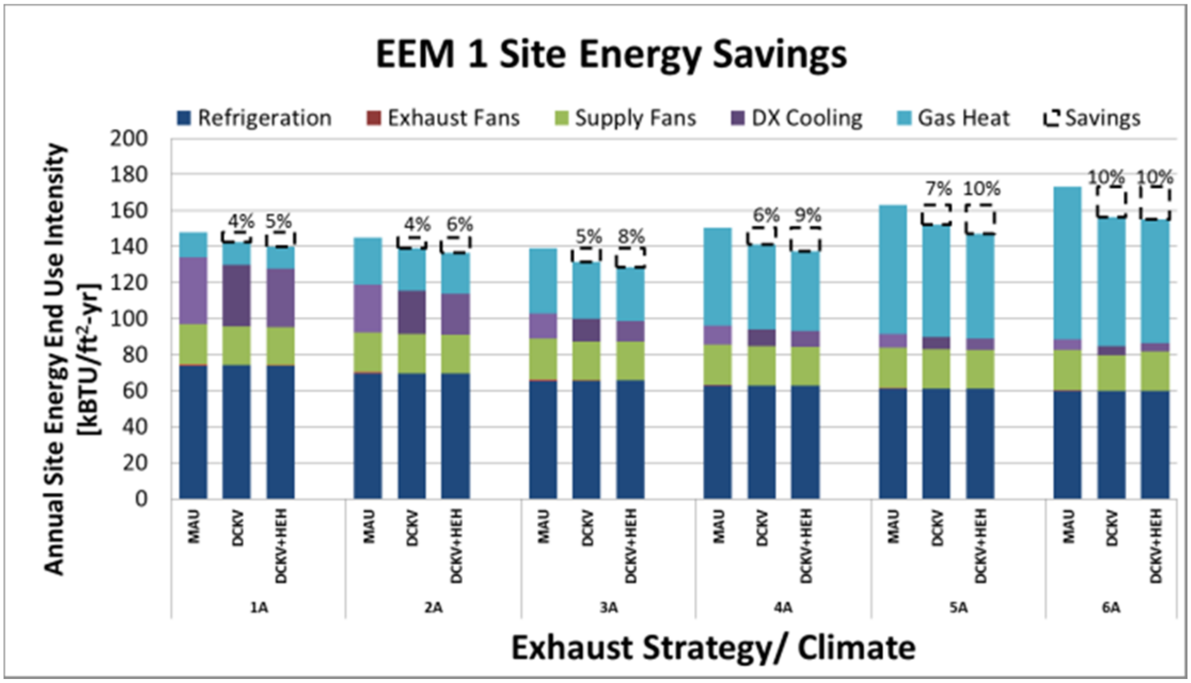

Figure i. Calculated energy savings with a DCKV scheme and with DCKV in conjunction with HEHs

Improvement of Outdoor Air Delivery Method (Energy Efficiency Measure 2)

EEM 2 investigations concluded that advanced OA delivery strategies, such as Pretreatment ("pre-treat"), DOAS, and Interior Dehumidifiers provided significant energy savings in the warmest climates but not in cooler climates. Savings over the baseline MAU strategy of up to 25% in combined refrigeration and HVAC site energy were calculated for the warmest climate studied, as well as savings of up to 47% of HVAC site energy and energy costs for the best performing systems.

The savings in energy consumption and energy costs for the Pre-Treat, DOAS, and Interior Dehumidifier strategies are shown in Figure ii. In all cases, an MAU was used to counteract kitchen exhaust flows and the baseline kitchen exhaust flow was assumed (no DCKV or HEH). Combined energy savings ranged from zero in cold-humid climates up to 25% for the Pre-Treat option in hot-humid climates.

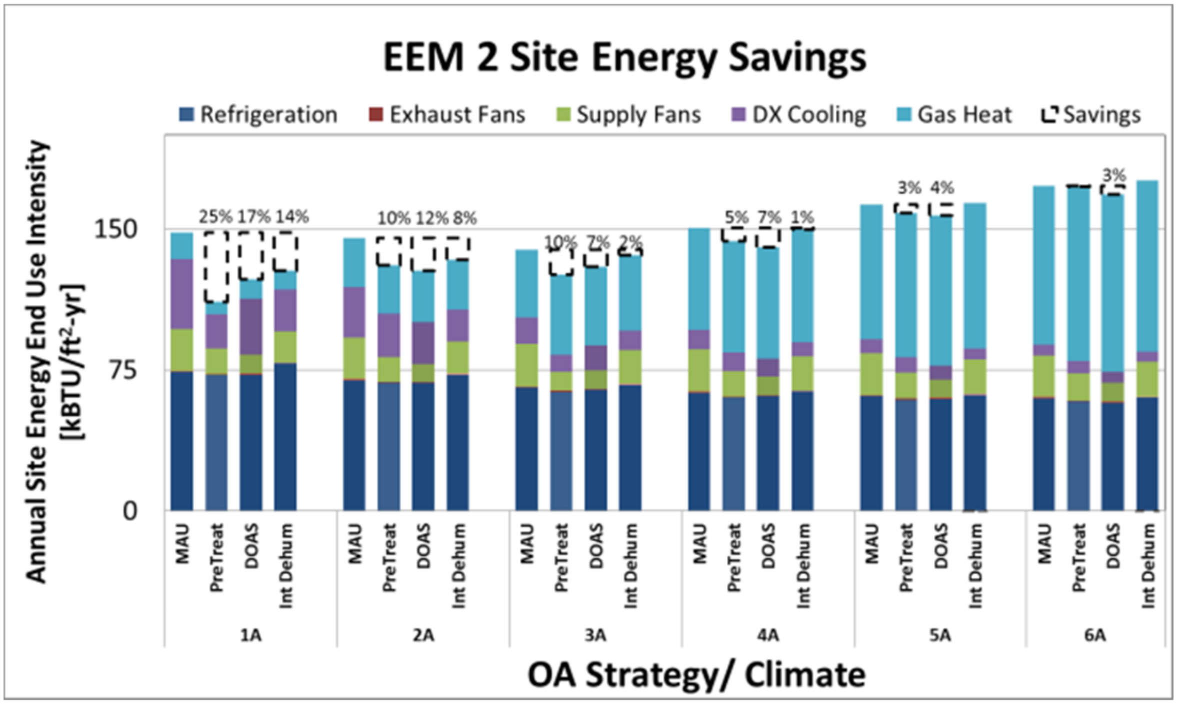

Figure ii. Calculated energy and cost savings with improved OA delivery

Improved Dehumidification Systems (Energy Efficiency Measure 3)

The final EEM was the use of novel dehumidification systems to help meet the conditioning needs of supermarkets. EEM 3 investigations demonstrated that even greater savings across all climates are available with advanced HVAC systems focused on dehumidification. For the Service and Sales Zones, we calculated total refrigeration and HVAC site energy savings of 31%–35% across all climates studied, 30%–36% combined refrigeration and HVAC energy cost savings, 49%–61% HVAC site energy savings, and 56%–62% HVAC cost savings.

As shown in Figure iii, site energy and energy cost savings of 30% or more of the total Service and Sales Zone HVAC and refrigeration energy costs were achieved with the advanced HVAC systems that incorporated dehumidification strategies. Again, these calculations were made assuming use of a kitchen MAU unit and baseline exhaust flows.

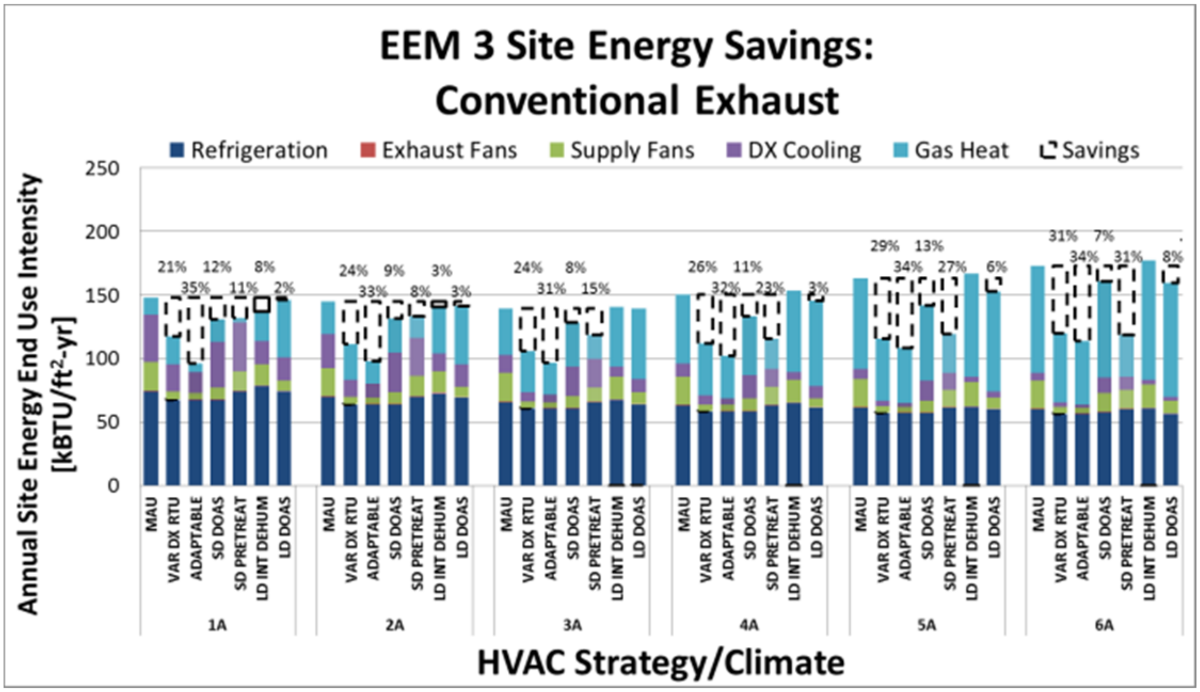

Figure iii. Calculated energy savings with improved OA delivery

In general, the calculations predicted that desiccant systems would perform best in colder climates where overcooling is detrimental, and DX-based systems were clear winners in warmer climates where additional cooling used for dehumidification could be used for sensible cooling. The Adaptive Multi-Path System and the Variable Capacity DX RTU saved a large portion of the HVAC energy in cold climates as well, as did the solid desiccant system in Pre-Treat configuration.

Conclusions

We conducted a total of 138 detailed simulations that investigated a variety of strategies for energy-efficient HVAC solutions for U.S. supermarkets. These solutions represent several different levels of complexity and associated initial costs. The best performing solutions were calculated to operate with more than 56% HVAC energy savings across all humid climates in the United States.

In this report, we also include a detailed procedure for conducting advanced supermarket energy analysis. We hope that the information provided here will result in more informed decisions about energy-efficient supermarket HVAC design and equipment selection and expand the frontiers of supermarket energy modeling.

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1 Introduction and Objectives

This research project was conducted as part of the U.S. Department of Energy's (DOE) Commercial Buildings Partnerships, a public-private, cost-shared initiative that demonstrated cost-effective, replicable ways to achieve dramatic energy savings in commercial buildings. CBP aimed to reduce energy use by 50% in new construction and 30% in existing buildings versus minimum code requirements set by ASHRAE Standard 90.1-2004 (or Appendix G of 90.1-2007), which were the applicable versions when CBP began. Companies and organizations, selected through a competitive process, teamed with DOE, national laboratory staff, and experts from the private sector who provided technical expertise to explore energy-saving ideas and strategies that were applied to specific building projects and that could be replicated across the market. The work presented here, conducted by the Commercial Buildings group at the National Renewable Energy Laboratory (NREL), focuses on reducing energy use in supermarkets.

Supermarkets are energy-intensive buildings. Also, their design, layout, and associated energy use patterns are rapidly evolving due to the increasing size of prepared food areas and new HVAC and refrigeration systems entering the market. Therefore, supermarkets present many opportunities for improvement from an energy standpoint. Thanks to the work of several researchers in the last few decades, discussed below, we understand that supermarkets operate in a way that is quite different from other building types and require unique heating, ventilation, and air-conditioning (HVAC) equipment and strategies. However, the net effect of the complex interactions of the various systems present in supermarkets on overall energy use is less understood. This type of analysis requires either construction of actual supermarkets employing the latest technologies, or detailed and original modeling performed at a level of sophistication not usually employed in building energy research.

Modeling of supermarkets must consider dynamic interaction of refrigeration equipment with the HVAC system, movement of air throughout the building, and HVAC systems that often are not included in standard software libraries. For these reasons, a great deal of extant research is limited to studying single phenomena (e.g., use of desiccant air conditioners in supermarkets) and uses simple modeling assumptions that allow for simulation of the building in a reasonable time at the expense of some accuracy (e.g., an assumption of constant kilowatt hours per ton of cooling in a dedicated outdoor air [DOAS] system). Recent developments in building simulation science allow for the inclusion of novel HVAC systems in DOE's EnergyPlus building simulation tool via co-simulation with an external program. This method allows a variety of energy efficiency measures (EEMs) to be studied and the interaction of building-related EEMs and HVAC-related EEMs with the refrigeration equipment to be understood more thoroughly.

To that end, we studied a series of EEMs in a building chosen to be representative of the current U.S. supermarket building stock. In this modeling study, we reviewed the most advanced HVAC equipment available to supermarkets, the most advanced kitchen exhaust equipment, and the variety of best practices gathered from interviews with industry professionals. We hope that the results will provide building owners, designers, and researchers with more quantitative understanding of different approaches to achieving energy-efficient supermarket HVAC, and that the simulation procedures herein help extend the frontier of building energy simulation.

1.1 Objectives

Specifically, we pursued five objectives in this work:

- Demonstrate the use of recently enhanced building-integrated HVAC modeling capabilities and publicly explain their use to building energy modelers through a study of supermarkets.
- Quantify and compare the energy impacts of using advanced exhaust equipment in supermarket kitchens.
- Quantify and compare the energy impacts of using various means of delivering and conditioning outdoor air (OA) being introduced into supermarkets.
- Quantify the advantages and disadvantages of using novel HVAC systems to condition supermarkets, with a special emphasis on dehumidification.
- Understand and quantify the interactions between the various EEMs studied, such as the effect of greater dehumidification on refrigeration energy use and the benefits of adding a DOAS system versus high-efficiency hoods (HEHs) in the kitchen, or both.

1.2 Structure of the Report

First, this report briefly explains the aspects of supermarket operation that make this building type unique and necessitate a detailed study of its HVAC systems. Then, the report introduces three sets of EEMs selected according to their energy-savings potential for supermarket applications and overall market interest based on correspondence with industry representatives. Next, the scope of the project and experimental matrix are presented. Lastly, results are presented and discussed. Appendices contain detailed information about the modeling assumptions used and the procedure for integrating building and HVAC simulation programs and sample input files.

2 Supermarket Energy Use

This section briefly describes the energy use patterns in supermarkets, the main drivers of energy use in supermarkets, and the unique facets of supermarket operation that define HVAC design. Section 2.1 presents a high-level quantification of overall supermarket energy use. Section 2.2 delineates this energy use by end use. Finally, Section 2.3 gives a detailed explanation of the unique facets of supermarket operation that drive energy use in this building type and that will be used in this study.

2.1 Overall Supermarket Energy Intensity

In general, supermarkets are very energy-intensive buildings. A 2003 Commercial Buildings Energy Consumption Survey reported that food sales account for more than 1.2 billion ft^2 of commercial building space and more than 250 trillion BTUs of total energy use per year (EIA 2005). As shown in Table 1, supermarkets ranked third in energy intensity, preceded by only restaurants and health care facilities. Supermarkets have the largest electric energy intensity (electricity use normalized by floor area) of any building type. In general, the energy intensity of supermarkets is growing because of market trends toward larger kitchen and prepared food areas, which make supermarkets more like restaurants in appearance and energy use patterns. These expanded kitchen areas require greater exhaust flow rates; therefore, greater amounts of OA must be conditioned and brought into the spaces to replace the air exhausted through the kitchen hoods.

Table 1. Energy Use Intensity, Including Both Gas and Electric (left) and Electricity Use Intensity (right) by Building Activity (EIA 2005)

Principal Building Activity	Energy Use Intensity (kBTU/sf)
Food Service	258.3
Inpatient Health Care	249.2
Food Sales	199.7
Other	164.4
Public Order and Safety	115.8
Enclosed and Strip Malls	102.2
Lodging	100
Outpatient Health Care	94.6
Public Assembly	93.9
Office	92.9
Education	83.1
Service	77
Retail (Other Than Mall)	73.9
Warehouse and Storage	45.2
Religious Worship	43.5
Vacant	20.9

Principal Building Activity	Electricity (kWh/SF)
Food Sales	49.4
Food Service	38.4
Inpatient	27.5
Health	22.9
Other	22.5
Office	17.3
Outpatient	16.1
Public Order	15.3
Non-Mall Retail	14.3
Lodging	13.5
Public Assembly	12.5
Education	11
Service	11
Warehouse/Storage	7.6
Religious	4.9
Vacant	2.4

2.2 Breakdown of Supermarket Energy Use

Several researchers have provided breakdowns of supermarket energy use by end use. Estimates vary and are climate-dependent, but the large refrigerated cases and walk-in refrigerators in supermarkets account for a large portion of the energy use (estimates range from 40%–60%). Estimates of typical HVAC energy use are 10%–25% of total building energy. This number varies considerably with building design, climate, outdoor air requirement, HVAC strategy, and refrigeration density.

Kosar and Dumitrescu (2005) report that 50% of the electricity consumption of a typical supermarket is due to refrigeration equipment and 10% is due to HVAC systems. The following chart gives a breakdown of the electrical energy expenditure in supermarkets as reported by ASHRAE (2014). This chart is useful because it shows a further breakdown of refrigeration energy. It should be noted that advanced lighting features such as light-emitting diode fixtures, dimmable lights, and skylights are experiencing large market uptake and reducing the portion of energy attributable to lighting in supermarkets.

Table 2. Breakdown of Electrical Energy by End Use From Various Sources

End Use	Percent of Total	
Walk-Ins	4%	50%
Condenser	3%	
Display Cases	15%	
Compressors	28%	
Space Conditioning		5%
Lighting		38%
Misc		7%

As shown in Table 2 (the estimates of total energy by end use) the large amount of refrigeration equipment present in supermarkets accounts for a large portion of energy expenditure. This amount is influenced by many variables, including several that are determined by the HVAC system, as discussed in the next section. This study pays particular attention to these variables to suggest means of reducing the energy expenditure of the refrigerated cases in supermarkets while also reducing HVAC energy.

2.3 Drivers of Supermarket HVAC/Refrigeration Energy Use

Supermarkets differ in their operation and in their HVAC needs from other building types, as presented in this section. Specifically, Section 2.3.1 discusses the unique sensible load situation in supermarkets, Section 2.3.2 discusses the unique latent loads and their effects on refrigeration and HVAC equipment, and Section 2.3.3 discusses the unique exhaust and ventilation requirements of supermarkets.

2.3.1 Sensible Loads

Many building types, such as offices and data centers, devote a large portion of HVAC energy to counteract sensible loads from equipment, lighting, and occupants. However, sensible loads in supermarkets, especially in the Sales Zone with its refrigerated cases and the back of the store with its walk-in coolers and freezers, are very different from other space types. Despite doors and air curtains designed to contain the microenvironment inside the refrigerated cases and walk-ins, a great deal of cold air spills into the larger space and heat is conducted and radiated through display case and walk-in boundaries as well. For the purposes of HVAC design, this is accounted for via the use of "sensible case credits," which are negative sensible loads applied to the space that represent the cooling effect from the display cases. In many cases, refrigerated cases provide the majority of the sensible cooling in the greater space of supermarket Sales Zones during the cooling season. In the heating season, the cooling provided by the cases must be counteracted with additional heat from the HVAC system.

These sensible case credits impact the HVAC operation by driving down the sensible heat ratio (SHR) demands of the space compared to a building without refrigeration. In the shoulder seasons in many climates, the space SHR may be zero or negative, signifying only dehumidification (no sensible cooling) is needed or dehumidification and some heating is needed, respectively. An electrical cooling and dehumidification strategy requires the HVAC system to cool air below its dew point temperature (DPT) for dehumidification and then provide a large amount of reheat to ensure the space does not drop to a dry bulb temperature (DBT) that is lower than desired. This is particularly a problem in cold climates. Recommendations exist for keeping space conditions at DBTs comparable to offices or other building types (75°F, 66°F–77°F) (Spyrou et al. 2013) but somewhat drier. However, in reality, building owners may be content to not provide the reheat required to maintain space conditions at this temperature during the cooling season and to allow the air around the refrigerated cases to drop in temperature below thermal comfort conditions.

2.3.2 Latent Loads

Several researchers have explained the need for drier conditions in supermarkets, especially in the refrigerated section of the Sales Zone. A good summary of the effects of space humidity on energy use in refrigerated cases is given in Kosar and Dumitrescu (2005). This review draws from several other works that go into greater detail on the effects of humidity on display case operation, including Farmarzi et al. (2000), Howell and Adams (1991), and Henderson and Khattar (1999).

2.3.2.1 Energy Concerns

The energy consequences of space humidity derive from a few different thermodynamic realities. The first is the effect on the energy use of the compressor used to cool the refrigerated display cases and walk-ins. To maintain product temperatures at desired levels, evaporators of refrigeration systems must be kept at temperatures much lower than that of a roof-top air conditioner. Medium-temperature (MT) cases typically maintain suction DPTs around 15°F and low-temperature (LT) cases maintain suction DPTs around –25°F, whereas the cooling coil in a common air handling unit may maintain a suction DPT of 50°F. The Carnot efficiency calculation shows that the lower the evaporator temperature, the lower the coefficient of performance (COP) of the cooling cycle.

In an ideal situation, the refrigerated display cases and walk-ins would not interact thermodynamically with the greater supermarket environment at all. In reality, large amounts of air infiltration from the surrounding environment into the display cases and walk-ins occur and humidity is inefficiently removed from the space by the case evaporator coil. It is obvious that the HVAC system should provide as much dehumidification as is economical to lessen the latent cooling provided by the refrigeration system.

In addition, other energy penalties occur due to humidity. Closed-door display cases must periodically run an anti-sweat cycle, accomplished by heating the glass doors through embedded electrical resistance heaters to prevent condensation from building on the door and the frames. Low-temperature cases must also go through periodic electric or hot gas defrost cycles to remove frozen condensate from their evaporator coils. Defrost cycles incur an energy penalty from the electricity used to melt the ice off the evaporator coil (if there is electric defrost) and the compressor energy to bring the case temperature back to set point (SP) after a defrost cycle.

Numerous researchers have quantified the effect of reduced space humidity on refrigeration energy use. Kosar and Dumitrescu (2005) give a summary of some of these works, which provide measured ranges of 3%–21% reduction in compressor energy use with a 20% relative humidity (RH) reduction in the space, a 4%–6% reduction in defrost energy, and a 15%–25% reduction in anti-sweat heater energy.

The degree to which the cases affect the balance of sensible and latent energy is changing significantly with the increasing prevalence of reach-in MT cases (with doors) replacing open cases. While adding doors to MT cases reduces refrigeration energy, it also may increase humidity in the space by removing the moisture sink that the refrigerated cases once were.

2.3.2.2 Aesthetic/Safety Concerns

In addition to energy concerns, humidity in supermarkets leads to other concerns. Fogging of display case windows obscures products and discourages customers. For this reason, supermarket owners often choose open (no door) display cases for much of their MT refrigerated cases, despite the fact that these cases have larger infiltration rates and thus are significantly less energy efficient.

Another concern is slip and fall risks in the aisles containing refrigerated cases. Water from the refrigerated cases can pool on the floor, causing a slippery surface. Again, this suggests drier conditions provided by the central HVAC system are desirable.

2.3.2.3 Lower Limits on Humidity Levels

System interactions limit the benefits of reductions in humidity levels, for several reasons: (1) fresh produce and other products may wilt more quickly when humidity levels in the surrounding air are too low; (2) maintaining very low humidity levels in one section of the store, such as the refrigerated section, and greater humidity in other zones, such as the kitchen, creates large vapor pressure driving forces between the zones that bring significant moisture directly to the refrigerated section, where it is unwanted; and (3) maintaining a building overall humidity level that is much lower than the outdoor condition becomes increasingly energy intensive because of the large air infiltration rates in most supermarkets that constantly add large amounts of moisture to the space. Also, large cross-envelope vapor pressure driving forces, which would be present if there were a large difference in humidity between the indoor environment and the outdoor environment, increase latent cooling demand on the HVAC system.

2.3.2.4 Desired Humidity in Supermarkets

Sources differ on the best space conditions to maintain in supermarkets. Munters recommends a 53°F DPT in the space and a 45°F–50°F supply DPT to maintain drier conditions. ASHRAE (2014) recommends a space condition less than 55% RH for proper refrigerated display case operation, with a maximum DBT of 75°F. In discussions with industry professionals involved in supermarket HVAC, the consensus was that the "sweet spot" at which humidity should be maintained is between 48ºF and 52ºF DPT, if possible. This is not always possible with traditional DX-based systems, particularly in humid climates. This study looks at the best way to maintain this level of humidity either through advanced, dehumidification-focused HVAC technologies and/or improved ventilation and exhaust strategies.

2.3.3 Exhaust and Ventilation Requirements

Most of the undesirable latent loads discussed previously are introduced into the conditioned space from the OA as opposed from internal moisture gains (i.e., occupants and produce misting). Supermarket OA requirements are unique in that the amount of OA intentionally brought into the supermarket is often dictated by make-up air (MAU) necessary to counteract large exhaust flow rates rather than ventilation or indoor air quality concerns. These large exhaust flow rates are due to the presence of cooking and cleaning equipment in the service area of supermarkets. Service areas often include more than one commercial oven and equipment such as a bread proofer, donut fryer, large range and fryer, and meat and seafood preparation areas with large ventilation requirements for odor control. In total, a typical supermarket may require make-up air quantities of around 0.5 cfm/ft^2, which is about five times the amount of air required by ASHRAE Standard 62.1-2013 for ventilation and indoor air quality.

2.3.3.1 Airflow Requirements

Supermarkets house several different thermal zones within a single space that are often not separated by walls. Each of these zones has unique conditioning requirements and zone equipment. For this reason, general airflow within and among the spaces is a crucial design consideration in supermarkets, and the fan power required to move this air is an important contributor to overall supermarket energy use. In general, a minimum amount of air recirculation must be provided at all times when the supermarket is occupied. This ensures that air remains well mixed (both horizontally and vertically) and that occupants are comfortable. Industry professionals do not completely agree on minimum supply requirements for maintaining mixed space conditions, with estimates ranging from 0.3–1 cfm/ft^2. For the purposes of this report, we chose an assumption of 0.5 cfm/ ft^2 minimum airflow during occupied hours for all zones to best reflect the consensus of available sources.

The second airflow consideration arises from the fact that exhausts are often located in a different zone than where the OA is being provided. For example, a typical supermarket may have the vast majority of its exhaust leaving from the kitchen and half of its OA being supplied to the sales floor. The OA must then have an uninhibited path to migrate to the kitchen to maintain reasonable pressurization in both zones. The sales floor should be pressurized relative to the kitchen to ensure this migration and prevent kitchen odors from reaching the sales floor.

For the reasons outlined here, the following section describes various ways of maintaining desired space conditions within a supermarket and emphasizes technologies and strategies for reducing supermarket energy use and cost.

3 Project Scope

With the unique HVAC needs of supermarkets described in the previous section, we analyzed three sets of EEMs. They correspond to three different aspects of the operating requirements for supermarkets and look at the various strategies available for reducing the energy impacts of these aspects of operation. The three EEMs are reducing exhaust air (EA) requirements, improving outdoor air delivery and distribution, and using advanced dehumidification equipment. These are discussed subsequently in Sections 3.1, 3.2, and 3.3, respectively.

3.1 Reducing Exhaust Air Requirements

Since overall HVAC energy use has a strong correlation to EA requirements, reduction of exhaust requirements is a promising area for overall supermarket energy use reduction. This reduction is currently being pursued through a few different strategies:

- Next-generation HEHs are available that more precisely engineer the air movement in the environment of the hood to capture heat and kitchen contaminants and reduce “spillage” into the surrounding environment. These hoods allow for a reduction in the required exhaust and therefore the required make-up air and its associated energy consequences.
- Demand controlled kitchen ventilation (DCKV) systems are realizing greater market uptake. A DCKV allows the exhaust flow rate to modulate in response to cooking activity in the kitchen. This modulation is achieved via optical and temperature sensors that detect cooking activity and modulate exhaust fan and interlocked make-up air fan speed in response. This saves both fan energy and the energy required to condition the make-up air.
- Additional strategies for reducing exhaust may be developed. These include, for example, strategically placing hotter cooking surfaces in the center of the exhaust hood to increase capture rate and reduce exhaust requirements.

Design specifications for EA flow rates for some systems are included in ASHRAE 90.1. Figures A1–A3 in Appendix A show the typical schedule for EA flow rates based on ASHRAE 90.1 (baseline), flow rates for a DCKV scheme, and flow rates for a DCKV exhaust schedule in conjunction with an HEH. Also shown in dotted lines are the associated OA supply schedules for each, which are large enough to offset exhaust flows plus maintain a positive pressurization in the building to avoid infiltration. It should be noted that interlocking the OA supply rate with the exhaust rate can significantly reduce OA requirements.

3.2 Improvement in Outdoor Air Distribution and Delivery

The large OA requirements in supermarkets suggest that further investigation into how the OA is delivered into the space will be beneficial. In discussion with industry practitioners, the general understanding was that supermarket designers often apply their own or general rules of thumb for determining how to deliver the required OA to the space. In this study, we quantify the energy performance characteristics of three different commonly used means of delivering OA, as well as one additional less-common method. By understanding these impacts, designers can make more informed decisions when specifying OA distribution and delivery schemes. Diagrams for each strategy are included in Figure 1.

In the first strategy, referred to as “MAS” (mixed air system) in Figure 1 and from here forward, the MAS combines untreated OA with return air (RA). The mixed air is then conditioned by a DX coil before being distributed throughout the dry goods and refrigerated sections. Supply air is conditioned to ensure that the space conditions remain near SPs of 70°F DBT and 50°F DPT. The advantages of this strategy include a simple, compact design and one set of ductwork that delivers both the OA and conditioned air to the space.

The disadvantages of the MAS strategy include the fact that a large amount of air is moved through a large pressure drop during all times when ventilation and make-up air is required, which increases fan energy use. In addition, greater supply flow rates than required to meet space loads may need to be specified to maintain an OA fraction into the cooling coils of less than 20%; this is an industry standard maximum value that ensures proper functioning of the system. In turn, this may necessitate a larger cooling capacity than is needed to maintain a desired cubic foot per meter per ton and ensure adequate dehumidification. For example, 50 tons of cooling may be required to meet loads, with a supply flow rate of 17500 cfm or 350 cfm/ton desired to provide adequate dehumidification. However, if 5000 cfm of OA are needed for make-up or ventilation purposes, a supply flow rate of 25000 cfm is required to ensure the OA fraction is below 20% and thus a 71-ton system (43% larger than is needed to meet loads) operating at 350 cfm/ton will be specified. Also, a clear flow path must be provided between the location in the store at which the OA is delivered and the place at which air is exhausted. If OA is supplied in the Sales Zone and exhausted in the kitchen, a relatively unobstructed path between the two zones must exist.

The second strategy is referred to as “MAU” in Figure 1 and from here forward. The mixed air RTU in this strategy introduces only the amount of OA to satisfy ventilation requirements, unconditioned, into the return of the main RTUs. A separate make-up air unit (MAU) delivers OA necessary to balance the exhaust in the kitchen and delivers this directly to the kitchen. In this case, the MAU conditions the air only as much as needed to provide for a minimum amount of comfort in front of the cook line. The mixed air RTU ensures the dry goods section is maintained at SPs of 70°F DBT and 50°F DPT. The refrigerated section conditions float based on the supply air delivered to this space and cross-mixing with the dry goods section of the Sales Zone.

The advantages of this strategy are: (1) the main RTU is smaller and more efficient because of the smaller burden from large OA flow rates; (2) the kitchen MAU can be interlocked with the exhaust fan to provide OA only when exhaust flows require it; and (3) designers need not be concerned with providing a clear path for transfer air from the sales zone to the kitchen. Disadvantages are that two sets of HVAC equipment (RTU and MAU) are required and the main RTU still must move a relatively large amount of air whenever the space is occupied.

One improvement that can be made to the MAU strategy is the inclusion of an interior dehumidifier in the refrigerated section of the Sales Zone. This has been studied thoroughly (Fricke and Sharma 2011). We look at this scenario as well, which is referred to as “Interior Dehumidifier.” All “Interior Dehumidifier” configurations include an MAU in the Service Zone as well for easy comparison to other configurations/systems.

The Mixed Air System with DOAS Pretreatment, “Pre-Treat,” preconditions all the OA to a dry, cool condition before it is mixed with the return air. The mixed air RTU then provides mostly sensible cooling based on the demands of the dry goods section—the refrigerated section space conditions are not actively controlled and will float. The advantages of this strategy are: (1) the DOAS pretreatment handles most of the latent load and thus the main RTU can be smaller and more efficient; (2) only one set of ductwork is required; and (3) all the OA can be brought in through the main system. Disadvantages include: (1) a large amount of air must be moved constantly through a large pressure drop whenever the building is occupied or exhaust flows; and (2) a clear path is needed for the make-up air to migrate from the sales floor area to the exhaust locations in the service area.

The DOAS with Recirculation System, “DOAS,” has the DOAS providing conditioned OA through separate ducting to the refrigerated section. Unlike the other scenarios, the DOAS actively controls the DBT and DPTs in the refrigerated section. A separate RTU conditions recirculated air to maintain the DBT and DPT SPs in the dry goods section. The DOAS conditions the OA to a very dry condition and thus handles the majority of the latent load for the entire space.

This strategy offers several advantages. These include the fact that the fan in the recirculation RTU delivers only a minimum amount of airflow to maintain mixing or the minimum to maintain sensible conditions, rather than the large amount of air required to dehumidify OA in a mixed air stream. Thus, fan power is greatly reduced in the recirculation RTU. The recirculation RTU may be sized smaller and configured for faster air speeds across the evaporator coil for increased SHRs because it no longer needs to handle the OA load, and the refrigeration systems operate more efficiently because of drier microclimatic conditions (see Section 2.3.2). Disadvantages include the increased capital costs of the two systems and two sets of ductwork being required (one supplying recirculated air to the dry goods section and one supplying conditioned OA to the refrigerated section).

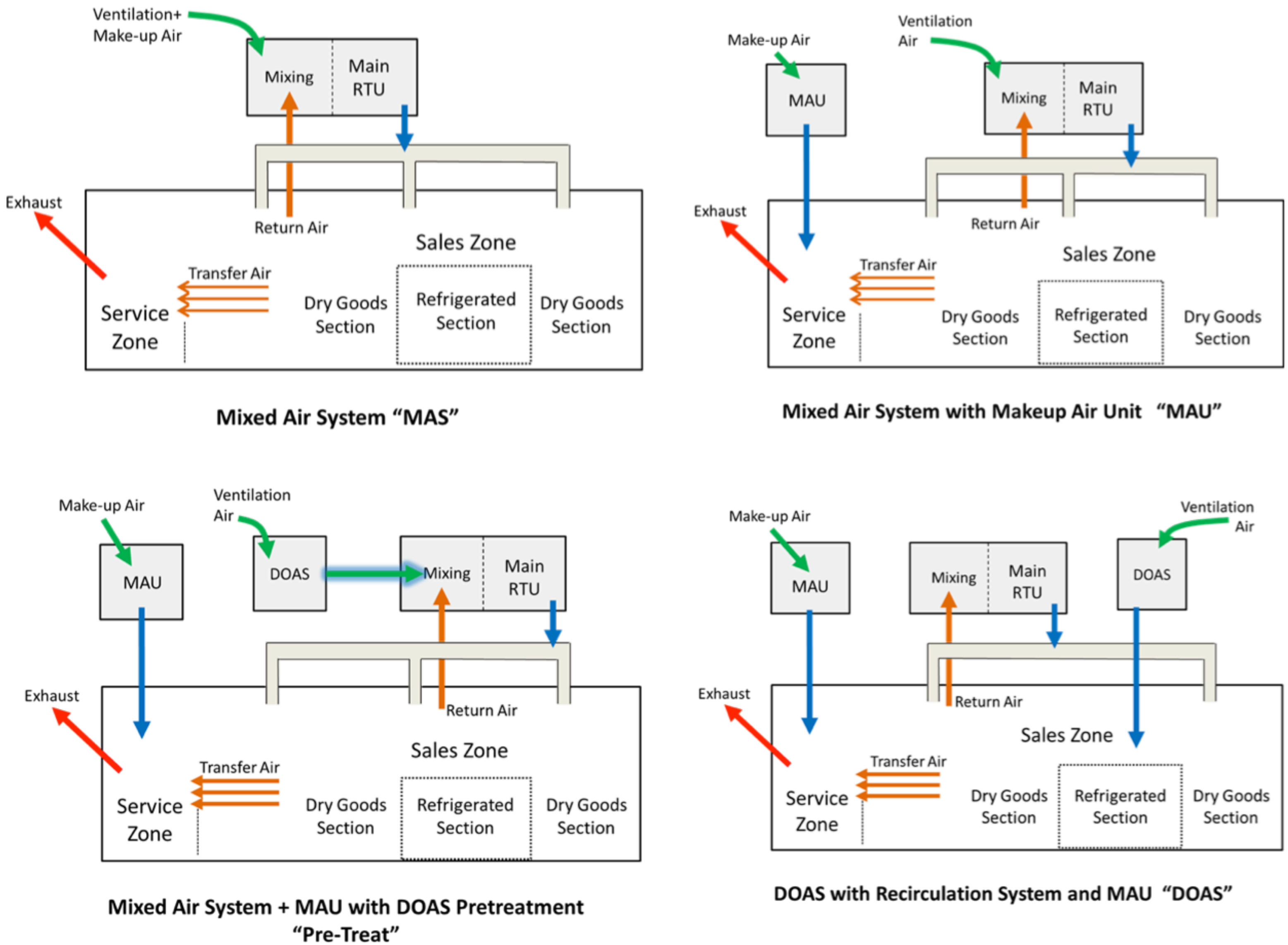

Figure 1. OA distribution schemes

3.3 Advanced Dehumidification Systems

Because of the large quantity of OA necessary and the low space humidity levels desired for efficient operation of the refrigeration systems, more advanced dehumidification systems may offer cost and energy savings in supermarkets compared to dehumidification provided by conventional fixed-capacity direct expansion cycles in roof-top mixed air units. To evaluate these benefits, we quantified the savings available with seven advanced dehumidification systems in various combinations with exhaust and OA strategies mentioned above. Additional advanced HVAC systems were considered but not included in the analysis after consultation with manufacturers and sales professionals because of lack of interest or applicability for supermarket applications. These systems are described in Appendix E.

3.3.1 Baseline Direct Expansion System

The baseline system is a direct expansion (DX) air conditioner that conditions mixed air. The psychrometric process and a schematic of the system are shown in Figure 2. In Figure 2 and all subsequent figures, "OA" designates a design outdoor air condition, "SP" designates a space temperature and humidity SP, MX designates the mixed air condition, red lines on the psychrometric chart represent heat and moisture transfer processes that are accomplished with an electric or natural gas input, and green lines represent passive processes.

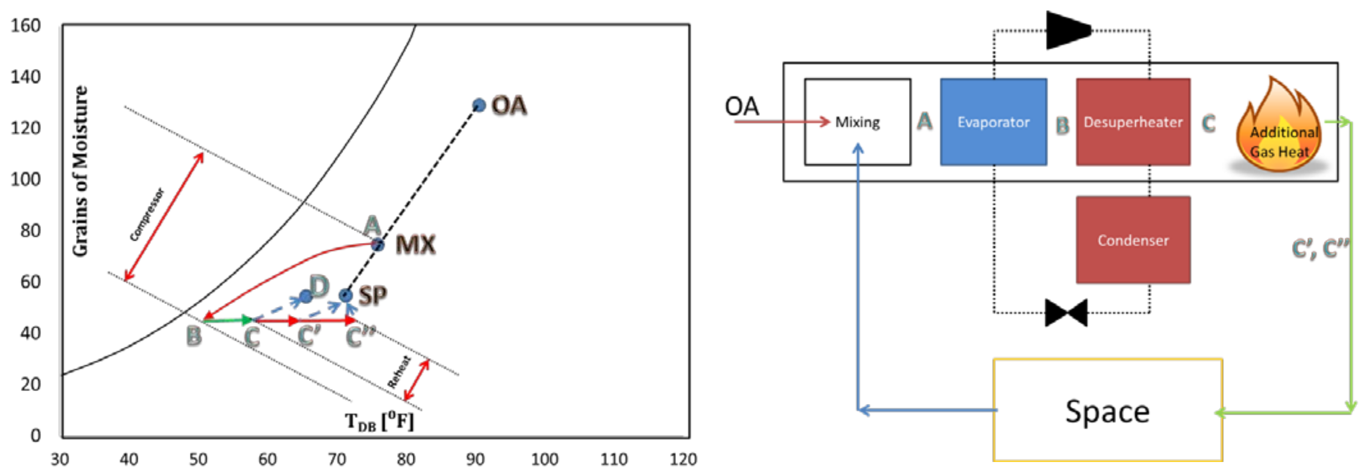

Figure 2. Baseline system psychrometric processes and system schematic

Supply air moves at a constant volume through the system and the compressor cycles on and off in response to thermostat and humidistat signals. Heat rejected from the cooling cycle of the DX system and additional gas furnace reheat are often needed to ensure thermal comfort in the space. The main advantage of this type of system is that it is compact with low initial costs. No additional units are needed. Also, one set of ductwork delivers both the OA and recirculated air.

Khattar and Brandemuehl (2002) have enumerated the disadvantages of this type of system. These include:

- Poor humidity control, because this system is intended mainly for handling sensible loads.
- The necessity for simultaneous heating and cooling caused by a need to overcool to dehumidify and then reheat in order to maintain comfortable space temperatures, because refrigerated cases negate much of the positive sensible loads in the space.
- Designers must choose either improved energy efficiency ratio (EER) and poorer dehumidification performance provided by a system with a high airflow/capacity or cfm/ton ratio, or the improved dehumidification performance of a low cfm/ton device, with a consequent decrease in EER.
- The need for a constant speed fan to run continuously and move a large amount of air through a system with a large pressure drop. Simulations in this study show that fan electricity use is of the same magnitude as the DX coil annual electricity for many buildings conditioned by this configuration.
- Re-evaporation of water from the evaporator coils into the airstream caused by frequent cycling of the compressor. Henderson et al. (2003) showed that, because of this phenomenon, as the run-time fraction of the cooling coil drops below 0.5 (which may occur frequently in supermarkets during the shoulder seasons) the SHR of the system approaches 1, meaning no dehumidification is provided. Conversely, supermarkets experience an appreciable number of hours annually when *only* dehumidification is needed.

3.3.2 Improved Mixed Air Systems

Several improvements to traditional DX systems provide a lower SHR without reheat, which improves performance at part-load conditions. These improvements are described next.

3.3.2.1 Variable-Capacity Direct Expansion Systems

Variable capacity HVAC compressors, using either digital scroll technology or a variable frequency drive connected to a compressor, allow the system to continuously modulate its cooling rate based upon the cooling demands of the space, which are rarely at peak design conditions. Conversely, in typical DX systems, constant speed compressors cycle to adjust the cooling delivered over time. This leads to start-up inefficiencies as well as excessive wear on the compressor motor. In many cases, because of the lack of capacity modulation, 90% of system power capacity is used to meet a load of only 50% (Mehltretter 2014). Variable capacity compressors offer continuously adaptable control of the delivered load, reduced noise and vibration due to gradual startup and reduction of switching on and off, a low starting current, more uniform temperature control, and better COPs when part-load conditions are the predominant operational state, which is the case for most buildings throughout the year (Cuevas and Lebrun 2009).

Variable capacity compressors are not recommended for certain applications because they may result in greater energy consumption. For variable capacity compressors that include a variable frequency drive, losses in the drive and motor reduce efficiency 3%–5% below a constant speed compressor (Mehltretter 2014) while switching losses in the inverter also exist (Cuevas and Lebrun 2009). These variable frequency drive losses must be recouped by savings gained at part-load conditions to make economic sense. One example where constant speed compressors provide improved performance are applications that require a large year-round sensible load, such as data centers in a hot, dry climate. Proper sizing and staging of the compressors would minimize cycling losses, and with the focus being on sensible cooling, the system would actually perform better at an SHR approaching 1. In practice, variable capacity compressors should be paired with variable speed supply fans to deliver maximum efficiency (Wang et al. 2011).

Several systems with variable capacity compressors are currently commercially available and typically include one or two variable capacity compressors in conjunction with one or two constant speed compressors. Constant speed compressors are responsible for handling the long-term fluctuations in load, while the variable capacity compressors are responsible for the short- and medium- term fluctuations (Mehltretter 2014).

3.3.2.2 Adaptive Multi-Path System

One improvement to the variable capacity DX system is the inclusion of dampers that optimize how the OA is mixed with the RA before entering the evaporator coil. This system, called the "Adaptive Multi-Path System" in this study, contains three independently modulating dampers that control the mixing ratio of the air stream and allow the DX coils to condition 100% OA, mixed air, or 100% RA. In this system, shown in Figure 3, optimized control algorithms ensure the most energy-efficient delivery of ventilation air and thermal comfort possible by modulating the position of the dampers and the system capacity. By adjusting the mixing of the air streams and modulating the capacity of the coils simultaneously, the Adaptive Multi-Path System continually modulates the system SHR to match the space loading SHR. Also included is a damper that allows return air to bypass the coil completely. Variable speed fans are also used to provide the most efficient conditioning.

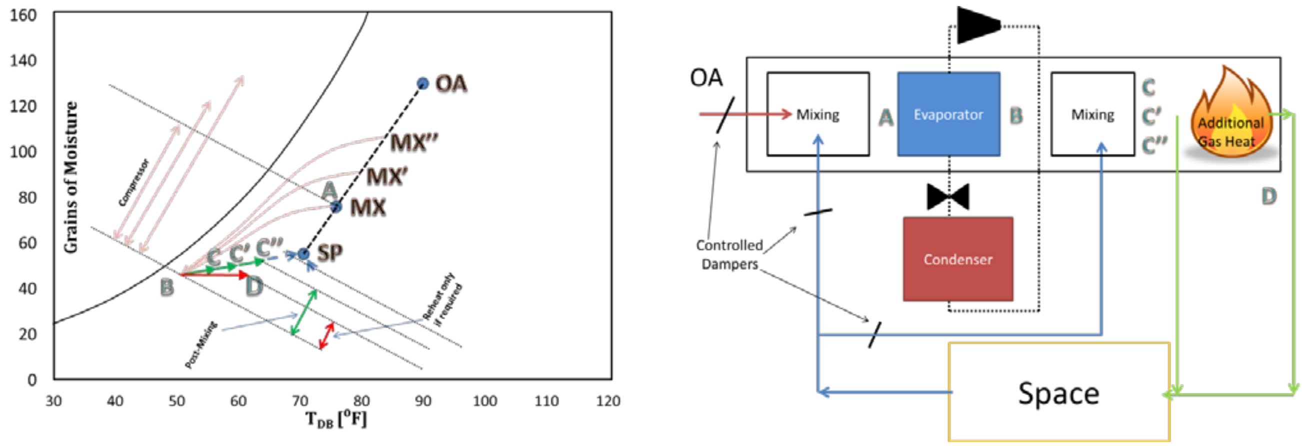

Figure 3. Adaptive Multi-Path System psychrometric processes and system schematic

3.3.3 Interior Dehumidifiers in Refrigerated Case Zone

In applications in which the latent load in one area of the store is very high and the sensible load is small or negative, as in the refrigerated section of a supermarket, a dedicated interior dehumidifier may provide an energy-efficient means of conditioning the space. Fricke and Sharma (2011) have studied this concept in detail. The two types of interior dehumidifiers studied are described next.

3.3.3.1 Liquid Desiccant Dehumidifier

Liquid desiccant (LD) systems have been gaining interest recently as a means of providing dehumidification without using vapor compression overcooling and reheat. Many different types of LD systems have been proposed. A review of available technologies is given in Lowenstein (2008). Among these is the Lithium Chloride low-flow liquid desiccant system studied in this project, described in detail in Lowenstein et al. (2006) and Lowenstein (2004).

Many different advantages of LD systems have been demonstrated. First, the most fundamental function of an LD system is to shift energy consumption from the electrical input of the compressor in a DX system to a thermal input in the regenerator of an LD system (Lowenstein 2008). In lieu of electricity, LDs can use waste heat, thermal energy generated onsite, such as a gas-fired boiler, or solar thermal energy, to name a few. Another advantage over DX systems is that desiccant systems on their own do not require synthetic refrigerants as do vapor compression systems, although LDs often must be combined with sensible cooling devices that may or may not involve the use of refrigerants. Lastly, because they dehumidify through contact with hygroscopic media rather than overcooling and reheat, desiccant systems can achieve DPTs below the freezing point of water, which would cause ice build-up in conventional systems (Lowenstein 2008).

The particular LD system modeled is not commercially available at the time of writing. At least one high-flow liquid desiccant system is commercially available, but performance data for this product was not available for this study.

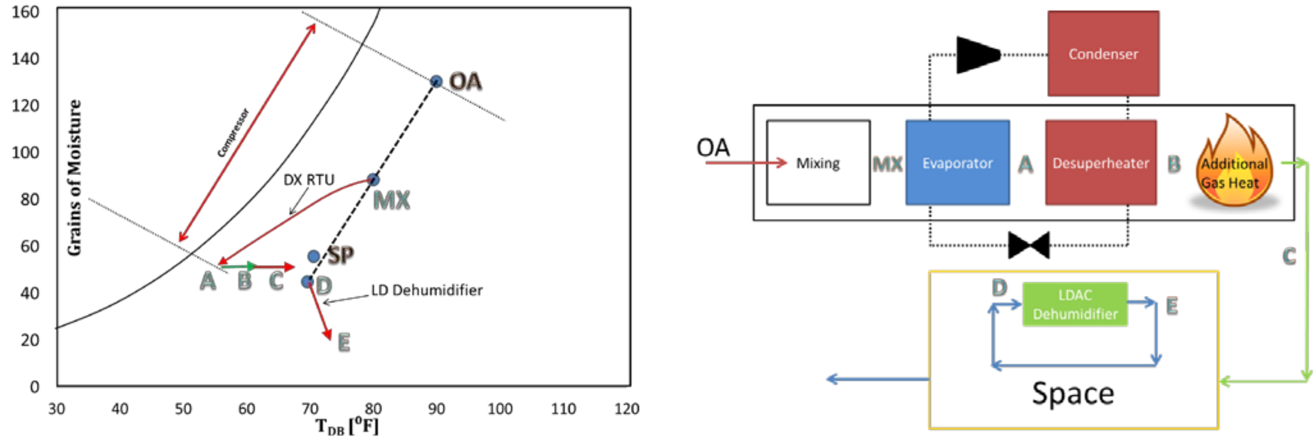

Figure 4. LD interior dehumidifier

3.3.3.2 Direct Expansion Dehumidifier

Dehumidifying locally with a DX dehumidifier in the refrigerated section of a supermarket offers similar advantages similar to an LD dehumidifier, by reducing the latent load on the refrigerated cases. The difference is that dehumidification with the DX dehumidifier is accomplished mechanically with an overcool-reheat strategy rather than chemically. Figure 5 depicts the system and processes.

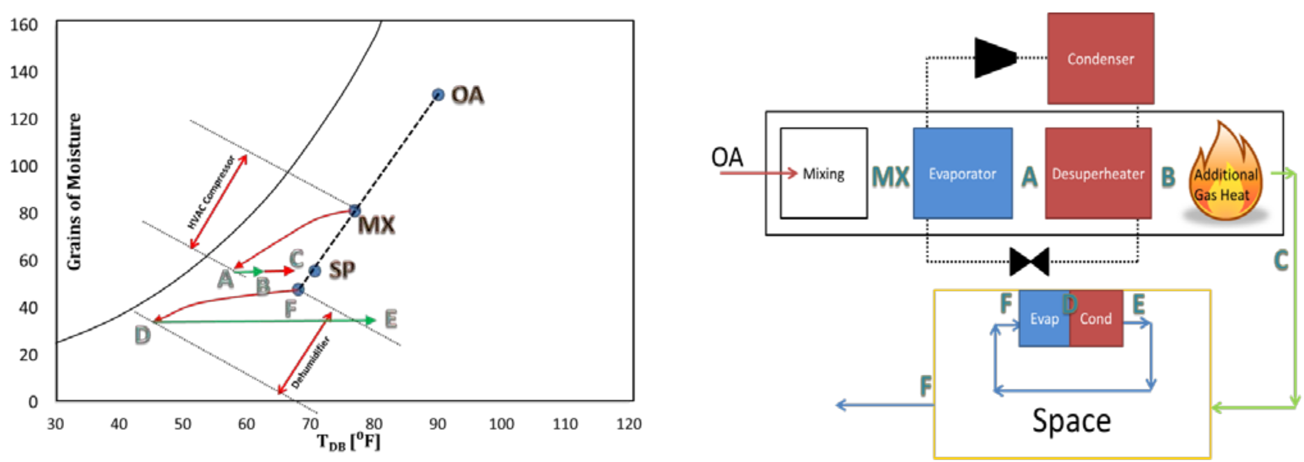

Figure 5. DX interior dehumidifier

The system modeled is a commercially available, ENERGY STAR®-rated system that is simple but effective. It contains an evaporator coil with a drip pan and a full condenser in the supply path providing reheat. Air exits the dehumidifier drier and warmer than the entering air, thus dehumidifying and bringing the refrigerated section to a more comfortable temperature. The system was originally designed to be a residential whole-house dehumidifier with enough capacity to treat ventilation air, if necessary. Fricke and Sharma (2011) estimate that a 20% savings in refrigeration and HVAC energy is achievable with this type of system, as well as a 27% reduction in gas use.

3.3.4 Dedicated Outdoor Air Systems

DOAS, also called dual path systems, have been used in applications in which a relatively high latent load exists, especially if this load is due to ventilation. The basic principle behind DOAS is that one system treats ventilation air by bringing it to a low DPT and this provides for all of the dehumidification needs of the space. Another parallel system treats only recirculated air or provides radiant cooling, and does 100% or nearly 100% sensible cooling. This can reduce total fan power used by 20%–30% (Mumma 2001), provide for dehumidification in a more efficient manner, and even reduce capital costs by around 15%, contrary to popular perception (Mumma 2001). Next, we briefly describe three DOAS system categories.

3.3.4.1 Direct Expansion Dedicated Outdoor Air Systems

The first DOAS strategy studied involves operating two DX systems in parallel. This system is usually run with a total enthalpy recovery wheel between the exhaust stream and the ventilation air stream (Mumma 2001). However, it is not clear that exhausting sufficient amounts of air from supermarkets without underpressurizing the building is possible. This is due to exhaust fans already operating in the bakery and deli zones and the large amount of infiltration that is present in most supermarkets due to doors being open for large portions of the day. These systems are usually operated with the ventilation air stream being dehumidified with a 42°F–48°F coil in office and retail buildings to remove enough moisture to provide for all of the space's dehumidification (Mumma 2001). However, for supermarkets, common practice is to keep the coil as cool as possible without freezing to dehumidify as much as possible. DBTs of both supplies are controlled to 55°F or below (Mumma 2001). This strategy is shown in Figure 6.

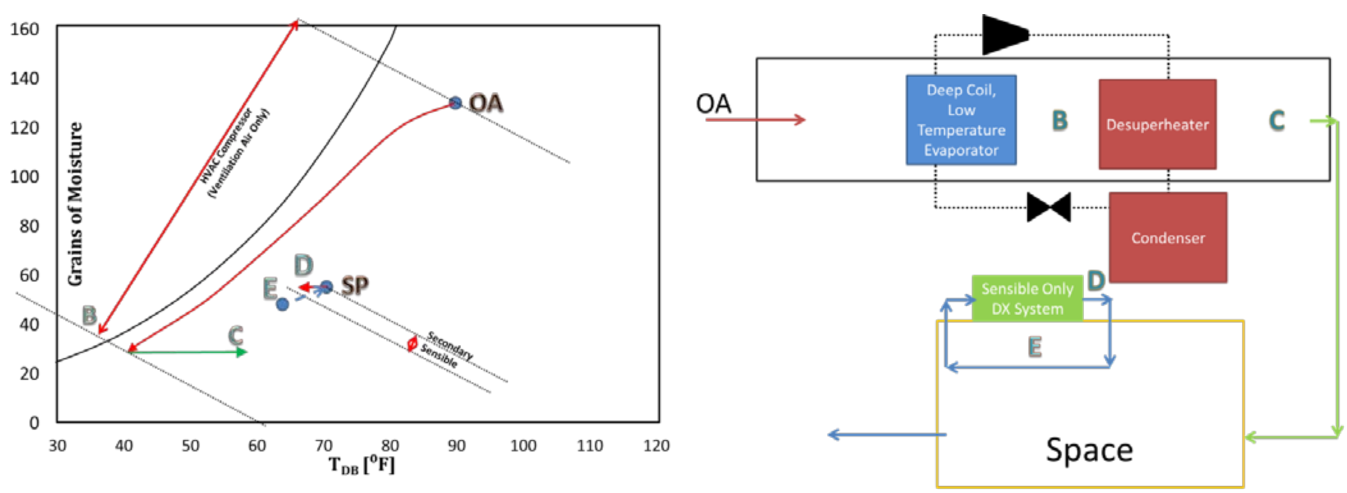

Figure 6. Psychrometric process and system schematic for DX DOAS strategy

There are several benefits to such a system. First, the sensible conditioning of the recirculation air can be done with a DX coil operating at a much higher evaporator temperature because only sensible cooling is being done. This leads to higher system COPs. This strategy also reduces fan power by constantly supplying only the quantity of air needed for ventilation, rather than constantly recirculating a large quantity of room air (Mumma 2001). It also eliminates the problem of off-the-shelf air conditioners having airflow per ton quantities larger than those needed for efficient dehumidification (Morris 2003).

For the purposes of this study, we also analyzed a system that has the DX DOAS system placed upstream of the main RTU in a "Pre-Treat" configuration. This is a convention often adapted in practice because it eliminates the need for a separate set of ductwork to bring the OA to the space. OA is preconditioned by a dedicated system before being introduced into the return of the main RTU that allows the main RTU to be downsized and run at a greater airflow per ton, thus making it more efficient.

3.3.4.2 Liquid Desiccant Dedicated Outdoor Air Systems

This strategy involves using the LD system described above in Section 3.3.3.1 to dry the ventilation air to a very low DPT and a parallel DX system to provide for sensible cooling. The LD system dries the air as much as possible, usually to around 20% RH, and it is delivered to the space. Little is known about the performance of this configuration in supermarkets because it has not been studied. The system is depicted in Figure 7.

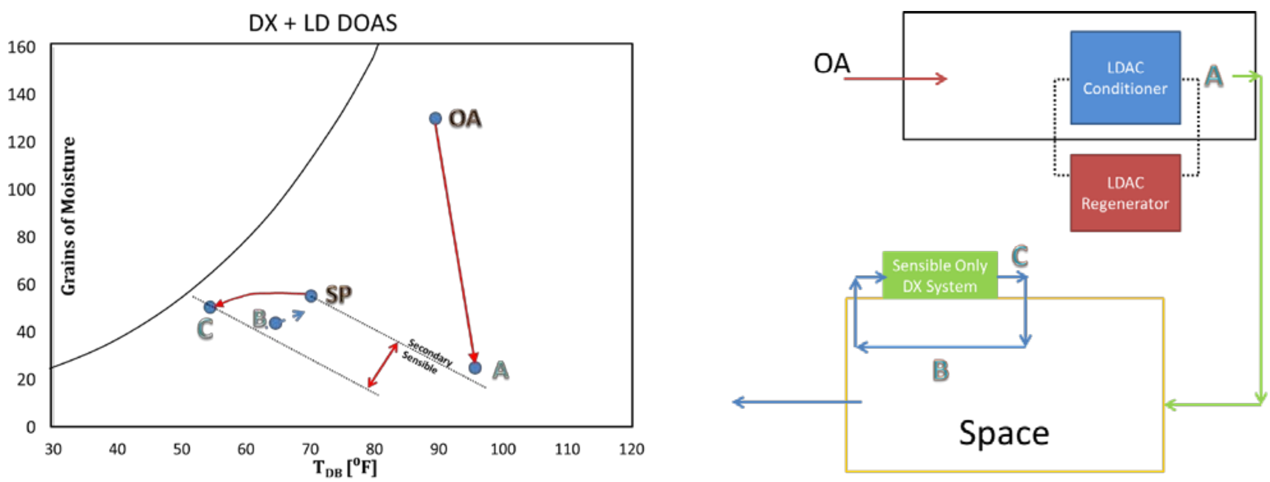

Figure 7. Psychrometric processes and system schematic for LD DOAS system

3.3.4.3 Condenser Heat-Regenerated Desiccant Wheel Dedicated Outdoor Air System

The last system studied, shown in Figure 8, is a dual-path solid desiccant (SD) dehumidifier that dehumidifies and cools air in one stream and uses condenser heat to regenerate the desiccant wheel in the parallel stream before exhausting the regeneration air. The wheel used in this system is also a Type 3 wheel (Kosar et al. 2007). Before entering the wheel, the air is precooled by a four-stage DX cooling coil. This configuration is capable of delivering about a 45°F DPT at design conditions.

Several benefits and shortcomings of this system have been suggested. First, it was shown to reduce the SHR further than either a wrap-around heat pipe (WAHP) or wrap-around desiccant wheel (WADW) system (Kosar et al. 2007). This was done, however, at the expense of a substantial increase in system power use, owing to the large amount of fan power needed to move air through each side of the desiccant wheel. Mumma (2007) describes this configuration as a good choice when low DPTs are needed, or when return air is not available. Supermarkets meet both of these criteria.

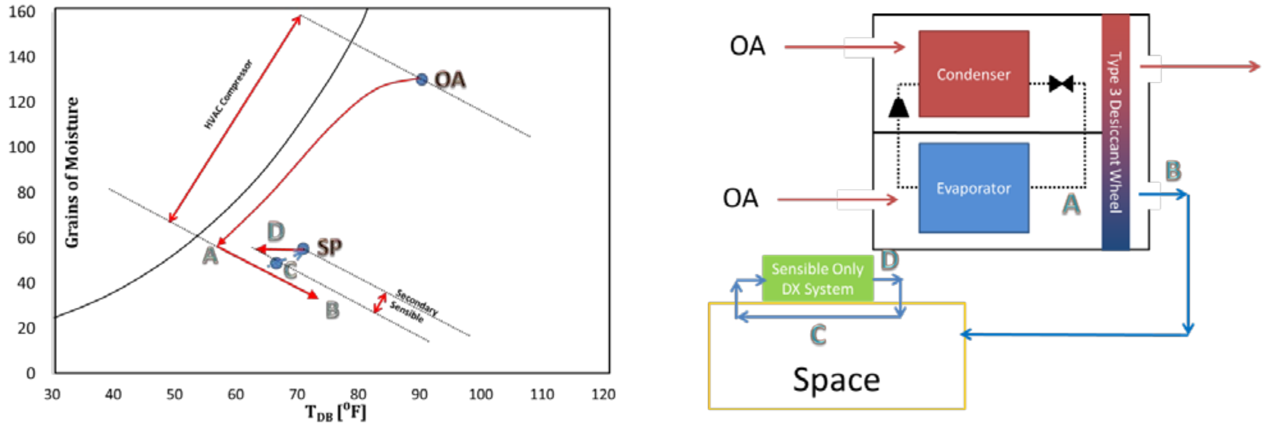

Figure 8. Psychrometric processes and system schematic for condenser heat-regenerated SD wheel DOAS strategy

3.4 Experimental Matrix

To study the three EEMs, we simulated several different combinations of the various EEMs in a typical reference building over the course of the year. We chose the combinations of the EEMs in consultation with HVAC professionals and manufacturers to study the strategies most likely to be implemented in practice and that would result in the greatest energy savings. We performed full-building simulations for each of these strategies over the course of a Typical Meteorological Year 3. Readers can refer to Appendix A to see details of the building, loads, system models, schedules, and other assumptions.

We conducted these simulations using DOE's EnergyPlus building simulation software, which we enhanced by co-simulating EnergyPlus with models of advanced systems not included in the EnergyPlus libraries. We wrote these models in the Modelica modeling language and simulated them with the Dymola software package. Detailed information on the integration of the two software packages can be found in Appendix B. See comparisons of the building simulation results obtained with each of the two programs in Appendix D.

We performed simulations for locations in which it is expected that the greatest benefits will be derived. Because many of the opportunities for improvement in HVAC performance in supermarkets are available through improved dehumidification and reduction of latent loads, we investigated climates with significant humidity loads, including both hot and cold climates (ASHRAE "A" climate zones). These areas include the entire eastern half of the United States, including many of the major population centers, and regions with high population density. Table 3 shows a list of the modeled representative cities and the corresponding Typical Meteorological Year weather file used for OA assumptions.

Table 3. Climate Zones and Representative Cities

No.	Climate Zone	Representative City	EPW Weather File Source
1	1A	Miami, Florida	Miami International Airport
2	2A	Houston, Texas	Bush Intercontinental Airport
3	3A	Atlanta, Georgia	Hartsfield-Jackson International Airport
4	4A	Baltimore, Maryland	Baltimore-Washington International Airport
5	5A	Chicago, Illinois	Chicago-O'Hare International Airport
6	6A	Minneapolis, Minnesota	Minneapolis-St. Paul International Airport

Table 4 summarizes the experimental matrix with the strategies simulated for each climate. We set up the experimental matrix and presented the results later in a manner such that comparisons could be made between sequential simulations. The first three simulations may be thought of as alternative baseline systems. MAS-a is an unrealistic situation, but one that is necessary to maintain a 50ºF DPT throughout the summer in humid climates using only a rooftop unit. MAS-b is the same system, but sized in a more realistic manner. MAU includes a make-up air unit and is the strategy often specified in supermarkets, although discussion with practitioners indicated that MAS-b is also specified in many instances. The MAU scenario is the baseline against which we compare all other systems in the results section. The other systems are organized by the EEM studied. We simulated each system marked with an asterisk in Table 4 under reduced exhaust conditions (due to the presence of HEHs and a DCKV system) and we present results for this scenario in Table 4.

Table 4. System Combinations Considered in the Study

Designation	EEM	Exhaust Schedule	OA Delivery	Mixed Air System	Recirc System	OA System	Simulation Program
MAS-a	Baseline	Conventional	MAS	Oversized Mixed Air DX RTU			EnergyPlus
MAS-b	Baseline	Conventional	MAS	Mixed Air DX RTU			EnergyPlus
MAU	Baseline	Conventional	MAU	Mixed Air DX RTU		MAU	EnergyPlus
MAU + DCKV	1	DCKV	MAU	Mixed Air DX RTU		MAU	EnergyPlus
MAU+ DCKV+ HEH	1	DCKV+ HEH	MAU	Mixed Air DX RTU		MAU	EnergyPlus
DX PRETREAT*	2	Conventional	Pretreat	Mixed Air DX RTU		MAU +DX PRETREAT/ RTU	EnergyPlus & Modelica
DX DOAS*	2	Conventional	DOAS		DX	MAU + DX DOAS	EnergyPlus & Modelica
DX INT DEHUM*	2	Conventional	MAS	DX	DX Dehumidifier	MAU	EnergyPlus & Modelica
LD INT DEHUM*	3	Conventional	MAS	DX	LD Dehumidifier	MAU	EnergyPlus & Modelica
VAR DX*	3	Conventional	MAS	Variable DX RTU		MAU	EnergyPlus & Modelica
LD DOAS*	3	Conventional	DOAS		DX	MAU + LD DOAS	EnergyPlus & Modelica
SD DOAS*	3	Conventional	DOAS		DX	MAU + SD DOAS	EnergyPlus & Modelica
SD PRETREAT*	3	Conventional	Pretreat	DX		MAU + SD PRETREAT/ RTU	EnergyPlus & Modelica
ADAPTABLE*	3	Conventional	Adaptable System				EnergyPlus & Modelica

*Denotes system modeled with both conventional and DCKV+HEH exhaust rates

4 Summary of Results

The modeling results revealed significant savings are available with advanced supermarket HVAC strategies in climates with significant humidity loads. The following subsections summarize the results of the study of the three EEMs in terms of site energy, energy costs, and source energy. Site and source energy results are presented in terms of the contribution of the refrigeration system and Sales and Service Zone HVAC systems to whole-building energy use intensity. Energy costs are presented as annual site-specific gas and electricity costs for the refrigeration system and the HVAC system in the Sales and Service Zones only. Other zones were not coupled to these two zones in our simulations, and so were not considered in the comparisons. Refrigeration energy was included in the presentation of results because it was affected by alternative HVAC strategies. Note that the savings presented would be much greater if shown as a percentage of HVAC energy only.

4.1 Comparison of Alternative Baselines

We first compare the three alternatives for a baseline system: an oversized MAS (MAS-a) that maintains DPT SPs throughout the year, a more realistically sized MAS-b, and a combined MAU + MAS system (MAU). As a reminder, the MAU scenario brings 6500 cfm of OA into the Sales Zone via a mixed air RTU and the remaining portion of OA necessary through the MAU in the Service Zone. The MAU scenario is the baseline against which all EEMs are compared in Section 4.2 and beyond.

In general, the study decided fairly conclusively that introducing all make-up air through a centrally located MAS rooftop unit and allowing it to travel to the service area is a very inefficient strategy for supermarkets. Figures 9–11 show that the MAU solution provides significant savings across all climates in terms of:

- Site energy (13%–31% combined refrigeration and HVAC, 13%–46% HVAC only)
- Energy costs (10%–32% combined HVAC, 33%–46% HVAC only)
- Source energy (22%–29% combined refrigeration and HVAC, 34%–46% HVAC only).

This observation holds true for several reasons: first, a large RTU must be specified to bring in the large quantity of OA and maintain an OA fraction in the mixing box that is at or near the 0.15–0.2 value desired. Maintaining this OA fraction is necessary to ensure a state of air going into the evaporator coils that allows the coils to perform adequate dehumidification. This results in high fan power consumption and large systems, in general. However, it should be noted that the lower quantity of heat generated by the supply fan with the more compact systems needs to be made up with natural gas in heating climates. Also, to provide for efficient refrigerated case operation, all OA has to be dehumidified, rather than just the smaller amount going to the refrigerated display cases. The MAUs need only temper the air for the comfort of the staff using kitchen equipment; for this study, the MAUs heated the air to 55°F and cooled the air to 65°F. Furthermore, inclusion of an MAU allows for easier interlocking of exhaust and OA systems, which adds motivation for including an MAU.

The MAU strategy provides better humidity control than the MAS-b system, which is not capable of always maintaining the desired DPT. System MAS-a, on the other hand, provides dehumidification control similar to the MAU, but is sized unrealistically large and has associated capital costs. Consultations with supermarket HVAC professionals suggested that the more efficient manner of using an MAU is implemented in a majority of cases. However, the same supermarket HVAC professionals stated that it is not uncommon for a large mixed air RTU to be used instead of MAUs, which is very energy inefficient.

In light of these results, we use the MAU scenario as the baseline against which all other HVAC strategies are compared.

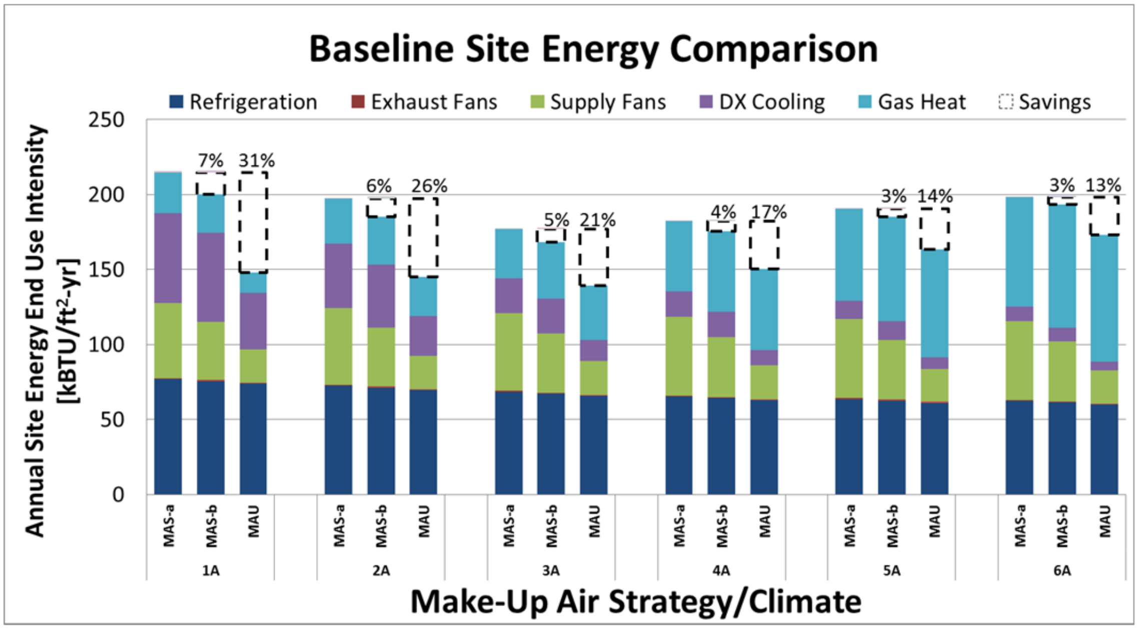

Figure 9. Site energy comparison for three alternative baseline systems

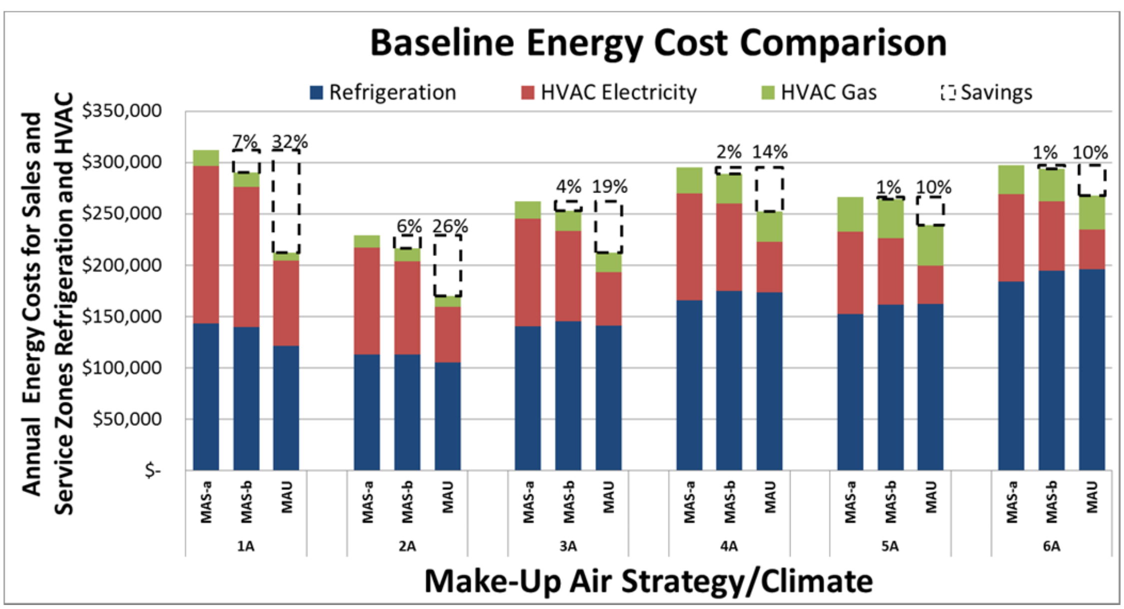

Figure 10. Energy cost comparison for three alternative baseline systems

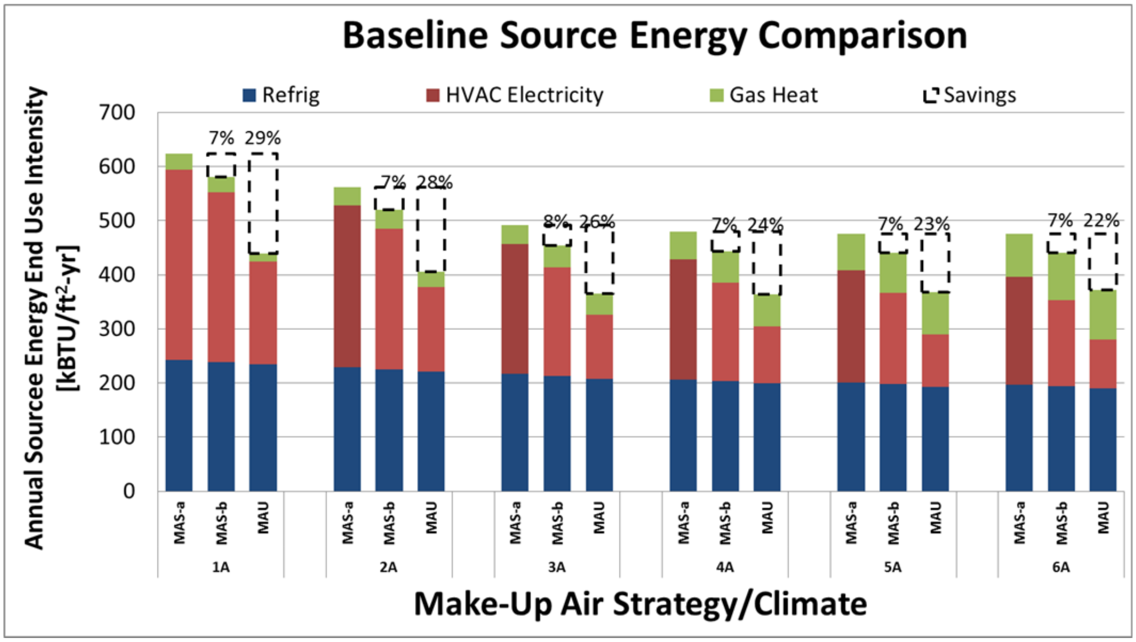

Figure 11. Source energy comparison for three alternative baseline systems

4.2 Energy Efficiency Measure 1: Exhaust Reduction Strategies

In general, the reduction of exhaust requirements produced significant savings. Figures 12, 13, and 14 show the site energy, energy cost, and source energy savings, respectively, for HVAC and refrigeration systems in the Sales and Service Zones, as modeled. Colder climates showed the greatest energy savings owing to the large relative energy expenditure on heating OA. This conclusion is heavily dependent on the assumed exhaust and OA supply schedules. It also is sensitive to the assumed supply conditions from the MAU and the OA delivery scheme. Air was assumed to be conditioned only slightly by the MAU to prevent workers in the kitchen from being either very cold or warm, and was assumed to be humid enough to sweat. If it were assumed that the kitchen were maintained with as tight a control as the Sales Zone, the reduction in kitchen exhaust would certainly have a greater effect. Similarly, if all OA were brought in through the Main RTU and conditioned as was necessary to maintain desired conditions in the refrigerated section, reduction of exhaust would also have a greater effect.

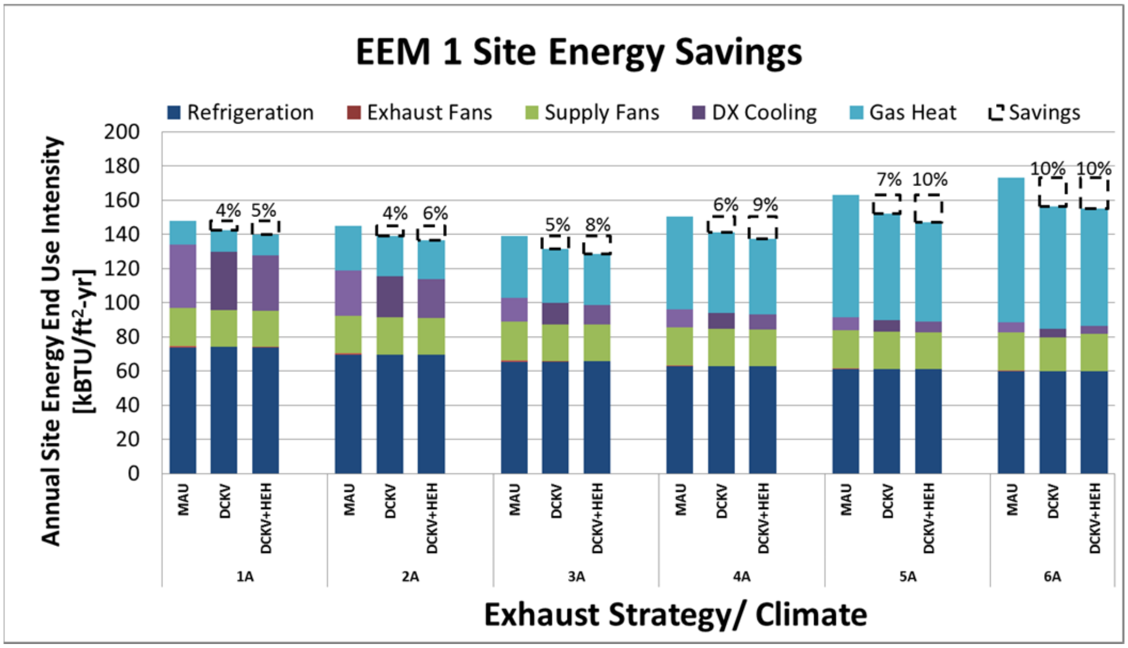

Figure 12. Site energy savings calculated with reduced exhaust strategies

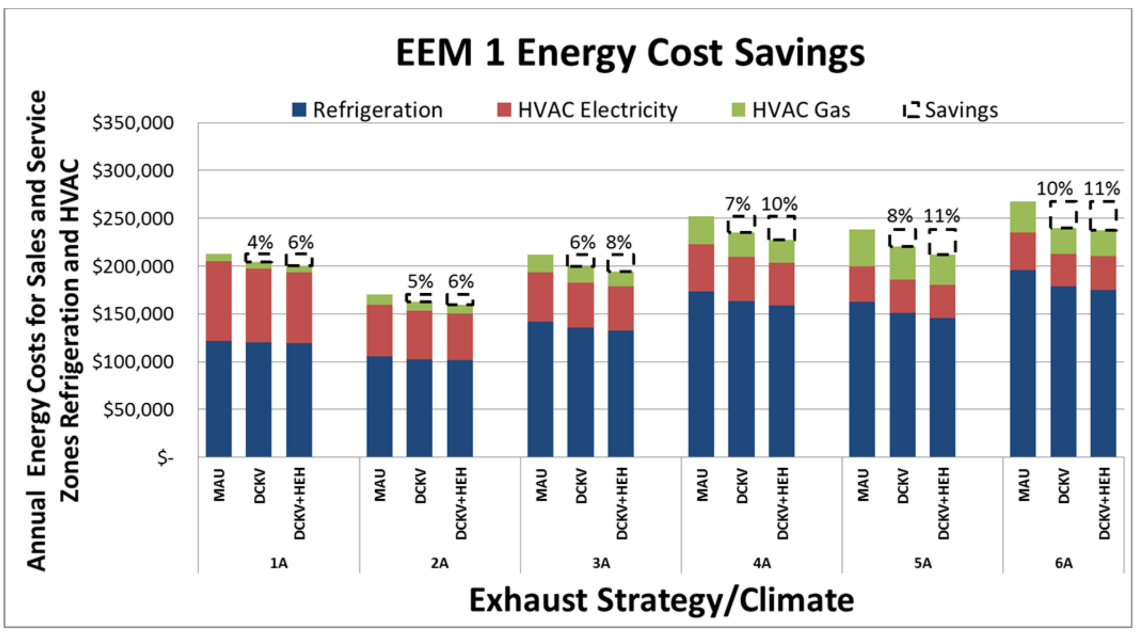

Figure 13. Energy cost savings calculated with reduced exhaust strategies

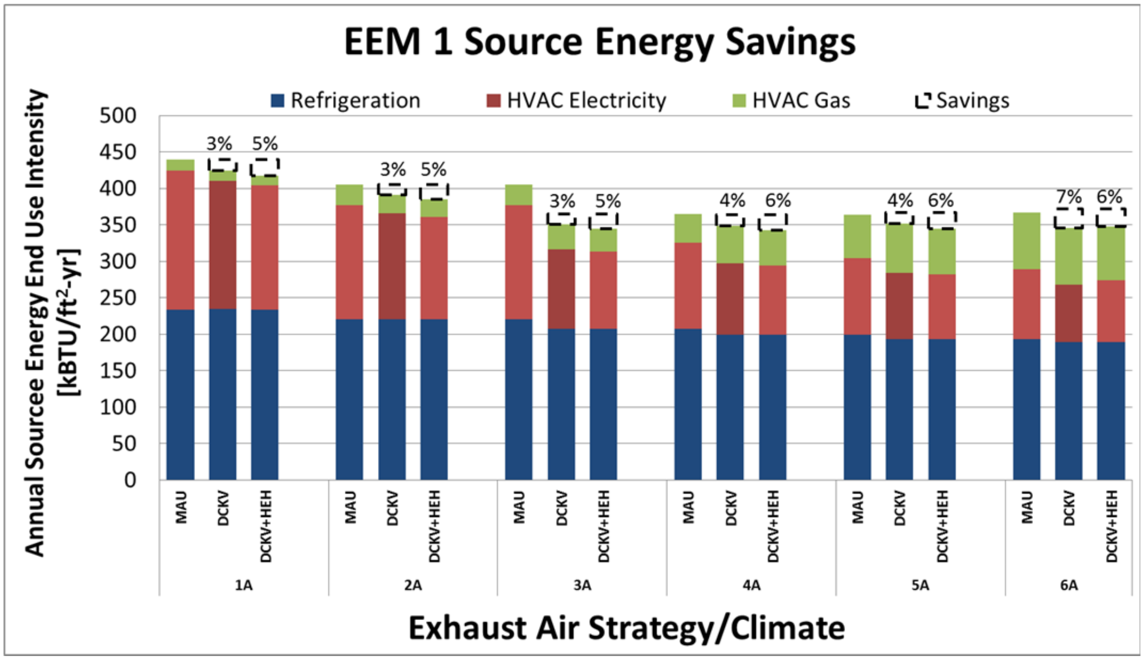

Figure 14. Source energy savings calculated with reduced exhaust strategies

4.3 Energy Efficiency Measure 2: Outdoor Air Delivery Strategies

The second EEM we investigated was the method of conditioning OA delivered to the Sales Zone. We compare four alternative DX-based strategies in Figures 15-17. These are the baseline "MAU," "Pre-Treat," "DOAS," and "interior dehumidifier" strategies described in Chapter 3. The baseline against which all strategies are compared is the MAU scenario. Figures 15–17 show significant savings in hotter climates with the advanced OA strategies. All three of the alternative OA conditioning strategies studied in this EEM eliminate the need for the main RTU to do much, if any, dehumidification. This allows for a greater airflow per ton ratio on the main RTU cooling coil and thus a higher evaporator temperature and more efficient operation. It also eliminates the need to dehumidify any recirculation air (only OA is dehumidified), as is done in the baseline MAU case. The Pre-Treat and DOAS strategies allow a much smaller airflow through the mixed air RTU, because an OA fraction of 0.15 into the RTU evaporator coil does not need to be maintained. This significantly reduces fan power.

The DX Pre-Treat strategy is most effective in hotter climates where very little gas heating is required (25% combined HVAC and refrigeration energy and 47% HVAC energy savings in climate zone 1A). The overcooling of the Pre-Treat system is used effectively by the mixed air system into which the pre-treated air is supplied to reduce the amount of cooling performed by the mixed air system. Refrigeration energy is reduced slightly by the colder and drier air provided with this strategy. In colder climates, this strategy is less effective, because overcooling must be counteracted by additional gas heat.

Similar to the Pre-Treat strategy, the DX DOAS strategy is most effective in warmer climates (17% combined HVAC and refrigeration energy and 32% HVAC energy savings in climate zone 1A). The DOAS strategy was shown to use the least fan power of the four strategies investigated.

The DOAS strategy still showed some overall benefit in colder climates, especially in terms of source energy, though the savings weren't as great as in warmer climates. This is due to the reduction in fan power and increase in gas use to counteract the overcooling for dehumidification.

Another effective strategy is the use of an interior DX dehumidifier in conjunction with an RTU focused on meeting the sensible loads in the space at 450 cfm/ton, which brings in all OA. This resulted in significant energy savings and savings in energy costs in warmer climates (14% combined HVAC and refrigeration energy and 33% HVAC energy savings in climate zone 1A). This is due to the fact that the dehumidification is being localized to only the area that needs it, eliminating the need to dehumidify large quantities of air as in the baseline case. The interior dehumidifier strategy usually results in the greatest refrigeration energy use, owing to higher space DBTs resulting from the full condenser heat being dissipated in the interior environment. The use of this type of strategy in supermarkets is relatively new and can benefit from additional design work, possibly including external heat rejection.

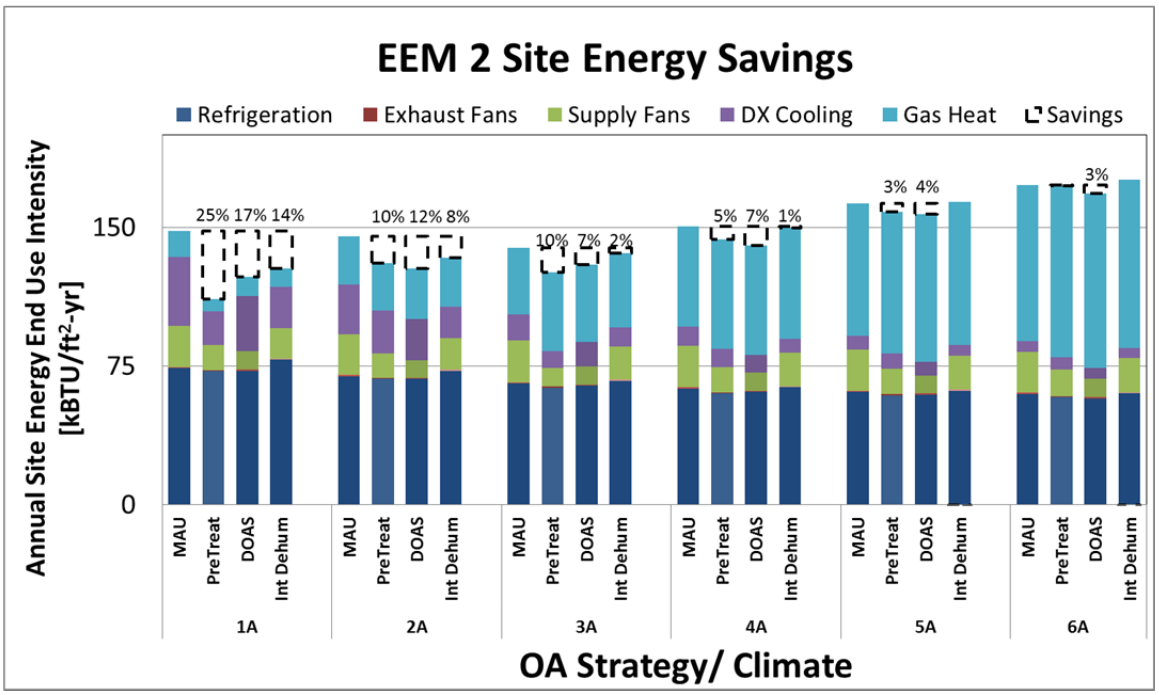

Figure 15. Site energy savings with four means of conditioning OA supplied to Sales Zone

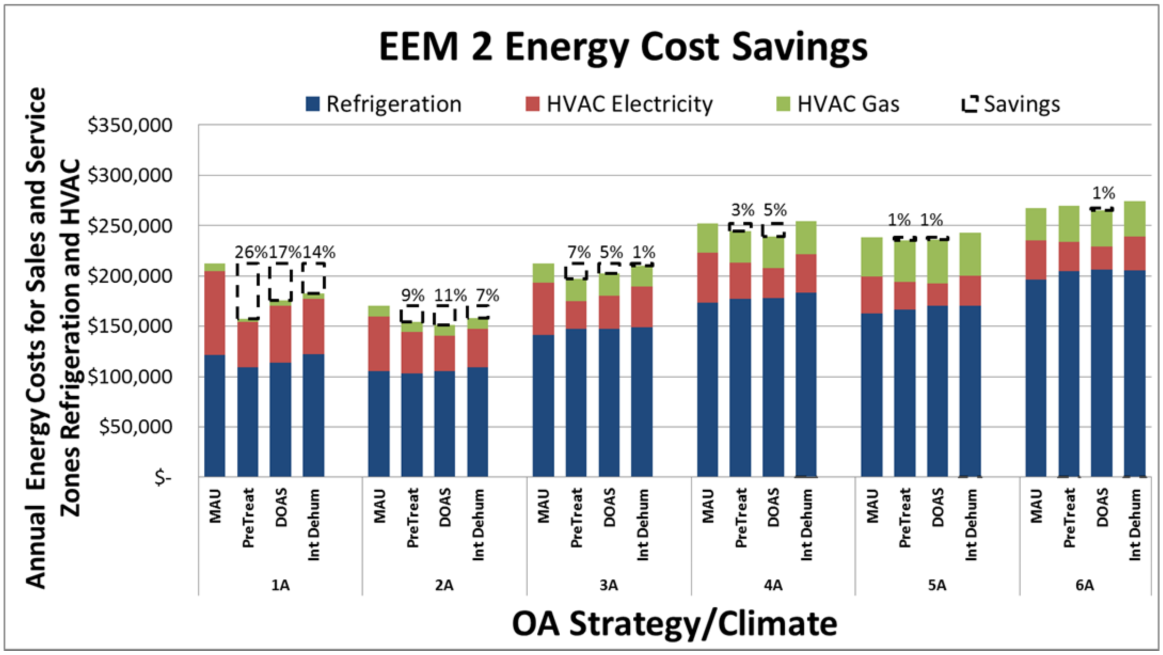

Figure 16. Energy cost savings with four means of conditioning OA supplied to Sales Zone

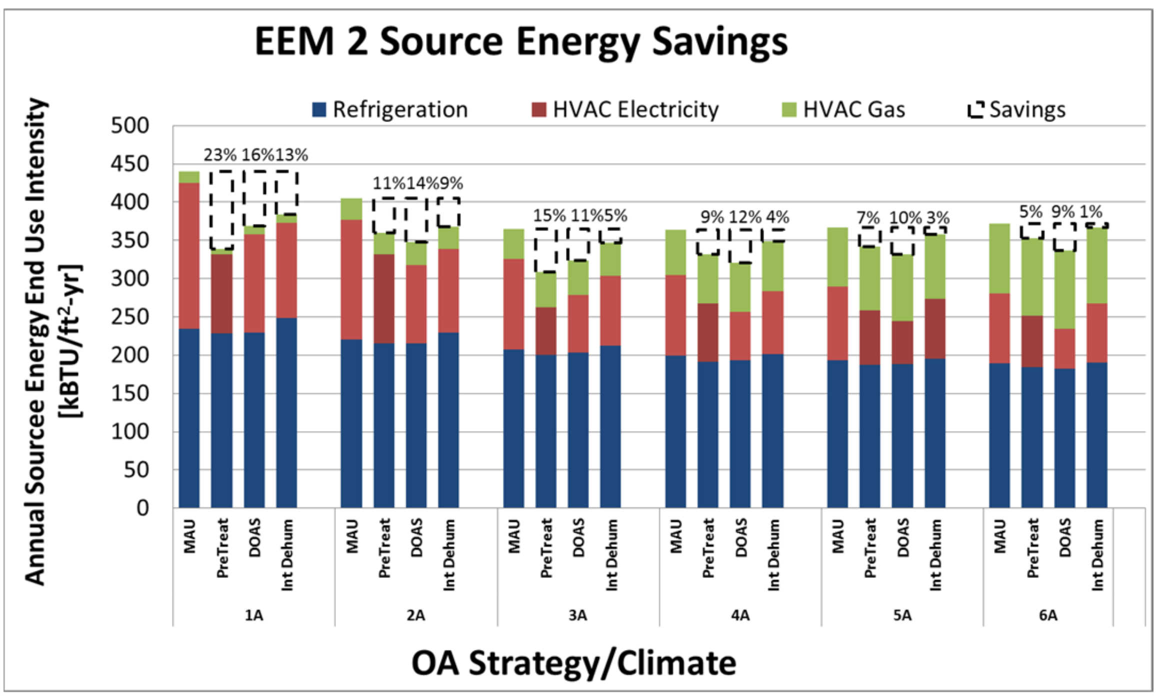

Figure 17. Source energy savings with four means of conditioning OA supplied to Sales Zone

4.4 Energy Efficiency Measure 3: Improved Dehumidification Systems

The final EEM investigated was the use of advanced dehumidification HVAC systems to provide for the conditioning needs of supermarkets. Large energy and costs savings were calculated with several of the systems. For the Service and Sales Zones, maximum total refrigeration and HVAC site energy savings of 31%–35% across all climates studied were calculated, as well as 30%–36% combined refrigeration and HVAC energy cost savings, 49%–61% HVAC site energy savings, and 56%–62% HVAC cost savings. The results of these simulations are presented in Figures 18, 19, and 20.

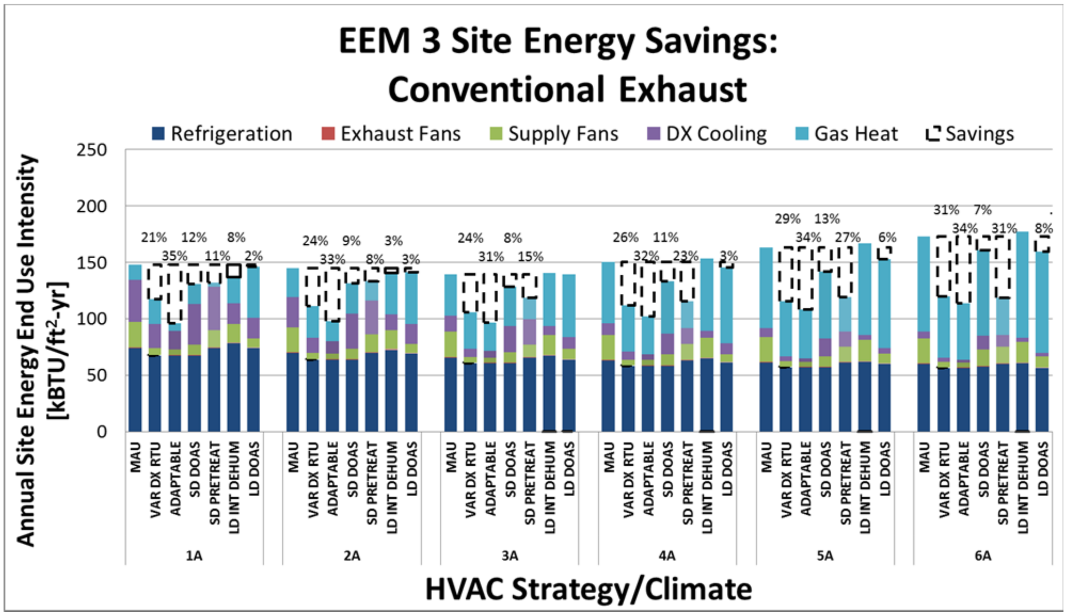

Figure 18. Site energy savings with advanced dehumidification systems

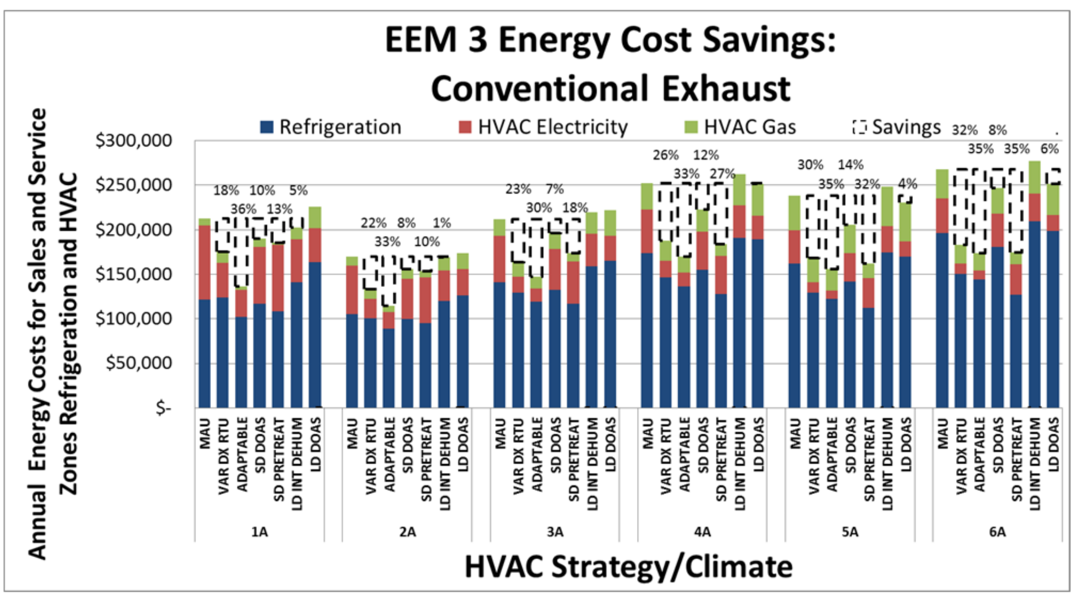

Figure 19. Energy cost savings with advanced dehumidification systems

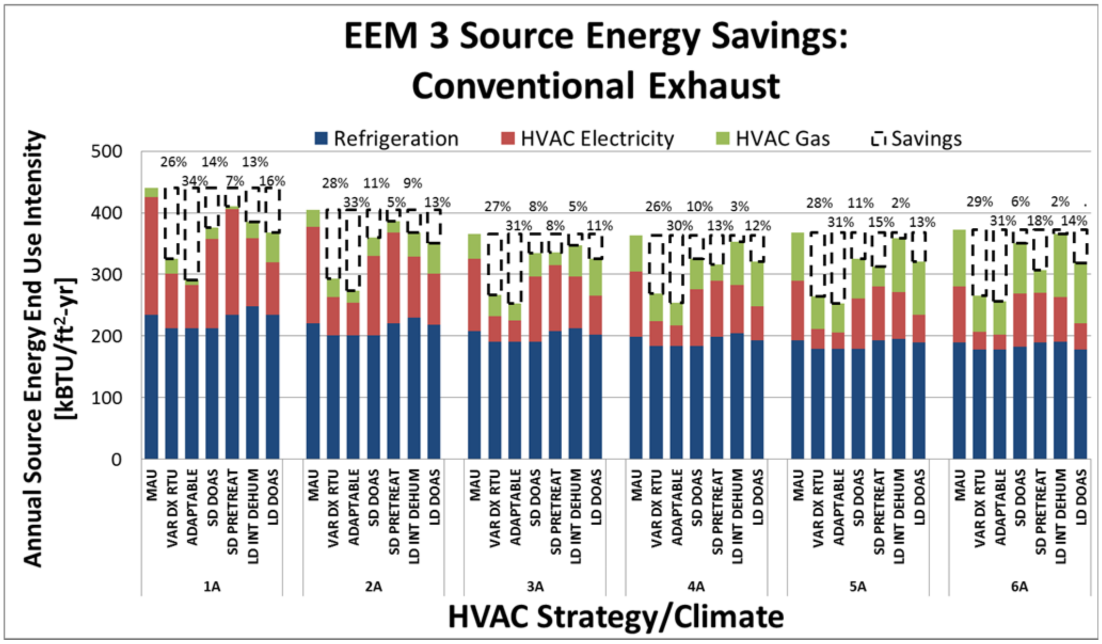

Figure 20. Source energy savings with advanced dehumidification systems

In general, the Adaptive Multi-Path System demonstrated substantial savings as measured by site energy use (31%–35% savings), source energy use (30%–34%), and energy cost (30%–36%) across all climates. This conclusion is not surprising because the Adaptive Multi-Path System integrates the best technologies of several of the other HVAC systems. It also adds additional features, such as optimal mixing of both the mixed air stream and the supply air stream, while providing constant supply volume. The Adaptive Multi-Path System showed large reductions in fan power, owing to its ability to provide necessary cooling with the minimum constant airflow (0.5 cfm/ft^2). It also reduced DX Cooling energy significantly due to the capacity modulation. Gas heat was also reduced via the reduction of overcooling afforded by capacity modulation and optimized mixing.

The Variable Capacity DX System (VAR DX RTU) showed similar trends to the Adaptive Multi-Path System, although it decidedly performed best in colder climates, likely due to the fact that it did not include the optimized mixing of RA and OA. The VAR DX RTU system, however, demonstrated substantial savings in all climates (21%–31% site energy savings, 18%–32% energy cost savings, and 26%–29% source energy savings).

The SD system performed well in all climates, with its best performance demonstrated in the Pre-Treat configuration in colder climates. Performance as a DOAS system was similar to the performance of the DOAS DX system in warmer climates but much better in colder climates (7%–13% site energy savings, 7%–14% energy cost savings, and 6%–14% source energy savings across all six climates). Unlike the DX-only systems studied, the SD system does not require overcooling to dehumidify air and, therefore, less gas is required to heat the space to the SP. This results in an advantage for the SD system, especially in colder humid climates. The SD system demonstrated its greatest savings in Pre-Treat configuration, as the commercial product modeled is most often specified. In this configuration, it offers the benefits of the DOAS system and decreases the mixed air RTU cooling energy by reducing the OA moisture load on the mixed air system and allowing for a larger airflow per ton to be specified; thus, it offers more efficient operation. The SD system in Pre-Treat configuration showed site energy, energy cost, and source energy savings of 11%–31%, 10%–35%, and 5%–18%, respectively, across the climates modeled. It also showed the least, or near-to-the-least refrigeration energy use of any system across climates.

In general, LD systems, both in the DOAS and interior dehumidifier configurations, saved source energy through the shifting of dehumidification energy use from electricity to gas (11%–16% as DOAS and 2%–13% as interior dehumidifier). Gas use is much greater with the LD systems, not because of space heating, but because of the gas required to regenerate the desiccant. However, in the final analysis, the LD systems saved little site energy (0%–8% as DOAS, negative to 8% as Interior Dehumidifier). Overall, energy cost savings were not substantial. The particular LD system modeled contains a single-effect regenerator, which is nearly obsolete at the time of writing. However, a model for the more efficient double-effect regenerator was not available for this study. Preliminary tests show that an LD system with a double-effect regenerator will have a thermal COP of 1.05–1.15, while a single-effect regenerator, as used in this project, produces a thermal COP of 0.75–0.8 at a 25ºC reference temperature.

It should also be noted that the systems modeled did not maintain identical space conditions, although conditions were always within the SP ranges specified in Appendix A. On average, the interior dehumidifier systems maintained a DBT in the refrigerated section of about 66ºF–68ºF DBT, while the DOAS systems required gas heat in most cases to maintain the refrigerated section DBT above the SP of 60ºF DBT. This means that the interior dehumidifiers provided better thermal comfort, which is not captured in cost and energy metrics, but did so with an associated refrigeration energy penalty.

4.5 Energy Efficiency Measure 1 + Energy Efficiency Measure 3: Improved Dehumidification Systems With Reduced Exhaust

We conducted a full set of simulations combining each of the best performing dehumidification systems with reduced exhaust requirements afforded by a DCKV strategy in conjunction with energy-efficient kitchen hoods. In isolation, each of these EEMs provided significant savings. However, the incremental benefit of adding DCKV or HEHs on top of an advanced HVAC system was shown not to be significant in any of the climates for any of the systems. This is likely due to the fact that the only variable strongly affected by the reduced exhaust rates is the MAU energy consumption, which is a smaller contributor to the overall HVAC energy.

4.6 Price Points for Desired Payback Periods

With the savings demonstrated in the previous sections, building owners and designers should be able to make more informed decisions on HVAC equipment selection and design. However, it was not possible to obtain from manufacturers the cost numbers needed to calculate traditional metrics like net present value or even simple payback time. Equipment costs are considered proprietary and are likely to vary widely by geographical region and purchasing power of the customer. Installation costs are difficult to estimate because in some cases the roof may require structural work, and the cost of ductwork, which is difficult to estimate, must be included. Lastly, some systems like LDAC are currently not mass-produced, so that costs for a commercial product do not exist.

In conversations with building owners, we determined that 3- and 5-year horizons were typical simple payback hurdles used in the industry to screen new technologies. This criterion allowed us to back-calculate the incremental first costs (equipment and installation) each technology would have to meet, compared to the baseline system, to clear these hurdles based on calculated energy savings versus the baseline scenario. Building owners and design teams can consult these numbers when procuring an HVAC system for their supermarkets to understand whether cost quotes for equipment and installation are likely to be consistent with a 3‒5 year simple payback. However, please note that the numbers presented below are intended to serve as rough guidelines rather than hard-and-fast rules. Energy savings should be modeled on a case-by-case basis, based on the details of the specific store in question, to achieve more accurate results.

With this in mind, the last set of results we now present is the calculated maximum incremental cost each system would require if it were to have either a 3-year or 5-year simple payback period in each climate modeled. If the total equipment and installation cost falls below the numbers included in Table 5, it is an indication of a favorable return on investment. The results labeled "MAU" are the incremental cost of adding an MAU over the mixed air RTU (MAS-b) configuration to hit the 3- and 5-year payback thresholds. All other figures indicate the incremental costs necessary to reach 3- and 5-year payback thresholds compared to the MAU scenario. As a reminder to the reader, the MAU scenario brings 6500 cfm of OA into the Sales Zone via a DX MAS and the remaining portion of OA necessary through the MAU in the Service Zone. This was determined to be the most common means of treatment of OA, per consultation with supermarket designers. The MAU scenario is the baseline against which all EEMs are compared.

Table 5. Price Points for 3- and 5-Year Payback Periods for Climate Zones 1A-3A

Incremental Cost of System for 3- and 5-year Simple Payback Period in Climate Zone 1A

MAU		MAU					
	3 Yr	$ 298,067.42					
	5Yr	$ 496,779.04					
EEM 1		DCKV	DCKV+HEH				
	3 Yr	$ 24,711.97	$ 35,523.42				
	5Yr	$ 41,186.62	$ 59,205.70				
EEM 2		PreTreat	DOAS	Int Dehum			
	3 Yr	$ 109,510.91	$ 90,285.61	$ 164,529.33			
	5Yr	$ 182,518.18	$ 150,476.02	$ 274,215.55			
EEM 3		VAR DX RTU	ADAPTABLE	SD DOAS	SD PRETREAT	LD INT DEHUM	LD DOAS
	3 Yr	$ 112,496.09	$ 227,634.38	$ 66,898.33	$ 81,767.63	$ 30,440.26	$ (41,090.55)
	5Yr	$ 187,493.49	$ 379,390.63	$ 111,497.22	$ 136,279.38	$ 50,733.76	$ (68,484.24)

Incremental Cost of System for 3- and 5-year Simple Payback Period in Climate Zone 2A

MAU		MAU					
	3 Yr	$ 177,198.47					
	5Yr	$ 295,330.79					
EEM 1		DCKV	DCKV+HEH				
	3 Yr	$ 23,019.43	$ 32,647.13				
	5Yr	$ 38,365.72	$ 54,411.88				
EEM 2		PreTreat	DOAS	Int Dehum			
	3 Yr	$ 47,490.24	$ 56,474.67	$ 36,363.43			
	5Yr	$ 79,150.39	$ 94,124.45	$ 60,605.71			
EEM 3		VAR DX RTU	ADAPTABLE	SD DOAS	SD PRETREAT	LD INT DEHUM	LD DOAS
	3 Yr	$ 110,650.68	$ 166,659.59	$ 42,733.79	$ 50,320.32	$ 3,597.28	$ (10,801.18)
	5Yr	$ 184,417.80	$ 277,765.98	$ 71,222.99	$ 83,867.21	$ 5,995.46	$ (18,001.97)

Incremental Cost of System for 3- and 5-year Simple Payback Period in Climate Zone 3A

MAU		MAU					
	3 Yr	$ 151,444.39					
	5Yr	$ 252,407.32					
EEM 1		DCKV	DCKV+HEH				
	3 Yr	$ 37,562.95	$ 53,660.40				
	5Yr	$ 62,604.92	$ 89,434.01				
EEM 2		PreTreat	DOAS	Int Dehum			
	3 Yr	$ 44,746.62	$ 29,778.84	$ 6,163.19			
	5Yr	$ 74,577.70	$ 49,631.40	$ 10,271.99			
EEM 3		VAR DX RTU	ADAPTABLE	SD DOAS	SD PRETREAT	LD INT DEHUM	LD DOAS
	3 Yr	$ 144,174.86	$ 193,586.51	$ 46,388.42	$ 113,524.87	$ (23,390.61)	$ (29,681.88)
	5Yr	$ 240,291.43	$ 322,644.18	$ 77,314.04	$ 189,208.12	$ (38,984.36)	$ (49,469.80)

Table 6. Price Points for 3- and 5-Year Payback Periods for Climate Zones 4A-6A

Incremental Cost of System for 3- and 5-year Simple Payback Period in Climate Zone 4A

System	Period						
MAU		MAU					
	3 Yr	$ 128,100.02					
	5Yr	$ 213,500.04					
EEM 1		DCKV	DCKV+HEH				
	3 Yr	$ 52,344.56	$ 75,190.85				
	5Yr	$ 87,240.93	$ 125,318.09				
EEM 2		PreTreat	DOAS	Int Dehum			
	3 Yr	$ 22,596.58	$ 38,644.36	$ (5,316.94)			
	5Yr	$ 37,660.97	$ 64,407.26	$ (8,861.56)			
EEM 3		VAR DX RTU	ADAPTABLE	SD DOAS	SD PRETREAT	LD INT DEHUM	LD DOAS
	3 Yr	$ 194,267.23	$ 247,446.76	$ 88,434.49	$ 206,225.71	$ (29,472.04)	$ 2,136.51
	5Yr	$ 323,778.72	$ 412,411.26	$ 147,390.82	$ 343,709.51	$ (49,120.06)	$ 3,560.85

Incremental Cost of System for 3- and 5-year Simple Payback Period in Climate Zone 5A

System	Period						
MAU		MAU					
	3 Yr	$ 83,450.66					
	5Yr	$ 139,084.43					
EEM 1		DCKV	DCKV+HEH				
	3 Yr	$ 55,487.96	$ 80,736.47				
	5Yr	$ 92,479.93	$ 134,560.78				
EEM 2		PreTreat	DOAS	Int Dehum			
	3 Yr	$ 9,391.79	$ 7,666.99	$ (12,290.98)			
	5Yr	$ 15,652.98	$ 12,778.31	$ (20,484.97)			
EEM 3		VAR DX RTU	ADAPTABLE	SD DOAS	SD PRETREAT	LD INT DEHUM	LD DOAS
	3 Yr	$ 211,398.88	$ 249,090.33	$ 98,092.87	$ 228,876.82	$ (29,231.45)	$ 25,177.55
	5Yr	$ 352,331.47	$ 415,150.55	$ 163,488.12	$ 381,461.37	$ (48,719.08)	$ 41,962.58

Incremental Cost of System for 3- and 5-year Simple Payback Period in Climate Zone 6A

System	Period						
MAU		MAU					
	3 Yr	$ 89,750.60					
	5Yr	$ 149,584.34					
EEM 1		DCKV	DCKV+HEH				
	3 Yr	$ 82,623.04	$ 91,203.04				
	5Yr	$ 137,705.06	$ 152,005.07				
EEM 2		PreTreat	DOAS	Int Dehum			
	3 Yr	$ (6,777.41)	$ 7,177.65	$ (18,979.24)			
	5Yr	$ (11,295.68)	$ 11,962.74	$ (31,632.06)			
EEM 3		VAR DX RTU	ADAPTABLE	SD DOAS	SD PRETREAT	LD INT DEHUM	LD DOAS
	3 Yr	$ 252,845.81	$ 281,629.24	$ 61,679.80	$ 280,026.61	$ (28,470.38)	$ 48,336.57
	5Yr	$ 421,409.69	$ 469,382.06	$ 102,799.66	$ 466,711.01	$ (47,450.63)	$ 80,560.95

4.7 Conclusion

In this study, we conducted a total of 138 detailed simulations that investigated a variety of means for providing for the HVAC needs of typical supermarkets in climates in the United States with high humidity loads. The study included detailed evaluation of interactions between advanced HVAC systems and other systems present in supermarkets.

The study demonstrated that large energy and cost savings are available with advanced HVAC systems in supermarkets. The best-performing solutions were calculated to operate with more than 50% HVAC energy savings across the humid climates in the United States, specifically:

- Preliminary investigations showed that inclusion of an MAU is highly recommended in all climates studied. Simulations showed a 13%–46% savings in HVAC site energy for the Sales and Service Zones in the climates studied and a 33%–46% savings in HVAC energy costs.
- EEM 1 investigations showed that inclusion of a DCKV system in conjunction with HEHs can result in 11%–16% savings over the baseline MAU strategy in HVAC site energy for the Sales and Service Zones in the climates studied and 11%–13% savings in HVAC energy costs.
- EEM 2 investigations concluded that advanced OA delivery strategies, including Pre-Treat, DOAS, and interior dehumidifiers, provided significant benefit in the warmest climates, but not in cooler climates. Savings over the baseline MAU strategy of up to 25% in combined refrigeration and HVAC site energy were calculated for the warmest climate studied, as well as savings of up to 47% of HVAC site energy and energy costs for the best-performing systems.
- EEM 3 investigations demonstrated that even greater savings across all climates are available with advanced HVAC systems focused on dehumidification. For the Service and Sales Zones, total refrigeration and HVAC site energy savings of 31%–35% across all climates studied were calculated, as well as 30%–36% combined refrigeration and HVAC energy cost savings, 49%–61% HVAC site energy savings, and 56%–62% HVAC cost savings.

We modeled other systems that provide varied savings with different associated initial costs. A detailed procedure for conducting similar simulations is included in this report. We hope that the information provided herein will allow for more informed decisions for HVAC design and selection in supermarkets and expand the frontiers of supermarket energy modeling.

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Appendix A. Modeling Methodology and Validation

We modeled several different air conditioning/dehumidification strategies for the purpose of this study and performed full-building simulations for each of these strategies over the course of a Typical Meteorological Year. We conducted these simulations using DOE's EnergyPlus building simulation software. We also expanded and enhanced the capabilities of EnergyPlus by co-simulating EnergyPlus with models of advanced systems that were not included in the EnergyPlus libraries. We wrote these models in the Modelica modeling language and performed simulations with the Dymola software package. The following paragraphs describe the modeling methods and assumptions used for each of the simulations.

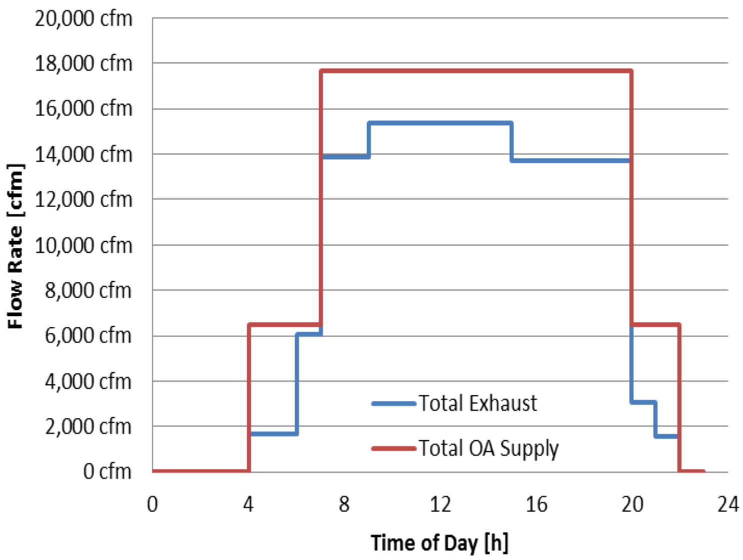
A.1 Energy Efficiency Measure Assumptions

A.1.1 Exhaust and Outdoor Airflow Schedules

The assumptions used for exhaust and OA supply schedules have significant effects on simulated energy use in supermarkets. In general, we chose exhaust requirements, with input from several industry practitioners, to approximate the requirements of a newly constructed or updated supermarket. This results in exhaust flow rates that are equivalent to existing supermarkets with large prepared food areas, but may be greater than those in some existing building stock. We made this decision in light of the trend towards larger kitchen/service areas and the coincident increase in exhaust requirements in supermarkets being built today. Therefore, the results should be valid for a greater period of time in the future.

We modeled three different exhaust/ventilation strategies to understand the effect of advanced exhaust strategies on building energy use. In all cases, the make-up air required to replace exhaust flows determined the amount of total OA that needed to be introduced into the space, rather than the ventilation requirements. With the exception of the small restroom exhaust flows, we assumed that all the exhaust exits through the exhaust hoods in the Service Zone. In all cases, total OA supply rates are equal to 1.15 times the total exhaust rate, to account for building pressurization. The first case is a baseline strategy in which a main exhaust fan in the service area operates at only one or possibly two speeds and is not controlled by sensors on the kitchen equipment. The second strategy is a DCKV strategy, in which sensors recognize operation of kitchen equipment and actuate exhaust fans, allowing exhaust needs and exhaust rates to more closely match. In the final strategy, a DCKV strategy is used in conjunction with HEHs that reduce overall exhaust needs. The OA delivery schedules for these three situations are shown in Figures A1–A3.

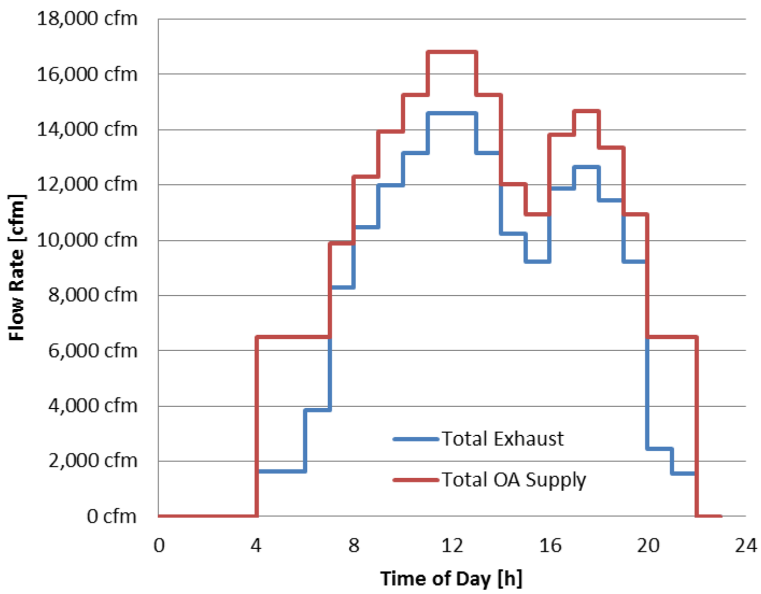
A.1.1.1 Baseline



Time of Day [h]	Ventilation Air	MakeupAir	Total Exhaust	Total OA
0	0 cfm	0 cfm	0 cfm	0 cfm
1	0 cfm	0 cfm	0 cfm	0 cfm
2	0 cfm	0 cfm	0 cfm	0 cfm
3	0 cfm	0 cfm	0 cfm	0 cfm
4	0 cfm	0 cfm	0 cfm	0 cfm
4	6,500 cfm	0 cfm	1,641 cfm	6,500 cfm
5	6,500 cfm	0 cfm	1,641 cfm	6,500 cfm
6	6,500 cfm	0 cfm	1,641 cfm	6,500 cfm
6	6,500 cfm	0 cfm	6,042 cfm	6,500 cfm
7	6,500 cfm	0 cfm	6,042 cfm	6,500 cfm
7	6,500 cfm	11,184 cfm	13,878 cfm	17,684 cfm
8	6,500 cfm	11,184 cfm	13,878 cfm	17,684 cfm
9	6,500 cfm	11,184 cfm	13,878 cfm	17,684 cfm
9	6,500 cfm	11,184 cfm	15,376 cfm	17,684 cfm
10	6,500 cfm	11,184 cfm	15,376 cfm	17,684 cfm
11	6,500 cfm	11,184 cfm	15,376 cfm	17,684 cfm
12	6,500 cfm	11,184 cfm	15,376 cfm	17,684 cfm
13	6,500 cfm	11,184 cfm	15,376 cfm	17,684 cfm
14	6,500 cfm	11,184 cfm	15,376 cfm	17,684 cfm
15	6,500 cfm	11,184 cfm	15,376 cfm	17,684 cfm
15	6,500 cfm	11,184 cfm	13,735 cfm	17,684 cfm
16	6,500 cfm	11,184 cfm	13,735 cfm	17,684 cfm
17	6,500 cfm	11,184 cfm	13,735 cfm	17,684 cfm
18	6,500 cfm	11,184 cfm	13,735 cfm	17,684 cfm
19	6,500 cfm	11,184 cfm	13,735 cfm	17,684 cfm
20	6,500 cfm	11,184 cfm	13,735 cfm	17,684 cfm
20	6,500 cfm	0 cfm	3,058 cfm	6,500 cfm
21	6,500 cfm	0 cfm	3,058 cfm	6,500 cfm
21	6,500 cfm	0 cfm	1,560 cfm	6,500 cfm
22	6,500 cfm	0 cfm	1,560 cfm	6,500 cfm
22	0 cfm	0 cfm	0 cfm	0 cfm
23	0 cfm	0 cfm	0 cfm	0 cfm

Figure A1. Exhaust and OA supply schedules for baseline operation

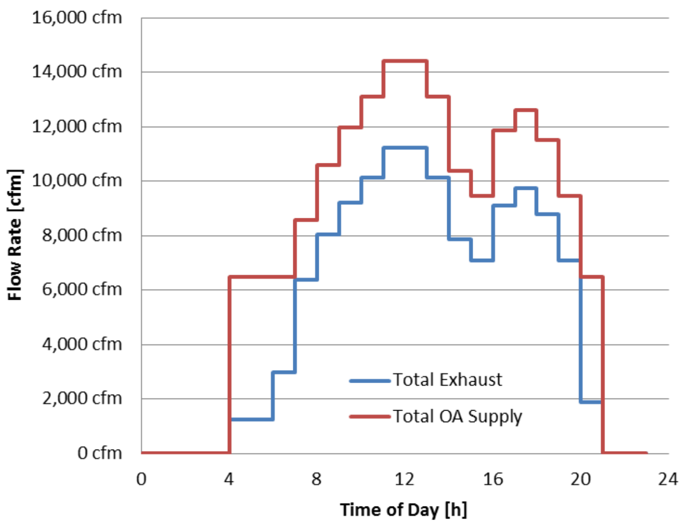
A.1.1.2 DCKV



Time of Day [h]	Ventilation Air	Makeup Air	Total Exhaust	Total OA Supply
0	0 cfm	0 cfm	0 cfm	0 cfm
1	0 cfm	0 cfm	0 cfm	0 cfm
2	0 cfm	0 cfm	0 cfm	0 cfm
3	0 cfm	0 cfm	0 cfm	0 cfm
4	0 cfm	0 cfm	0 cfm	0 cfm
4	6,500 cfm	0 cfm	1,641 cfm	6,500 cfm
5	6,500 cfm	0 cfm	1,641 cfm	6,500 cfm
6	6,500 cfm	0 cfm	1,641 cfm	6,500 cfm
6	6,500 cfm	0 cfm	3,862 cfm	6,500 cfm
7	6,500 cfm	0 cfm	3,862 cfm	6,500 cfm
7	6,500 cfm	3,391 cfm	8,292 cfm	9,891 cfm
8	6,500 cfm	3,391 cfm	8,292 cfm	9,891 cfm
8	6,500 cfm	5,790 cfm	10,472 cfm	12,290 cfm
9	6,500 cfm	5,790 cfm	10,472 cfm	12,290 cfm
9	6,500 cfm	7,437 cfm	11,970 cfm	13,937 cfm
10	6,500 cfm	7,437 cfm	11,970 cfm	13,937 cfm
10	6,500 cfm	8,759 cfm	13,172 cfm	15,259 cfm
11	6,500 cfm	8,759 cfm	13,172 cfm	15,259 cfm
11	6,500 cfm	10,308 cfm	14,580 cfm	16,808 cfm
12	6,500 cfm	10,308 cfm	14,580 cfm	16,808 cfm
13	6,500 cfm	10,308 cfm	14,580 cfm	16,808 cfm
13	6,500 cfm	8,759 cfm	13,172 cfm	15,259 cfm
14	6,500 cfm	8,759 cfm	13,172 cfm	15,259 cfm
14	6,500 cfm	5,536 cfm	10,242 cfm	12,036 cfm
15	6,500 cfm	5,536 cfm	10,242 cfm	12,036 cfm
15	6,500 cfm	4,431 cfm	9,237 cfm	10,931 cfm
16	6,500 cfm	4,431 cfm	9,237 cfm	10,931 cfm
16	6,500 cfm	7,306 cfm	11,851 cfm	13,806 cfm
17	6,500 cfm	7,306 cfm	11,851 cfm	13,806 cfm
17	6,500 cfm	8,182 cfm	12,647 cfm	14,682 cfm
18	6,500 cfm	8,182 cfm	12,647 cfm	14,682 cfm
18	6,500 cfm	6,860 cfm	11,445 cfm	13,360 cfm
19	6,500 cfm	6,860 cfm	11,445 cfm	13,360 cfm
19	6,500 cfm	4,431 cfm	9,237 cfm	10,931 cfm
20	6,500 cfm	4,431 cfm	9,237 cfm	10,931 cfm
20	6,500 cfm	0 cfm	2,436 cfm	6,500 cfm
21	6,500 cfm	0 cfm	2,436 cfm	6,500 cfm
21	6,500 cfm	0 cfm	1,560 cfm	6,500 cfm
22	6,500 cfm	0 cfm	1,560 cfm	6,500 cfm
22	0 cfm	0 cfm	0 cfm	0 cfm
23	0 cfm	0 cfm	0 cfm	0 cfm

Figure A2. Exhaust and OA supply schedules for DCKV schedule

A.1.1.3 Demand Controlled Kitchen Ventilation and High-Efficiency Hoods



Time of Day [h]	Ventilation Air	MakeupAir	Total Exhaust	Total OA Supply
0	0 cfm	0 cfm	0 cfm	0 cfm
1	0 cfm	0 cfm	0 cfm	0 cfm
2	0 cfm	0 cfm	0 cfm	0 cfm
3	0 cfm	0 cfm	0 cfm	0 cfm
4	0 cfm	0 cfm	0 cfm	0 cfm
4	4,123 cfm	2,377 cfm	1,262 cfm	6,500 cfm
5	4,123 cfm	2,377 cfm	1,262 cfm	6,500 cfm
6	4,123 cfm	2,377 cfm	1,262 cfm	6,500 cfm
6	4,123 cfm	2,377 cfm	2,970 cfm	6,500 cfm
7	4,123 cfm	2,377 cfm	2,970 cfm	6,500 cfm
7	4,123 cfm	4,440 cfm	6,377 cfm	8,563 cfm
8	4,123 cfm	4,440 cfm	6,377 cfm	8,563 cfm
8	4,123 cfm	6,469 cfm	8,054 cfm	10,592 cfm
9	4,123 cfm	6,469 cfm	8,054 cfm	10,592 cfm
9	4,123 cfm	7,863 cfm	9,206 cfm	11,986 cfm
10	4,123 cfm	7,863 cfm	9,206 cfm	11,986 cfm
10	4,123 cfm	8,981 cfm	10,130 cfm	13,104 cfm
11	4,123 cfm	8,981 cfm	10,130 cfm	13,104 cfm
11	4,123 cfm	10,292 cfm	11,213 cfm	14,415 cfm
12	4,123 cfm	10,292 cfm	11,213 cfm	14,415 cfm
13	4,123 cfm	10,292 cfm	11,213 cfm	14,415 cfm
13	4,123 cfm	8,981 cfm	10,130 cfm	13,104 cfm
14	4,123 cfm	8,981 cfm	10,130 cfm	13,104 cfm
14	4,123 cfm	6,255 cfm	7,877 cfm	10,378 cfm
15	4,123 cfm	6,255 cfm	7,877 cfm	10,378 cfm
15	4,123 cfm	5,320 cfm	7,104 cfm	9,443 cfm
16	4,123 cfm	5,320 cfm	7,104 cfm	9,443 cfm
16	4,123 cfm	7,752 cfm	9,114 cfm	11,875 cfm
17	4,123 cfm	7,752 cfm	9,114 cfm	11,875 cfm
17	4,123 cfm	8,492 cfm	9,726 cfm	12,615 cfm
18	4,123 cfm	8,492 cfm	9,726 cfm	12,615 cfm
18	4,123 cfm	7,374 cfm	8,802 cfm	11,497 cfm
19	4,123 cfm	7,374 cfm	8,802 cfm	11,497 cfm
19	4,123 cfm	5,320 cfm	7,104 cfm	9,443 cfm
20	4,123 cfm	5,320 cfm	7,104 cfm	9,443 cfm
20	4,123 cfm	2,377 cfm	1,874 cfm	6,500 cfm
21	4,123 cfm	2,377 cfm	1,874 cfm	6,500 cfm
21	4,123 cfm	2,377 cfm	1,200 cfm	6,500 cfm
21	0 cfm	0 cfm	0 cfm	0 cfm
22	0 cfm	0 cfm	0 cfm	0 cfm
23	0 cfm	0 cfm	0 cfm	0 cfm

Figure A3. Exhaust and OA air supply schedules for DCKV strategy with HEHs

A.1.2 Outdoor Air Distribution/Delivery Schemes

We determined overall OA requirements from exhaust needs as described above. With these requirements, we investigated four different delivery methods, depicted in Figure 1. The assumptions for each strategy are given next.

A.1.2.1 Mixed Air System Delivering All Outdoor Air "Mixed Air Systems"

In this delivery scheme, all OA is introduced to the building through a main mixed air RTU located in the Sales Zone (for modeling purposes, multiple RTUs with similar operation are modeled as one large RTU without loss of accuracy). All supply air, including OA, is distributed to dry-goods and refrigerated sections of the Sales Zone, in proportion to their respective floor areas, via a splitter located downstream of the main RTU. Because all exhaust is exhausted from the kitchen hoods, a cross-mixing object in EnergyPlus is included to move transfer air from the Sales Zone to the Service Zone. Transfer air flow rates are equal to OA flow rates throughout the simulation.

A.1.2.2 Make-Up Air Unit Located in Service Zone "MAU"

In the next scheme, an MAU is located in the Service Zone that responds to the exhaust requirements. A Main RTU in the Sales Zone brings in OA and mixes it with return air; the mixed air is delivered to the dry goods and refrigerated sections in proportion to their floor areas. However, only 6500 cfm (0.2 cfm/ft^2) of OA is delivered during occupied hours and none when the building is unoccupied. This amount provides for all ventilation requirements and some portion of the make-up air. This was determined in consultation with industry professionals to be a typical design and leaves a remaining portion of OA, which can be supplied by widely available MAUs. The Main RTU fan only runs when the building is occupied or if a call for dehumidification, cooling, or heating exists.

The balance of the required OA is introduced into the Service Zone only, through an MAU located in that zone. OA flow rates vary to meet the required supply. The MAU is assumed to have a continuously variable fan that can modulate OA supply rates to counteract exhaust precisely. The MAU delivers air between 55ºF and 65ºF near saturation and the Service Zone temperature is allowed to fluctuate.

A.1.2.3 Outdoor Air Pretreatment "Pre-Treat"

In the Pre-Treat configuration, the OA is cooled and dehumidified prior to being mixed with return air and entering the main RTU. The cooling coil that preconditions OA is controlled based on the humidity of the entering OA and is adjusted to deliver nearly space-neutral conditions into the mixing box. This allows the Main RTU cooling coil to be sized smaller, at 400 cfm/ton for the purposes of this study. Airflow through the main RTU is maintained at 0.5 cfm/ft^2 of Sales Zone floor space. The Main RTU fan runs continuously during occupied hours and at night only when a dehumidification, cooling, or heating call exists in the space.

A.1.2.4 Dedicated Outdoor Air System

In the final OA delivery and conditioning scheme studied, a true DOAS configuration is modeled. All OA required for ventilation (a constant volume of 6500 cfm in these simulations) is introduced into the Refrigeration Zone, via a dedicated OA (DOAS) unit. Two mixing objects are included in this set of simulations. A cross-mixing object moves a volume of transfer air equivalent to the amount of OA from the refrigerated section to the dry goods section. Another object moves an equal amount of air from the Sales Zone to the Service Zone, where it is exhausted.

A.1.3 Heating, Ventilation, and Air-Conditioning Systems

The HVAC systems used in the supermarket are the main object of investigation in this work. The following section describes the various systems that were investigated and the modeling assumptions used for each. A summary of the sizing assumptions and type of models used is given in Table A1.

Table A1. Sizing and Model Information for HVAC Solutions Studied

Letter Designation	*RTU Size*	*RTU Flow Rate*	*OA Brought in Through Sales Zone*	*DOAS/ Pre-Treat Size*	*DOAS/ Pre-Treat Flow Rate*	*RTU Type of Model*	*DOAS or Dehumidifier Type of Model*	*Maturity of Advanced Product*
MAS-a	325 cfm/ton	3.692cfm/sf	ALL OA	NA	NA	EnergyPlus Library	NA	NA
MAS-b	400 cfm/ton	2.72cfm/sf	ALL OA	NA	NA	EnergyPlus Library	NA	NA
MAU	350 cfm/ton	1.33cfm/sf	0.2cfm/sf	NA	NA	EnergyPlus Library	NA	NA
MAU + DCKV	350 cfm/ton	1.33cfm/sf	0.2cfm/sf	NA	NA	EnergyPlus Library	NA	NA
MAU+ DCKV+ HEH	350 cfm/ton	1.33cfm/sf	0.2cfm/sf	NA	NA	EnergyPlus Library	NA	NA
DX PRETREAT*	450 cfm/ton	0.5 cfm/sf	0.2cfm/sf	150cfm/ton	ALL OA	EnergyPlus Library	EnergyPlus Library	Commercially Available
DX DOAS*	450 cfm/ton	0.5cfm/sf	0.2cfm/sf	150cfm/ton	0.65 cfm/SF	EnergyPlus Library	EnergyPlus Library	Commercially Available
DX INT DEHUM*	450 cfm/ton	1.33cfm/sf	0.2cfm/sf	NA	NA	EnergyPlus Library	Empirical from published data measured at NREL	Commercially Available but not Designed for this Application
LD INT DEHUM*	450 cfm/ton	1.33cfm/sf	0.2cfm/sf	NA	NA	EnergyPlus Library	First Principles validated with Empirical Data	In R&D
VAR DX *	271-325 cfm/ton by climate	0.5cfm/sf	0.2cfm/sf	NA	NA	Proprietary Software verified by NREL		Commercially Available
ADAPTABLE*	271-325 cfm/ton by climate	0.5 cfm/sf	0.2 cfm/sf	NA	NA	Proprietary Software verified by NREL		Commercially Available
LD DOAS*	450 cfm/ton	0.5cfm/sf	0.2cfm/sf	150cfm/ton	0.65 cfm/SF	EnergyPlus Library	First Principles validated with Empirical Data	In R&D
SD DOAS*	450 cfm/ton	0.5cfm/sf	0.2cfm/sf	150cfm/ton	0.65 cfm/SF	EnergyPlus Library	Performance Map from Manufacturer	Commercially Available
SD PRETREAT*	450 cfm/ton	0.5 cfm/sf	0.2cfm/sf	Sized by Manufacturer	0.65 cfm/SF	EnergyPlus Library	Performance Map from Manufacturer	Commercially Available

*Information valid for both baseline and DCKV+HEH exhaust rates

A.1.3.1 Mixed Air Systems (MAS-a and MAS-b)

For strategies MAS-a and MAS-b, a single RTU conditions all OA and all recirculated air required to maintain desired space conditions in the Sales Zone. These simulations are conducted entirely in EnergyPlus with a packaged unitary system. Simulation of an identical system was also done in Modelica, in order to gain a good comparison of the two platforms prior to beginning the study. The comparison is shown below in Appendix D. Each system contains a single-stage DX cooling coil, a desuperheater, a gas furnace, and a constant volume supply fan. During occupied hours, a constant volume of air is moved through the main RTU and delivered to the space. The cooling and heating coils cycle on and off as needed in response to thermostat and humidistat signals. During unoccupied times, the RTU supply fan operates only when a heating, cooling, or dehumidification call is received from the space.

The cooling coil models the performance of a commercially available product tested in the Thermal Test Facility at NREL. The coil is controlled based on the space conditions in the Sales Zone in order to deliver desired humidity and cooling SPs. Two different cooling coil sizes are modeled, corresponding to MAS-a and MAS-b.

MAS-a is an unrealistic situation but useful for comparison to other strategies. In MAS-a, a cooling coil and corresponding supply fan is modeled that provides a 50ºF DPT in the refrigerated section throughout the year. This is accomplished by maintaining a recirculation rate so that mixed air with an OA fraction of no greater than 0.15 enters the cooling coil at any time and by specifying a cooling coil with an airflow/capacity ratio of 325 cfm/ton. This results in an unreasonably large system, but allows the system to meet desired humidity SPs at all times throughout the year.

More realistic is MAS-b, which models a packaged unit with a design flow rate so that mixed air entering the cooling coil has an OA fraction no greater than 0.2 and the cooling coil is sized to treat 400 cfm/ton. This results in a 39% smaller system, but it is unable to maintain a 50ºF DPT for several hours in the more humid climates modeled.

The modeled desuperheater delivers a constant amount of heat to the airstream equal to 30% of the sum of the energy removed during cooling and the power input to the compressor. In reality, a control strategy may be devised in which hot gas is diverted when space heating set-points are met so that desuperheat energy is not constantly added to the airstream. However, for the simulations in this report, we determined that this strategy resulted in much higher gas use, because dehumidification periods were often followed immediately by heating periods to bring the space back to the heating SP. During these heating periods, desuperheat is not available because the cooling coil is not in operation and gas must be used. For this reason, we decided to add reheat whenever the cooling coil was in operation. For supermarkets, where most cooling is done for the purposes of dehumidification and lower SHRs are desired, this strategy will be effective in the vast majority of cases.

The heating coil responds to a thermostat in the Sales Zone and delivers a supply air temperature of 90ºF. Heating operation is independent of the cooling coil operation and the two coils are allowed to operate simultaneously. This occasionally occurs when a dehumidification call exists while the space DBT is below the heating SP. The efficiency of the furnace is assumed to be 0.8.

The supply fan for MAS-a and MAS-b is a constant volume fan delivering the design flow rate at all occupied times and during all heating, cooling, and dehumidification calls. The fan efficiency is assumed to be 0.6.

A.1.3.2 Make-Up Air Units (MAU, MAU+DCKV, MAU+DCKV+HEH)

The MAU systems employ a make-up air unit in the Service Zone supplying the difference between the required OA flow rate and the OA flow rate delivered through the main RTU. The MAU is a two-stage DX coil with a heating coil and supply fan. In these instances, the main RTU is similar to the MAS, but smaller. The Main RTU is sized such that the OA fraction entering the cooling coil is no greater than 0.15 and the cooling coil operates at 350 cfm/ton at rating conditions.

The MAU DX coil performance is that of a DOAS coil taken from the EnergyPlus library. The heating coil is a gas-fired furnace with an efficiency of 0.8. The MAU provides air between 55ºF and 65ºF and the temperature in the service area is allowed to float. Humidity of the air leaving the cooling coil is not controlled. Both coils are sized using the EnergyPlus auto sizing routine. The supply fan on the MAU is a variable volume fan interlocked with the exhaust fans. The supply fan delivers 115% of the exhaust flow rate at all times.

A.1.3.3 Outdoor Air Pretreatment Strategies (Pre-Treat)

The Pre-Treat Strategies employ an OA unit located upstream of the mixing box in which recirculated air and OA are mixed before going to the main Sales RTU. The OA unit for DX-based Pre-Treat systems is a four-stage DX coil only. For both the Pre-Treat DX system and the DOAS DX system described below, a control algorithm for a commercially available product is implemented, which maintains the condition of the air leaving the evaporator near 49ºF.

The Pre-Treat coil uses curves for a DOAS coil included in the EnergyPlus libraries. Each stage is 25% of the total capacity of the unit. The unit is sized such that the airflow/capacity ratio is 150 cfm/ton for the greatest OA flow rate expected. Stages are turned on and off based on the OA humidity ratio and flow rate. Heating, when needed, is provided by the main Sales RTU downstream of the OA unit.

The Sales RTU is sized at an airflow/capacity ratio of 450 cfm/ton and the supply fan is sized for a flow rate equal to the greatest OA flow rate expected, which is 0.5 cfm/ft^2 of Sales area.

A.1.3.4 Adaptive Multi-Path System and Variable Capacity Direct Expansion System

Both the Variable Capacity DX System and the Adaptive Multi-Path System are commercially available products with several proprietary algorithms built into their operation that take the following as inputs: predicted loads in the space, outdoor conditions, and indoor conditions. Multiple compressors are turned on and off in response to demand. For the Adaptive Multi-Path System, three damper positions vary continuously to optimize cooling. For these reasons, a rigorous physical description of the system in Modelica could not be built and simulated in the time allotted, as was done for less complex DX-based systems. Instead, the manufacturer provided supply conditions and resulting power draws for its system, operating on a similar building in its own software. These supply conditions were fed to an EnergyPlus building in order to determine the effect on refrigeration energy consumption and space conditions. The manufacturer demonstrated the accuracy of its simulation software by comparing results with field data (shown in Appendix C). The Adaptive Multi-Path System was sized based on recommendations from the manufacturer for each climate modeled, and the appropriate number of units were specified.

A.1.3.5 Interior Dehumidifiers

Two strategies were investigated in which a dehumidifier is placed entirely in the space, in addition to the RTU that provides only sensible cooling. The following sections describe the methods used to model and size these systems. In both cases, dehumidifiers operate at full capacity when refrigerated section DPT rises above 50ºF. The main RTU conditioning for both the dry goods and refrigerated section is sized to only provide sensible cooling, at 450 cfm/ton.

A.1.3.5.1 Direct Expansion Interior Dehumidifier

The DX dehumidifier model is created from experimental data (Christensen and Winkler 2009) that is converted to a polynomial to predict performance. The polynomials are implemented as a Modelica component that uses inlet conditions as input, and then outputs outlet conditions and power use. A sufficient number of DX dehumidifiers are specified to remove the entire latent load.

A.1.3.5.2 Liquid Desiccant System Used as Interior Dehumidifier

This strategy employs an LD System to remove the latent load from the space and a DX coil to provide sensible cooling to the ventilation air. The model of the LD dehumidifier is built up from library components and user-generated models of the LD conditioner and regenerator. These models are built from a first-principles finite element model for the conditioner and from purely empirical data for the regenerator. Outputs of the finite element analysis are mapped to inputs and a set of polynomials was generated that predicted conditioner and regenerator performance. The LD System is used to provide additional dehumidification and thus allow the refrigeration system to operate more efficiently. Therefore, the LD System is sized to maintain a DPT of 45ºF in the refrigerated section of the Sales Zone.

A.1.3.6 Dedicated Outdoor Air System Strategies

All DOAS strategies employ two parallel systems in the Sales Zone: a DOAS system conditions 100% OA and delivers it directly to the refrigerated section, and a recirculation RTU conditions 100% return air and delivers it to the dry goods section. These two zones are mixed at a rate equal to the OA flow rate in the refrigerated section. The recirculation RTU is sized at an airflow/capacity ratio of 450 cfm/ton and the supply fan is sized for a flow rate of 0.5 cfm/ft^2 of Sales area. The DOAS system is controlled based on a thermostat and humidistat in the refrigerated section. The system maintains a DPT below 50ºF in the refrigerated section at all times. Additional gas heat is added to the refrigerated section if needed to keep the space above 60ºF DBT at all times.

A.1.3.6.1 Direct Expansion Dedicated Outdoor Air System Strategies

The DX DOAS strategies employ a unit delivering OA directly to the refrigerated section of the Sales Zone—in this case, the unit is a four-stage DX coil with a desuperheater reheat coil. The coil uses curves for a DOAS coil, included in the EnergyPlus libraries, which are distributed with the standard EnergyPlus installation. Each stage is 25% of the total capacity of the unit. The unit is sized such that the airflow/capacity ratio is 150 cfm/ton for the greatest OA flow rate expected. The manufacturer of one of these systems provide a sequence of operations that, when implemented, maintains air leaving the evaporator near 49ºF when the unit is in operation.

A.1.3.6.3 Liquid Desiccant Dedicated Outdoor Air System

The LD System is mainly a dehumidifier with a cooling tower that can be assumed to remove the additional sensible load generated during the absorption process occurring within the LD conditioner. The same model used for the interior LD dehumidifier is used for the LD conditioner and regenerator. The LD conditioner is sized based on recommendations from the manufacturer.

A.1.3.6.4 Solid Desiccant Dedicated Outdoor Air System and Solid Desiccant Pre-Treat

The model used to simulate performance of a condenser heat-regenerated desiccant wheel includes a proprietary control strategy that activates the four compressors available, based on outdoor and space conditions. In this case, the mode is general for all climates and does not need a sizing routine. Manufacturer-provided data are formulated into polynomials in the OA conditions and space conditions, which predict power and performance over the range of operation expected. Validation of these polynomials is given in Appendix C. The parallel sensible coil is sized to remove the entire sensible load.

A.2 Building

The supermarket model used for all investigations is a 47,000-ft² supermarket divided into four zones: Backroom, Sales Zone, Offices, and Service Zone. We modeled a building compliant with 10-year old energy codes in order to capture the performance of a large portion of existing U.S. building stock. In general, the building was specified in order to be compliant with ASHRAE 90.1-2004, Appendix G, with exceptions or modifications as noted.

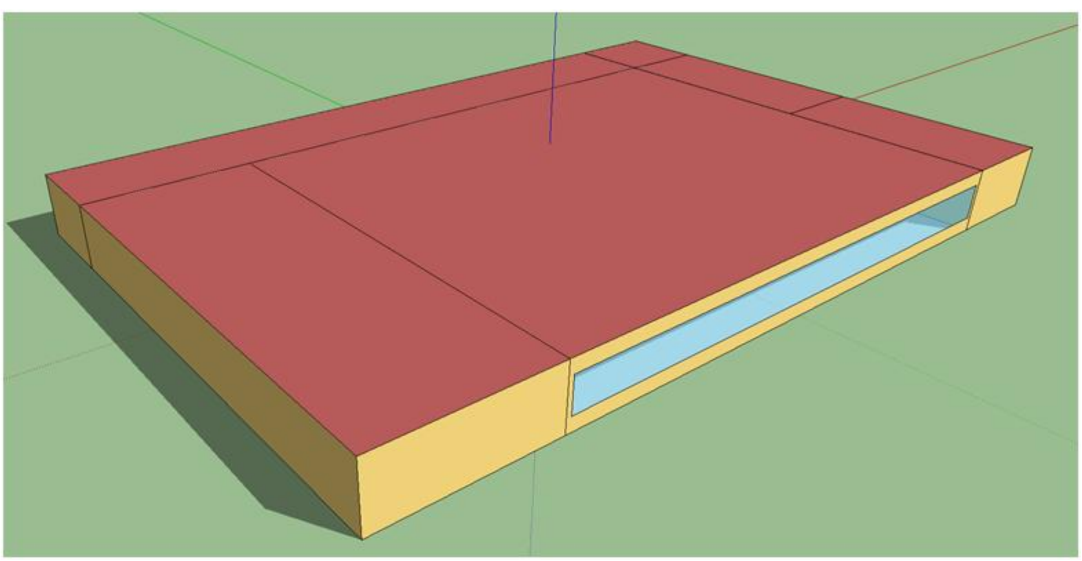


Figure A4. Rendering of supermarket building in EnergyPlus

A.2.1 Zoning

The building models were divided into four total zones, as shown in Figure A5 and described here:

- Sales: 32,500-ft² zone, including refrigeration equipment and a high occupancy.
    - All of the refrigerated cases in the Sales Zone are located within a "subzone" that is 10,000 ft^2, referred to as the "refrigerated section". The refrigerated section is completely within the Sales Zone. The remaining portion of the Sales Zone is referred to as the "dry goods section."
- Service: 7,000-ft² zone, including refrigeration equipment and cooking equipment/exhaust
- Office: 2,500-ft² zone of conventional office design.
- Backroom: 5,000-ft² zone of conventional warehouse design.

Zone occupancy and load profiles were set to match the profiles established by the ASHRAE 90.1-2004 User Manual Tables G-B and G-E through G-N. Occupancy and Load Design Values are given below in Table A2.

Table A2. Occupancy and Load Design Values

	Space Criteria	Area	Design Occupancy Density (sf/per)	Design Lighting Power Density (W/sf)	Design Misc. Electric Load Density (W/sf)
Sales	Manufacturing: High Bay	32,503	300	1.7	0.25
Service	Food Preparation	7,001	750	1.2	0.2
Office	Office	2,500	275	1.1	0.75
Backroom	Active Storage	5,000	15,000	0.8	0.1

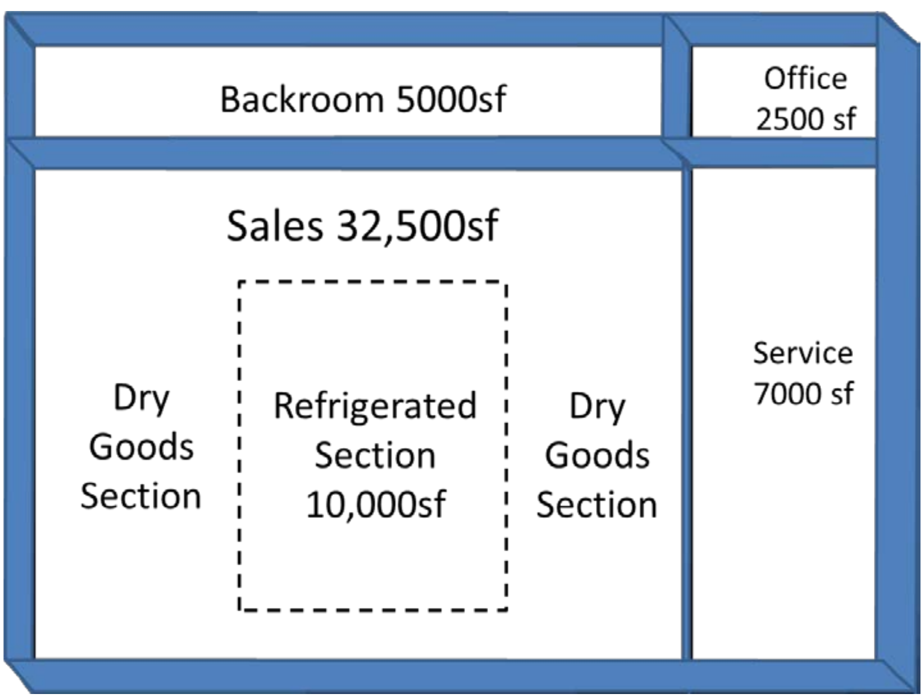


Figure A5. Designation of HVAC zones within supermarket

A.2.2 Building Envelope

We chose building envelopes for each climate zone to meet baseline requirements established by ASHRAE 90.1-2004 Tables 5.5-1 through 5.5-8 and Appendix G. Exterior walls were considered solid-grouted concrete masonry units with required continuous insulation to maintain the typical mass-wall construction of large retail buildings, in lieu of the steel-frame construction specified by Appendix G. The exterior roof is Insulation Entirely Above Deck (IEAD) with a U-value corresponding to minimum requirements of each climate zone as defined by ASHRAE 90.1-2004, shown below in Table A3. Exterior fenestrations are minimal with foam-core steel exterior doors and clear, double-pane exterior window constructions with standard ½-in. air gaps.

Table A3. Thermal Properties Assumed for Building Envelope

ASHRAE Climate Zone	Wall U-Value	Roof U-Value	Window U-Value	Window SHGC	Swinging Door U-Value	Non-Swinging Door U-Value
1	0.58	0.063	1.22	0.25	0.7	1.45
2	0.58	0.063	1.22	0.25	0.7	1.45
3	0.151	0.063	0.57	0.25	0.7	1.45
4	0.151	0.063	0.57	0.39	0.7	1.45
5	0.123	0.063	0.57	0.39	0.7	1.45
6	0.104	0.063	0.57	0.39	0.7	0.5

Table A4. Window-Wall Ratio for Simulated Building Walls

	Window Wall Ratio
North	0.0%
East	0.0%
South	0.0%
West	20.2%

Infiltration air leakage rates were estimated using Chartered Institution of Building Services Engineers-T23 Building Tightness Specifications for Supermarkets with infiltration to meet the "Good Practice" qualification. The pressure drop coefficient per Chartered Institution of Building Services Engineers -T23 in cfm/ft^2 was corrected to match the reference wind speed of the BLAST coefficients (7.5 mph) using "Infiltration Modeling Guidelines for Commercial Building Energy Analysis." Air changes per hour were then calculated volumetrically per zone and scheduled to reduce infiltration to 25% during occupied hours.

Table A5. Infiltration Assumptions for Simulated Building

	Volume [ft3]	Infiltration ACH
Sales	812,619	0.15
Service	175,026	0.22
Office	62,509	0.24
Backroom	125,018	0.23

ACH values at 7.5 mph wind speed.

A.3 Schedules

We assigned schedules for occupancy, equipment, HVAC, lighting, and service water heating per the ASHRAE 90.1-2004 User Manual Tables G-B and Tables G-E through G-N. Table G-B provides "Acceptable Occupant Densities, Receptacle Power Densities, and Service Hot Water Consumption," which are to be defined on a space-use level. Tables G-E through G-N show occupancy-defined hourly load ratios for estimated peak lighting, occupancy, equipment, etc.

A.4 Set Points

Determining the SPs and allowable temperature and DPT ranges in supermarkets is important from a modeling standpoint and a real world operation standpoint. It is well understood that refrigerated and walk-in cases incur lower loads in drier, cooler environments. When a DX-based system is used to condition the air, cold dry air is supplied and the resulting space conditions near the refrigerated cases may be very cold and uncomfortable for customers. This problem is alleviated by using reheat provided by a desuperheater coil and/or additional natural gas or electric heat. The problem may also be alleviated by using LD or SD dehumidification systems to dry the air.

For modeling purposes, conducting an apples-to-apples energy comparison between a desiccant technology and a DX-only system presents a challenge because of the disparity in the resulting space conditions. Desiccant systems often provide a more comfortable space condition because they do not require overcooling. Consequently, the space conditions surrounding the refrigerated cases are warmer. DX-only systems, even with reheat, cause cooler, less comfortable space conditions. Yet, as long as they can maintain the same DPT ranges as the desiccant system (often not the case), the refrigerated cases will operate in a more optimal (cool, dry) environment.

For the purposes of modeling, we placed a few controls on all systems. These are summarized in Table A6 and explained here:

- Space is maintained at or above 60°F DBT at all times in the refrigerated section.
- For the MAS, MAU, and Pre-Treat scenarios in Figure 1 (all but DOAS), we assumed the thermostat was properly located in the dry-goods sales section of the store. Because the HVAC system was controlled based on that thermostat, the space conditions in the refrigerated section were allowed to float. Due to the sensible case credits from the refrigerated cases, the refrigerated section was typically 10°F colder than the dry-goods sales floor, as would be expected. During supermarket field demonstrations, NREL measured the refrigerated case aisle temperatures, which can get down to the low 50s at 2 feet above the floor and upper 50s at 5 feet above the floor adjacent to open MT cases. While we did not control the HVAC system to ensure that the refrigerated section maintained equal to or above 60°F, we verified that every system in all climates met this requirement in order to provide a more apples-to-apples comparison between the HVAC systems.
- For the DOAS scenario, we assumed that the thermostat and humidistat controlling the DOAS system were located in the refrigerated section, because the dry goods and refrigerated sections were controlled relatively independently.

- Both the refrigerated and greater dry-goods sections of the Sales Zone are maintained at or below a 50°F DPT when the HVAC system is capable of maintaining this condition. When the space was conditioned with a realistically sized RTU, this RTU was unable to maintain such a low DPT throughout the year in certain climates, as was expected. DPT control was chosen over RH control because RH control causes over-dehumidification when space DBT drops below the DBT SP. When DX systems are used for dehumidification, using RH control effectively produces a "moving target" because the RH increases as the DBT decreases, without any change in absolute humidity.
- A night setback strategy is implemented with a cooling SP of 85°F during unoccupied hours and 75°F during occupied hours, a heating SP of 60°F during unoccupied hours, and a heating SP of 68°F during occupied hours in the dry goods section. The refrigerated section is maintained above 60°F throughout the night and does not require cooling in any simulation.
- Other than these stipulations, the system is allowed to operate in the manner that it performs best. For this reason, the resulting space conditions are not identical from simulation to simulation. The effect of varying space conditions is captured in the resulting refrigeration energy use.
- This report does not capture the effect of different system designs on comfort. More discussion of this issue is shown subsequently when discussing the performance of the desiccant-based systems. The reader should take into account the better thermal comfort provided by the desiccant systems, especially the LD systems.

Table A6. SPs in Dry Goods and Refrigerated Sections of Sales Zone

Letter Designation	*Dry Goods Section Dry-Bulb Setpoints*	*Dry Goods Section Dew-Point Setpoints*	*Refrigerated Section Dry-Bulb Setpoints*	*Refrigerated Section Dew-Point Setpoints*
MAS-a	68-75°F	<50°F	Dry-Bulb	FLOATS, Never above 50°F
MAS-b	68-75°F	<50°F	FLOATS, Never below 60°F	FLOATS, ***Sometimes above 50°F***
MAU	68-75°F	<50°F	FLOATS, Never below 60°F	FLOATS, Never above 50°F
MAU + DCKV	68-75°F	<50°F	FLOATS, Never below 60°F	FLOATS, Never above 50°F
MAU+ DCKV+ HEH	68-75°F	<50°F	FLOATS, Never below 60°F	FLOATS, Never above 50°F
DX PRETREAT*	68-75°F	<50°F	FLOATS, Never below 60°F	FLOATS, Never above 50°F
DX DOAS*	68-75°F	<50°F	60-75°F	<50°F
DX INT DEHUM*	68-75°F	<50°F	FLOATS, Never below 60°F	<50°F
LD INT DEHUM*	68-75°F	<50°F	FLOATS, Never below 60°F	<50°F
VAR DX *	68-75°F	<50°F	60-75°F	<50°F
ADAPTABLE*	68-75°F	<50°F	60-75°F	<50°F
LD DOAS*	68-75°F	<50°F	60-75°F	<50°F
SD DOAS*	68-75°F	<50°F	60-75°F	<50°F
SD PRETREAT*	68-75°F	<50°F	FLOATS, Never below 60°F	FLOATS, Never above 50°F

*Information valid for both baseline and DCKV+HEH exhaust rates

A.5 Refrigeration Systems

The operation of the refrigeration system is affected by the HVAC system via changes in the space conditions. These effects are included in several terms that define the energy use of the refrigeration system. EnergyPlus defines the total refrigerated case evaporator load as

$$\dot{Q}_{case}=\dot{Q}_{walls}+\dot{Q}_{rad}+\dot{Q}_{inf,sens}+\dot{Q}_{inf,lat}+\dot{Q}_{lights}+\dot{Q}_{as}+\dot{Q}_{def}+\dot{Q}_{fan}+\dot{Q}_{restock}$$

where

$\dot{Q}_{case}$ is the total load on the refrigerated case evaporator (W)

$\dot{Q}_{walls}$ is the conduction load through case walls (W)

$\dot{Q}_{rad}$ is the radiation load through case walls (W)

$\dot{Q}_{inf,sens}$ is the sensible component of the load caused by infiltration into the case (W)

$\dot{Q}_{inf,lat}$ is the latent component of the load caused by infiltration into the case (W)

$\dot{Q}_{lights}$ is the load generated by interior case lights (W)

$\dot{Q}_{as}$ is energy used by anti-sweat heaters (W)

$\dot{Q}_{def}$ is the energy used by defrost cycles in the case (W)

$\dot{Q}_{fan}$ is the fan heat load (W)

$\dot{Q}_{restock}$ is the load added by warmer products being placed in the case (W) (California 2014)

Of the terms on the right hand side of the equation, $\dot{Q}_{walls,}$ $\dot{Q}_{inf,sens,}$ $\dot{Q}_{inf,lat,}$ $\dot{Q}_{as,}$ and $\dot{Q}_{def}$ are input at rating conditions and modified through multipliers based on space conditions via empirical modification curves built into EnergyPlus. In this way, the effect on refrigeration energy of keeping space conditions at, say, a lower DPT, is captured. For further information, the reader may refer to the EnergyPlus Engineering Reference (California 2014)

A.6 Economics and Energy Calculations

We included a brief economic analysis of simulated energy expenditures in this study. The only metrics we give are the total cost of energy for HVAC and refrigeration in the Sales and Service Zones for the year and price point for certain simple payback periods derived from this. We took assumptions for heating and gas prices from EIA (2005) and made a few assumptions for energy calculations. We converted from site to source energy using EnergyPlus default site-source conversion factors, which are 1.084 and 3.167 for gas and electricity, respectively. We took the heating value of natural gas delivered to consumers from EIA (2005). These values are given Table A7.

Table A7. Energy Prices and Natural Gas Heating Values Assumed for Simulations

	Gas		Electricity
	[$/1000 ft^3]	Heating Value [Btu/ft^3]	$/kWh
Miami	11.9	1,016	0.1004
Houston	8.57	1,024	0.08
Atlanta	11.21	1,016	0.1009
Baltimore	11.96	1,045	0.1076
Chicago	11.8	1,016	0.0888
Minneapolis	8.39	1,023	0.0986

The original intent of the study was to provide a full economic analysis, including several payback analyses; however, it quickly became clear that manufacturers were reluctant to release their pricing structures. In many cases, the prices of the systems vary widely between clients and locations. For this reason, a full economic analysis was not possible for this report. We hope that this information will better inform manufacturers about price point goals for their products, and that building owners can make more informed decisions about purchasing options.

Appendix B: Protocol or Co-Simulating EnergyPlus and External Program

This investigation is representative of a large expansion of the types of annual building simulations that can be run with DOE's EnergyPlus building simulation software. This expansion is accomplished by creating HVAC systems not currently available in the EnergyPlus library in the Modelica language and running co-simulations of EnergyPlus and Modelica to understand the interactions between the systems and the building. This approach is made possible through the EnergyPlusToFMU utility created at Lawrence Berkeley National Laboratory, available for free download at http://simulationresearch.lbl.gov/fmu/EnergyPlus/export/index.html. With this utility, users can export a Functional Mockup Unit of an entire EnergyPlus building with all desired features and import that information into an external software program with a higher level modeling language and greater capabilities. In this case, EnergyPlus was exported to Dymola, which implements the Modelica language. This allows creation of new systems, tighter and easier control of models through a commercial graphical user interface, and more detailed post-processing of results.

Systems were developed from existing components from the Modelica Standard Library (MSL) and the Modelica Buildings Library (MBL), in addition to components created during this study. The MSL is the standard with each software package that implements the Modelica language, and the MBL is created by the Simulation Research Group at Lawrence Berkeley National Laboratory. For example, a typical system model may include simple components such as a source and duct from the MSL, a DX cooling coil from the MBL, and a desiccant wheel model created during this study, all working together to predict system performance.

Once the systems were created, they were connected to a climate-specific reference building as defined in an EnergyPlus Input Data File (.idf). Only the Sales Zone HVAC system was modeled in Modelica, because this is the system of interest in this study and is expected to see the greatest benefit from advanced air conditioning and dehumidification strategies. The other zones' HVAC systems, as well as the heat transfer through architectural elements of the building, the refrigeration systems, and all other processes in the building, were simulated in EnergyPlus.

Weather data from the .epw file called by EnergyPlus was passed to Dymola to be used as a boundary condition at each time step. All simulations were conducted with the Dymola graphical user interface (GUI) used as the master program and the EnergyPlus solver as the slave. This passing of information between programs was accomplished through the Functional Mockup Interface, a standardized means of interfacing various simulation programs implemented in both EnergyPlus and Dymola. EnergyPlus models were exported to Dymola using the recently released *EnergyPlus to FMU* utility, which was refined during the course of this study. A co-simulation was then run, during which EnergyPlus and Dymola each solved their respective problems independently while exchanging necessary data at each time step.

B.1 Programming Protocol for Integration of EnergyPlus and New Systems

While several methods for creating the simulations in the current study exist, only one method was used. This method uses a fully integrated building (defined in and simulated by EnergyPlus) and HVAC system (defined in and simulated by Dymola), which were simulated using the Dymola interface as the master and EnergyPlus as the slave. This method was the most straightforward, given the resources available for the current study. A method for doing this integration is given subsequently, in detail, in order to facilitate the simulation of novel HVAC systems integrated with EnergyPlus buildings in the future. An .idf implementing this strategy and an .idf that uses an external spreadsheet for inputting conditions of pre-treated OA are also included.

Creating the .idf

For the process used to simulate the buildings in the current study, the first step was to create an .idf file that would be exported as an FMU. Within the .idf, we defined a Sales Zone HVAC system with nodes set up to interface with the external program. Objects allowing for export with the Functional Mockup Interface (FMI) standard were also included in the idf. A general process for doing so is shown next. This process draws heavily from and extends the instructions in the *EnergyPlus to FMU* Users' Guide, available at:
http://simulationresearch.lbl.gov/fmu/EnergyPlus/export/userGuide/index.html

1. Add an Air Loop on the zone of interest that contains four objects in the following order:
    a. OA Mixing Box
    b. Zone Splitter
    c. Thermal Zone
    d. Zone Mixer
2. Add a Controller: OutdoorAir object to control the OA Mixing Box
    a. Set the minimum *and* maximum OA fraction on this controller to 1. This is accomplished through use of a schedule with all schedule values equal to 1 (*HVAC:Zone:AllONES)* in the subsequent example
    b. Define the required OA flow rate schedule and ensure it is equal to the system flow rate (*HVAC:Zone:FlowRateSchedule)* shown here

Controller:OutdoorAir,
, !- Name
, !- Relief Air Outlet Node Name
,!- Return Air Node Name
, !- Mixed Air Node Name
, !- Actuator Node Name
, !- Minimum OA Flow Rate {m3/s}
, !- Maximum OA Flow Rate {m3/s}
NoEconomizer, !- Economizer Control Type
, !- Economizer Control Action Type
, !- Economizer Maximum Limit DBT {C}
, !- Economizer Maximum Limit Enthalpy {J/kg}
, !- Economizer Maximum Limit DPT {C}
, !- Electronic Enthalpy Limit Curve Name
, !- Economizer Minimum Limit DBT {C}

NoLockout, !- Lockout Type
FixedMinimum, !- Minimum Limit Type
HVAC:Zone:FlowRateSchedule, !- Minimum OA Schedule Name
HVAC:Zone:AllONES, !- Minimum Fraction of OA Schedule Name
HVAC:Zone:AllONES; !- Maximum Fraction of OA Schedule Name

3. Add Output:Variable objects:
   a. One for zone DBT
   b. One for zone DPT temperature or humidity ratio
   c. OA DBT
   d. OA DPT temperature or humidity ratio

   Example:

Output:Variable,ZONE,Zone Air Temperature,Timestep;
Output:Variable,ZONE,Zone Air Humidity Ratio,Timestep;
Output:Variable,HVAC System Outdoor Air Node,System Node Wetbulb Temperature,Timestep;
Output:Variable, HVAC System Outdoor Air Node,System Node Temperature,Timestep;

4. Add an object setting up the external interface

   Example:

ExternalInterface,
FunctionalMockupUnitExport; !- Name of External Interface

5. Add ExternalInterface:FunctionalMockupUnitExport:From:Variable objects:
   a. Zone DBT
   b. Zone wet bulb temperature or humidity ratio
   c. OA DBT
   d. OA wet bulb temperature or humidity ratio

   Example:

ExternalInterface:FunctionalMockupUnitExport:From:Variable,
ZONE, !- Output:Variable or Output:Meter Index Key Name
Zone Air Temperature, !- Output:Variable or Output:Meter Name
Zone_Temp ; !- FMU Variable Name
ExternalInterface:FunctionalMockupUnitExport:From:Variable,
ZONE, !- Output:Variable or Output:Meter Index Key Name
Zone Air Humidity Ratio, !- Output:Variable or Output:Meter Name
Zone_HR; !- FMU Variable Name
ExternalInterface:FunctionalMockupUnitExport:From:Variable,
HVAC System Outdoor Air Node!- Output:Variable or Output:Meter Index Key Name
System Node Temperature, !-Output:Variable or Output:Meter Name
OA_DBT; !-FMU Variable Name

ExternalInterface:FunctionalMockupUnitExport:From:Variable,
HVAC System Outdoor Air Node!- Output:Variable or Output:Meter Index Key Name
System Node Wetbulb Temperature, !-Output:Variable or Output:Meter Name
OA_WBT; !-FMU Variable Name

6. Add ExternalInterface:FunctionalMockupUnitExport:Actuator objects:
    a. OA System Node DBT
    b. OA System Node DPT
    c. OA controller mass flow rate (if you wish to control supply flow rate externally)

    Example:

ExternalInterface:FunctionalMockupUnitExport:To:Actuator,
ExternalSupplyTdb, !- Name
HVAC System Outdoor Air Node, !- Actuated Component Unique Name
Outdoor Air System Node, !- Actuated Component Type
Drybulb Temperature, !- Actuated Component Control Type
Sales_DBT_in, !- FMU Variable Name
15; !- Initial Value [C]
ExternalInterface:FunctionalMockupUnitExport:To:Actuator,
ExternalSupplyTwb, !- Name
HVAC System Outdoor Air Node, !- Actuated Component Unique Name
Outdoor Air System Node, !- Actuated Component Type
Wetbulb Temperature, !- Actuated Component Control Type
Sales_WBT_in, !- FMU Variable Name
15; !- Initial Value [C]
ExternalInterface:FunctionalMockupUnitExport:To:Actuator,
ExternalFlowRate, !- Name
HVAC System Outdoor Air Controller, !- Actuated Component Unique Name
Outdoor Air Controller, !- Actuated Component Type
Air Mass Flow Rate, !- Actuated Component Control Type
Sales_MFR_in, !- FMU Variable Name
3.5; !- Initial Value

7. Follow all other instructions in the *EnergyPlus to FMU* User's Guide.

Converting the .idf to an FMU

Once the .idf file was created, the *EnergyPlus to FMU* utility available at http://simulationresearch.lbl.gov/fmu/EnergyPlus/export/userGuide/download.html was used to convert the .idf to a Functional Mockup Unit (FMU) which could be imported into Dymola.

Importing into Dymola

Before importing into Dymola, the user must ensure that the cosimulation radio button is checked under Simulation->Setup->FMI. Then use File->Import->FMU to import the FMU. Only one FMU can be imported per Dymola session, and only if no other model is loaded with the name of the FMU. It is best to import the FMU at the very beginning of a session.

Running the Simulation

Once the FMU is imported, a Dymola model is created that functions exactly as other models in available Modelica libraries. It does not need to be imported again. After the FMU is imported, a few steps must be taken to ensure the simulation runs:

1. Ensure the working directory is the same as it was when the FMU was imported, or move the files generated during import to the current working directory.
2. Double-click on the FMU model in the Dymola GUI.
3. Give initial values to all variables.
4. On the “FMI” tab, fmi_StopTime must be set to the simulation StopTime, defined in Simulation->Setup.
5. The fmi_CommunicationStepSize must be set to the EnergyPlus time step defined in the .idf, in seconds.
6. fmi_pullInputsForInitialization must be set to false when a full feedback loop is being modeled, as it is in the current study for systems other than DOAS systems.
7. Modelica.Blocks.Discrete.FirstOrderHold objects should be placed at the outlet of each variable output from the FMU to convert the discrete signal sent from the FMU to a continuous signal more easily processed by the Dymola solver.
8. Co-simulation often results in stiff systems that must be run with the Dassl algorithm.
9. Often, a smaller convergence tolerance than usual must be used to ensure convergence.

Next is a snippet of an .idf containing the HVAC objects and parameters necessary for exporting EnergyPlus as an FMU and introducing conditioned air modeled in an external program into a building simulation done with EnergyPlus. The .idf from which these objects are taken replaces the OA conditions on the Refrigeration DOAS object with air conditions from an external program, which can be the supply conditions from a DOAS unit modeled in an external program. The entire idf could not be included because of its length, but the snippet should be instructive for an advanced user and allow them to create their own simulations:

```
ExternalInterface,
 FunctionalMockupUnitExport; !- Name of External Interface
ExternalInterface:FunctionalMockupUnitExport:From:Variable,
 ZONE:Refrigeration, !- Output:Variable or Output:Meter Index Key Name
 Zone Air Temperature, !- Output:Variable or Output:Meter Name
 T_RoomMean;           !- FMU Variable Name
ExternalInterface:FunctionalMockupUnitExport:From:Variable,
 Zone:Refrigeration, !- Output:Variable or Output:Meter Index Key Name
 Zone Air Humidity Ratio, !- Output:Variable or Output:Meter Name
 HR_RoomMean; !- FMU Variable Name
ExternalInterface:FunctionalMockupUnitExport:From:Variable,
 Service RTU OSA Node, !- Output:Variable or Output:Meter Index Key Name
 System Node Temperature, !- Output:Variable or Output:Meter Name
```

```
 OA_DBT; !- FMU Variable Name
ExternalInterface:FunctionalMockupUnitExport:From:Variable,
 Service RTU OSA Node, !- Output:Variable or Output:Meter Index Key Name
 System Node Humidity Ratio, !- Output:Variable or Output:Meter Name
 OA_HR; !- FMU Variable Name

ExternalInterface:FunctionalMockupUnitExport:To:Actuator,
 ExternalSupplyTdb, !- Name
 Refrigeration DOAS OSA Node, !- Actuated Component Unique Name
 Outdoor Air System Node, !- Actuated Component Type
 Drybulb Temperature, !- Actuated Component Control Type
 Sales_DBT_in, !- FMU Variable Name
 15; !- Initial Value
ExternalInterface:FunctionalMockupUnitExport:To:Actuator,
 ExternalSupplyTwb, !- Name
 Refrigeration DOAS OSA Node, !- Actuated Component Unique Name
 Outdoor Air System Node, !- Actuated Component Type
 Wetbulb Temperature, !- Actuated Component Control Type
 Sales_WBT_in, !- FMU Variable Name
 15; !- Initial Value
ExternalInterface:FunctionalMockupUnitExport:To:Actuator,
 ExternalFlowRate, !- Name
 Refrigeration DOAS OA Controller, !- Actuated Component Unique Name
 Outdoor Air Controller, !- Actuated Component Type
 Air Mass Flow Rate, !- Actuated Component Control Type
 Sales_MFR_in, !- FMU Variable Name
 7.5; !- Initial Value

AirLoopHVAC,
 Refrigeration DOAS, !- Name
 , !- Controller List Name
 Refrigeration DOAS Availability Managers, !- Availability Manager List Name
 8.345, !- Design Supply Air Flow Rate {m3/s}
 Refrigeration DOAS BranchList, !- Branch List Name
 , !- Connector List Name
 Refrigeration DOAS Air Loop Inlet,!- Supply Side Inlet Node Name
 Refrigeration DOAS Return Air Outlet, !- Demand Side Outlet Node Name
 Refrigeration DOAS Supply Path Inlet, !- Demand Side Inlet Node Names
 Refrigeration DOAS Air Loop Outlet; !- Supply Side Outlet Node Names

AirLoopHVAC:OutdoorAirSystem:EquipmentList,
 Refrigeration DOAS OA System Equipment, !- Name
 OutdoorAir:Mixer, !- Component 1 Object Type
 Refrigeration DOAS OA Mixing Box; !- Component 1 Name

AirLoopHVAC:OutdoorAirSystem,
```

```
 Refrigeration DOAS OA System, !- Name
 Refrigeration DOAS OA System Controllers, !- Controller List Name
 Refrigeration DOAS OA System Equipment, !- Outdoor Air Equipment List Name
 Refrigeration DOAS Availability Managers; !- Availability Manager List Name

AirLoopHVAC:ControllerList,
 Refrigeration DOAS OA System Controllers, !- Name
 Controller:OutdoorAir, !- Controller 1 Object Type
 Refrigeration DOAS OA Controller; !- Controller 1 Name

OutdoorAir:Mixer,
 Refrigeration DOAS OA Mixing Box, !- Name
 Refrigeration DOAS Air Loop Outlet, !- Mixed Air Node Name
 Refrigeration DOAS OSA Node, !- Outdoor Air Stream Node Name
 Refrigeration DOAS Relief Air Outlet, !- Relief Air Stream Node Name
 Refrigeration DOAS Air Loop Inlet;!- Return Air Stream Node Name

Branch,
 Refrigeration DOAS Main Branch, !- Name
 8.345, !- Maximum Flow Rate {m3/s}
 , !- Pressure Drop Curve Name
 AirLoopHVAC:OutdoorAirSystem, !- Component 1 Object Type
 Refrigeration DOAS OA System, !- Component 1 Name
 Refrigeration DOAS Air Loop Inlet,!- Component 1 Inlet Node Name
 Refrigeration DOAS Air Loop Outlet, !- Component 1 Outlet Node Name
 Passive;
OutdoorAir:Node,
 Refrigeration DOAS OSA Node, !- Name
 9.14400000000001; !- Height Above Ground {m}

OutdoorAir:Node,
 Service RTU OSA Node, !- Name
 9.14400000000001; !- Height Above Ground {m}
```

B.2 Sample .idf Snippet for Inputting Spreadsheet Values of Supply Conditions

If no feedback is needed from the model of the building, a simple spreadsheet file can be used to input the conditions of the air supplied by the external air unit. This file can be generated in an external program and thus linked with an EnergyPlus simulation. The following snippet shows the HVAC code for the Sales Zone of a supermarket that is conditioned by an RTU with a pre-treat DOAS system modeled externally. The supply conditions leaving the pre-treat unit are contained in files “DrybulbTemps.csv” and WetbulbTemps.csv” and are used to modify the boundary conditions of the OA mixer in EnergyPlus, thus effectively supplying the pre-treated air to the mixing box of the EnergyPlus RTU.

```
!! External File containing DryBulb Temperature Schedule

Schedule:File,
 Drybulb Temperature Sch, !- Name
 Temperature, !- Schedule Type Limits Name
 F:\DybulbTemps.csv, !- File Name
 2, !- Column Number
 1, !- Rows to Skip at Top
 8760, !- Number of Hours of Data
 comma, !- Column Separator
 No, !- Interpolate to Timestep
 15; !- Minutes per Item

!! External File containing WetBulb Temperature Schedule
Schedule:File,
 Wetbulb Temperature Sch, !- Name
 Temperature, !- Schedule Type Limits Name
 F:\WetbulbTemps.csv, !- File Name
 3, !- Column Number
 1, !- Rows to Skip at Top
 8760, !- Number of Hours of Data
 comma, !- Column Separator
 No, !- Interpolate to Timestep
 15; !- Minutes per Item



ZoneHVAC:EquipmentConnections,
 Sales, !- Zone Name
 Zone1Equipment, !- Zone Conditioning Equipment List Name
 Zone1Inlets, !- Zone Air Inlet Node or NodeList Name
 Sales Exhaust Nodes, !- Zone Air Exhaust Node or NodeList Name
 Sales Air Node, !- Zone Air Node Name
 Sales Outlet Node; !- Zone Return Air Node Name

Fan:OnOff,
 Sales Supply Fan 1, !- Name
 Fan_OffAtNight, !- Availability Schedule Name
 0.7, !- Fan Efficiency
 600.0, !- Pressure Rise {Pa}
 autosize, !- Maximum Flow Rate {m3/s}
 0.9, !- Motor Efficiency
 1.0, !- Motor In Airstream Fraction
 Sales Mixed Air Node, !- Air Inlet Node Name
 Sales DX Cooling Coil Air Inlet Node; !- Air Outlet Node Name

Fan:ZoneExhaust,
 Sales Exhaust Fan, !- Name
 HVACOperationSchd, !- Availability Schedule Name
 1.0, !- Fan Efficiency
 1.0e-006, !- Pressure Rise {Pa}
```

```
 1.0785, !- Maximum Flow Rate {m3/s}
 Sales Exhaust Fan Node, !- Air Inlet Node Name
 Sales Exhaust Fan Outlet Node Name, !- Air Outlet Node Name
 Zone Exhaust Fans, !- End-Use Subcategory
 Fan_OffAtNight; !- Flow Fraction Schedule Name

Coil:Cooling:DX:SingleSpeed,
 Sales ACDXCoil 1, !- Name
 ALWAYS_ON, !- Availability Schedule Name
 AUTOSIZE, !- Rated Total Cooling Capacity {W}
 AUTOSIZE, !- Rated Sensible Heat Ratio
 3.66668442928701, !- Rated COP {W/W}
 AUTOSIZE, !- Rated Air Flow Rate {m3/s}
 , !- Rated Evaporator Fan Power Per Volume Flow Rate {W/(m3/s)}
 Sales DX Cooling Coil Air Inlet Node, !- Air Inlet Node Name
 Sales Heating Coil Air Inlet Node, !- Air Outlet Node Name
 WindACCoolCapFT, !- Total Cooling Capacity Function of Temperature Curve Name
 WindACCoolCapFFF, !- Total Cooling Capacity Function of Flow Fraction Curve Name
 WindACEIRFT, !- Energy Input Ratio Function of Temperature Curve Name
 WindACEIRFFF, !- Energy Input Ratio Function of Flow Fraction Curve Name
 WindACPLFFPLR, !- Part Load Fraction Correlation Curve Name
 1000, !- Nominal Time for Condensate Removal to Begin {s}
 1.5, !- Ratio of Initial Moisture Evaporation Rate and Steady State Latent Capacity
 3, !- Maximum Cycling Rate {cycles/hr}
 45; !- Latent Capacity Time Constant {s}


Coil:Heating:Gas,
 Sales Furnace Heating Coil 1, !- Name
 HeatingCoil_Availability,!- Availability Schedule Name
 0.8, !- Gas Burner Efficiency
 AUTOSIZE, !- Nominal Capacity {W}
 Sales Heating Coil Air Inlet Node, !- Air Inlet Node Name
 Sales Reheat Coil Air Inlet Node; !- Air Outlet Node Name

AirLoopHVAC:Unitary:Furnace:HeatCool,
 Sales Rooftop DX w/ Gas Heat, !- Name
 ALWAYS_ON, !- Availability Schedule Name
 Sales Mixed Air Node, !- Furnace Air Inlet Node Name
 Sales Reheat Coil Air Inlet Node, !- Furnace Air Outlet Node Name
 ALWAYS_ON, !- Supply Air Fan Operating Mode Schedule Name
 80, !- Maximum Supply Air Temperature {C}
 autosize, !- Supply Air Flow Rate During Cooling Operation {m3/s}
 autosize, !- Supply Air Flow Rate During Heating Operation {m3/s}
 autosize, !- Supply Air Flow Rate When No Cooling or Heating is Needed {m3/s}
 Sales, !- Controlling Zone or Thermostat Location
 Fan:OnOff, !- Supply Fan Object Type
 Sales Supply Fan 1, !- Supply Fan Name
 BlowThrough, !- Fan Placement
 Coil:Heating:Gas, !- Heating Coil Object Type
 Sales Furnace Heating Coil 1, !- Heating Coil Name
```

```
 Coil:Cooling:DX:SingleSpeed, !- Cooling Coil Object Type
 Sales ACDXCoil 1, !- Cooling Coil Name
 None; !- Dehumidification Control Type

Controller:OutdoorAir,
 Sales OA Controller 1, !- Name
 Sales Relief Air Outlet Node, !- Relief Air Outlet Node Name
 Sales Outdoor Air Mixer Inlet Node, !- Return Air Node Name
 Sales Mixed Air Node, !- Mixed Air Node Name
 Sales Outside Air Inlet Node, !- Actuator Node Name
 1.4584, !- Minimum Outdoor Air Flow Rate {m3/s}
 1.4584, !- Maximum Outdoor Air Flow Rate {m3/s}
 NoEconomizer, !- Economizer Control Type
 ModulateFlow, !- Economizer Control Action Type
 , !- Economizer Maximum Limit DBT {C}
 , !- Economizer Maximum Limit Enthalpy {J/kg}
 , !- Economizer Maximum Limit Dewpoint Temperature {C}
 , !- Electronic Enthalpy Limit Curve Name
 , !- Economizer Minimum Limit DBT {C}
 NoLockout, !- Lockout Type
 FixedMinimum, !- Minimum Limit Type
 MinOA_Sched; !- Minimum Outdoor Air Schedule Name

AirLoopHVAC:ControllerList,
 Sales OA Sys 1 Controllers, !- Name
 Controller:OutdoorAir, !- Controller 1 Object Type
 Sales OA Controller 1; !- Controller 1 Name

AirLoopHVAC,
 Sales HVAC System, !- Name
 , !- Controller List Name
 Furnace 1 Avail List, !- Availability Manager List Name
 autosize, !- Design Supply Air Flow Rate {m3/s}
 Sales Air Loop Branches, !- Branch List Name
 , !- Connector List Name
 Sales Outdoor Air Mixer Inlet Node, !- Supply Side Inlet Node Name
 Sales Return Air Mixer Outlet, !- Demand Side Outlet Node Name
 Sales Zone Equipment Inlet Node, !- Demand Side Inlet Node Names
 Sales Reheat Coil Air Inlet Node; !- Supply Side Outlet Node Names

AirLoopHVAC:OutdoorAirSystem:EquipmentList,
 Sales OA Sys 1 Equipment,!- Name
 OutdoorAir:Mixer, !- Component 1 Object Type
 Sales OA Mixing Box 1; !- Component 1 Name

AirLoopHVAC:OutdoorAirSystem,
 Sales OA Sys 1, !- Name
 Sales OA Sys 1 Controllers, !- Controller List Name
 Sales OA Sys 1 Equipment,!- Outdoor Air Equipment List Name
 Outdoor Air 1 Avail List;!- Availability Manager List Name
```

```
OutdoorAir:Mixer,
 Sales OA Mixing Box 1, !- Name
 Sales Mixed Air Node, !- Mixed Air Node Name
 Sales Outside Air Inlet Node, !- Outdoor Air Stream Node Name
 Sales Relief Air Outlet Node, !- Relief Air Stream Node Name
 Sales Outdoor Air Mixer Inlet Node; !- Return Air Stream Node Name

AirLoopHVAC:ZoneSplitter,
 Sales Zone Supply Air Splitter, !- Name
 Sales Zone Equipment Inlet Node, !- Inlet Node Name
 Sales Inlet Node; !- Outlet 1 Node Name

AirLoopHVAC:SupplyPath,
 Sales FurnaceSupplyPath, !- Name
 Sales Zone Equipment Inlet Node, !- Supply Air Path Inlet Node Name
 AirLoopHVAC:ZoneSplitter,!- Component 1 Object Type
 Sales Zone Supply Air Splitter; !- Component 1 Name

AirLoopHVAC:ZoneMixer,
 Sales Zone Return Air Mixer, !- Name
 Sales Return Air Mixer Outlet, !- Outlet Node Name
 Sales Outlet Node; !- Inlet 1 Node Name

AirLoopHVAC:ReturnPath,
 Sales FurnaceReturnPath, !- Name
 Sales Return Air Mixer Outlet, !- Return Air Path Outlet Node Name
 AirLoopHVAC:ZoneMixer, !- Component 1 Object Type
 Sales Zone Return Air Mixer; !- Component 1 Name

Branch,
 Sales Air Loop Main Branch, !- Name
 autosize, !- Maximum Flow Rate {m3/s}
 , !- Pressure Drop Curve Name
 AirLoopHVAC:OutdoorAirSystem, !- Component 1 Object Type
 Sales OA Sys 1, !- Component 1 Name
 Sales Outdoor Air Mixer Inlet Node, !- Component 1 Inlet Node Name
 Sales Mixed Air Node, !- Component 1 Outlet Node Name
 PASSIVE, !- Component 1 Branch Control Type
 AirLoopHVAC:Unitary:Furnace:HeatCool, !- Component 2 Object Type
 Sales Rooftop DX w/ Gas Heat, !- Component 2 Name
 Sales Mixed Air Node, !- Component 2 Inlet Node Name
 Sales Reheat Coil Air Inlet Node, !- Component 2 Outlet Node Name
 ACTIVE; !- Component 2 Branch Control Type

BranchList,
 Sales Air Loop Branches, !- Name
 Sales Air Loop Main Branch; !- Branch 1 Name

NodeList,
 Sales Exhaust Nodes, !- Name
 Sales Exhaust Fan Node; !- Node 1 Name
```

```
NodeList,
 Zone1Inlets, !- Name
 Sales Inlet Node; !- Node 1 Name

NodeList,
 SalesOutsideAirInletNodes, !- Name
 Sales Outside Air Inlet Node; !- Node 1 Name

OutdoorAir:NodeList,
 SalesOutsideAirInletNodes; !- Node or NodeList Name 1

EnergyManagementSystem:Sensor,
 Temp_OA_Drybulb, !- Name
 Environment, !- Output:Variable or Output:Meter Index Key Name
 Site Outdoor Air Drybulb Temperature; !- Output:Variable or Output:Meter Name

! These sensors read the OA Drybulb and OA Wetbulb Temperatures from the Excel schedule

EnergyManagementSystem:Sensor,
 AirTdb_From_Sch, !- Name
 Air Drybulb Temperature Sch, !- Output:Variable or Output:Meter Index Key Name
 Schedule Value; !- Output:Variable or Output:Meter Name

EnergyManagementSystem:Sensor,
 AirTwb_From_Sch, !- Name
 Air Wetbulb Temperature Sch, !- Output:Variable or Output:Meter Index Key Name
 Schedule Value; !- Output:Variable or Output:Meter Name

! These 2 actuators are used to set the Air condition at the OA Mixer Inlet Node of the Sales Zone
EnergyManagementSystem:Actuator,
 Sales_OutsideAirInletNode1_Tdb, !- Name
 Sales Outside Air Inlet Node, !- Actuated Component Unique Name
 Outdoor Air System Node, !- Actuated Component Type
 Drybulb Temperature; !- Actuated Component Control Type

EnergyManagementSystem:Actuator,
 Sales_OutsideAirInletNode1_Twb, !- Name
 Sales Outside Air Inlet Node, !- Actuated Component Unique Name
 Outdoor Air System Node, !- Actuated Component Type
 Wetbulb Temperature; !- Actuated Component Control Type

! This piece tells the program to run every HVAC system timestep before the Zone Load Predictor (was:
AfterPredictorAfterHVACManagers)
EnergyManagementSystem:ProgramCallingManager,
 ProgramCaller, !- Name
 AfterPredictorAfterHVACManagers, !- EnergyPlus Model Calling Point
 External; !- Program Name 1

! This is the actual code that executes every HVAC system timestep
EnergyManagementSystem:Program,
```

```
External, !- Name
SET Sales_OutsideAirInletNode1_Tdb = AirTdb_From_Sch, !- Program Line 1
SET Sales_OutsideAirInletNode1_Twb = AirTwb_From_Sch; !- Program Line 2
```

Appendix C: System Model Validation

We validated the models included in the simulations for each component using several different techniques. For some of the systems, we built up models from existing components in either the EnergyPlus or Modelica Buildings library that were previously validated. For three systems, empirical data were available and we compared outputs of the models against experimental data. For some systems, complicated control strategies and/or lack of individual component models required that, for modeling purposes, we draw a control volume around the entire system and use a performance map of the entire system. We generated a map of inputs at the outdoor intake or room air return and outputs of supply air and power use and compared them against outputs of manufacturer software. Lastly, we built up the WADW system models (not used) from existing MBL components in combination with a polynomial prediction of the outputs of a previously published finite element model (Kosar et al. 2007). See Figures C1 and C2 for graphs showing comparison between outputs of the models generated in this study and their respective data.

C.1 Adaptive Multi-Path System and Variable-Capacity Direct Expansion Dedicated Outdoor Air System

The model of the Adaptive Multi-Path System and the variable capacity DX DOAS systems was created by the manufacturer and validated with field data. The manufacturer first provided a report demonstrating that its software was capable of fairly accurately predicting the performance of its units.

In coordination with the manufacturer, we performed the validation of this system in a somewhat indirect manner. The manufacturer installed two 40-ton units on a grocery store in Markham, Ontario, and instrumented the store and system to measure temperature, humidity, and power draw. To validate its software, the manufacturer compared this data to the outputs of its software, which was then used to generate supply conditions for the simulations of this report.

Over an 11-month period, total modeled power consumption was shown to be 7.8% below measured power consumption. This trend holds fairly constant on a month-to-month basis as well, as shown in in Figure C1.

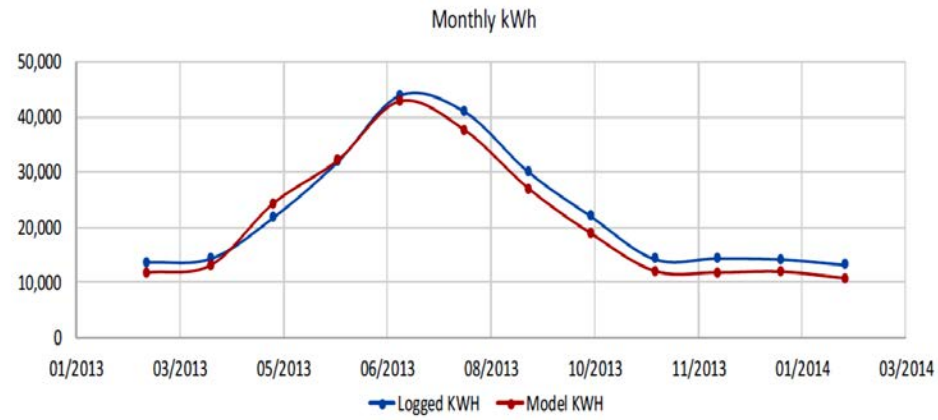


Figure C1. Comparison of modeled and measured data for Adaptive Multi-Path System

A similar comparison was done for space DBTs and DPTs. Supply temperatures were not measured in the field and thus could not be compared. Figure C2 below, taken from the manufacturer's report, shows the comparison of the space DBTs and DPTs. The SPs in the manufacturer's model were changed in mid-December of 2013 to match the store's SPs, which were not accurately input in the first portion of the comparison. All model outputs used in the current study are created with the correct SPs used after December of 2013. During this time, Figure C2 shows relatively good agreement between modeled and measured data.

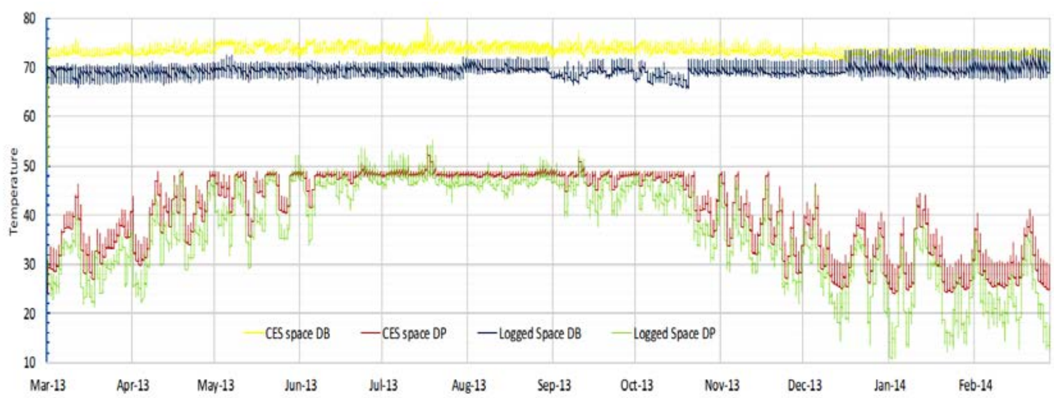


Figure C2. Comparison of modeled and measured space DBTs and DPTs

Credit: Consolidated Energy Solutions

C.2 Liquid Desiccant Dehumidifier

An LD dehumidifier was used both in a DOAS configuration and as an internal dehumidifier. The LD dehumidifier contained in these systems is a developmental product not currently available commercially. It has been tested in the laboratory as well as demonstrated in grocery stores in the United States. The two main component models of the system are an absorber/dehumidifier and a desiccant regenerator. A physical numerical finite difference model of the regenerator was created and an empirical fit of laboratory performance of the regenerator was also created.

The numerical model of the absorber was compared against laboratory data previously measured at NREL to assess its validity. A description of the experimental setup and some preliminary results are given by Lowenstein et al. (2006). Specifications of the conditioner and regenerator tested are given in Figure C3. With the stated assumptions and methods employed, the modeled moisture removal rate in the absorber compared well with the 32 lab conditions tested, as shown in Figure C4. Prediction of mass transfer/moisture removal in the absorber was accomplished with purely physical descriptions (no empirical constants).

	Absorber	***Regenerator***
H	1.22 m	0.61 m
L	0.30 m	0.17 m
W	0.26 m (42 plates)	0.13 m (21 plates)
Air Gap	3.1 mm	3.1 mm
Face Velocity of Air	1-2.5 m/s	0.77-1.24 m/s
Water Flow Rate per Length L per Plate	0.049-0.098 [kg/s/m/plate]	0.072-0.15 [kg/s/m/plate]
Desiccant Flow Rate per Length L per Plate	0.0015-0.0092 [kg/s/m/plate]	0.0021-0.0088 [kg/s/m/plate]
Inlet Air Temperature *Humidity Ratio* [kg_{water}/kg_{air}]	30 °C 0.0097-0.023	30-57 °C 0.0098-0.024
Inlet Desiccant Temp *Concentration* [$kg_{LiCl}/kg_{solution}$]	26-45 °C 0.355-0.44	60-57 °C 0.31-0.41
Inlet Water Temp	20-30 °C	71-93 °C

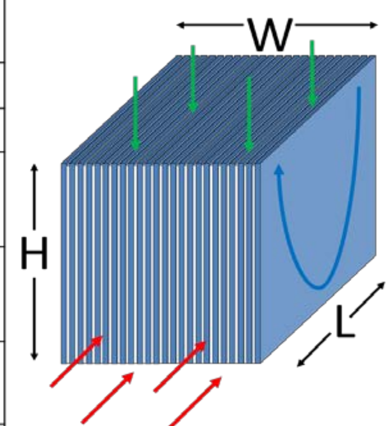


Figure C3. Specifications and operating conditions for tested absorber and regenerator

The figure to the right shows the flow directions of air (RED, into the page), water (BLUE, entering at the top rear of the exchanger moving downward and reversing direction) and desiccant (GREEN parallel to the page) as well as the dimensions referred to in the nearby table. All experiments used a lithium chloride-water solution as the desiccant.

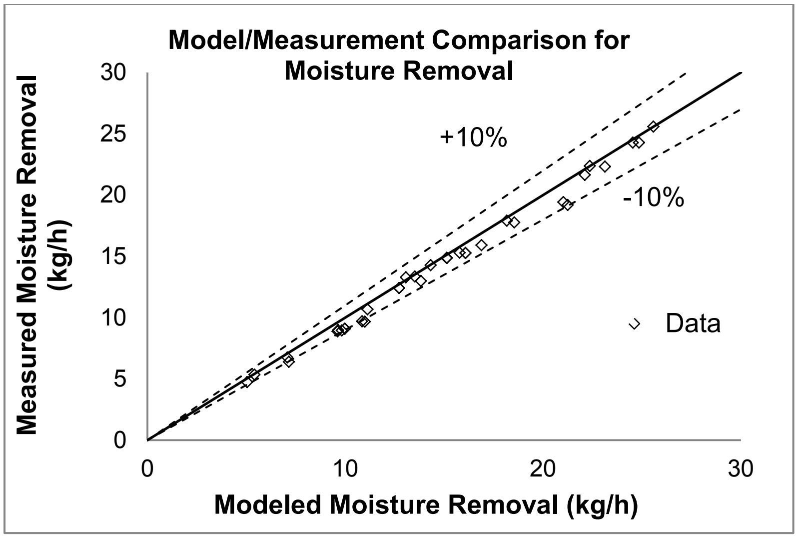


Figure C4. Comparison of absorber model and laboratory data showing good agreement

Uncertainty in measured value (due to precision of chilled mirrors used to measure humidity) is less than 5% of the measured value in all cases.

However, temperature of the fluid streams could not be measured immediately at the boundary of the modeled domain because of practical limitations, such as presence of the sump and the water distribution header. For this reason, some heat transfer occurred between the fluids and the ambient air before the temperature measurement point. This led to a discrepancy between the modeled and measured temperatures of the fluid, which increased with temperature difference between exiting fluid and the ambient air in the laboratory. Figure C5 shows this discrepancy for each of the three fluids. The vertical dashed line represents the ambient conditions in the space.

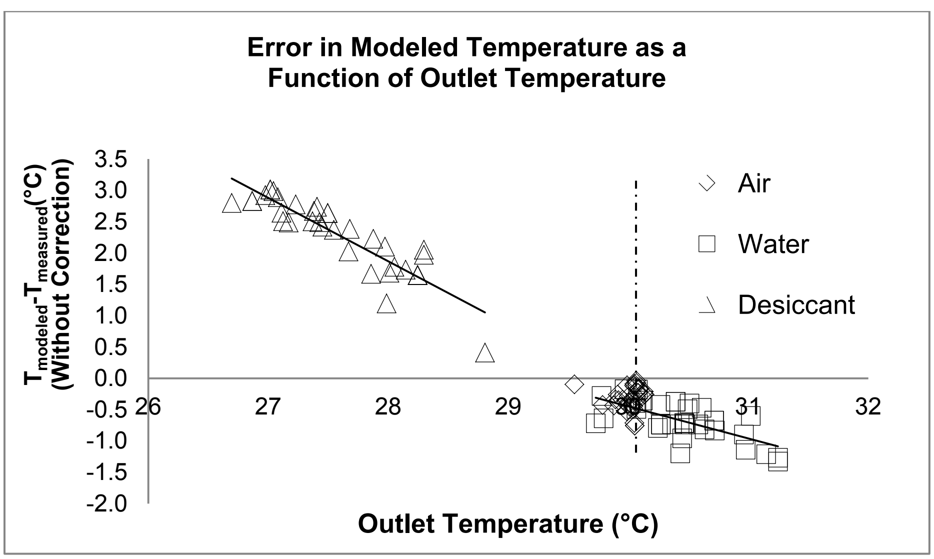


Figure C5. Discrepancy between measured and modeled temperature in three fluids of the absorber

An empirical heat transfer coefficient between the exiting fluid and the ambient, *h* in the equation below, was thus assumed to account for these losses:

$$\dot{m}_{\text{fluid}}(T_{\text{fluid,modeled}\mathbf{(edge\ of\ domain)}}-T_{\text{fluid,modeled}\mathbf{(measuring\ point)}})=h(T_{\text{fluid, modeled}\mathbf{(edge\ of\ domain)}}-T_{\text{ambient}})$$

The value *h* was chosen so as to minimize the root mean squared discrepancy between modeled and measured temperatures for the 32 laboratory conditions tested. These adjustments changed the modeled outlet fluid temperatures by an average of 2.3°C in the relatively hot desiccant, an average 0.6°C in the exiting water temperature and an average of 0.2°C change in the leaving air temperature. Instruments used to measure the temperature of the three fluids were accurate to 0.3°C, 0.3°C, and 0.4°C for the desiccant, water, and air, respectively. These adjustments correspond to physical processes that do occur in reality and have very little bearing on the analyses that have been conducted with this model, but were necessary to completely capture the operation of the absorber.

When the heat transfer coefficient mentioned previously was employed, modeled exit temperatures matched well with measured temperatures, as shown in Figure C6. The error bars in Figure C6 represent the bounds of the precision of the measuring instruments.

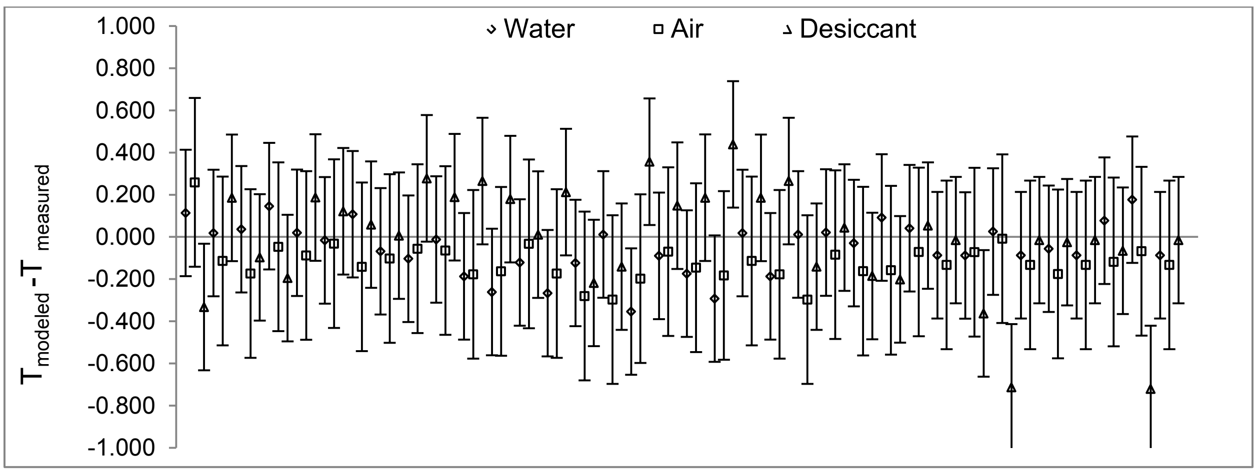


Figure C6. Discrepancy between modeled and measured outlet temperatures of three fluids. Error bars represent precision of measuring instruments.

C.3.1 Regenerator Model

An empirical model of the performance of a parallel plate regenerator made from laboratory data was used for the regenerator. This model consists of five equations relating performance to inlet variables. The first three equations predict measured performance with coefficients of determination (R^2) of 0.99, 0.98, and 0.98, respectively, and the last two equations ensure conservation of mass and energy.

C.4 Interior Direct Expansion Dehumidifier

This system has been tested at NREL and the technical report is available (Christensen and Winkler 2009). This report contains six polynomial equations predicting the power consumption and dehumidification performance of the dehumidifier as measured in the laboratory. These equations are implemented directly into the Modelica model of the dehumidifier used in the current study. The six equations predict the measured performance with coefficients of determination (R^2) of 0.998, 0.987, 0.999, 0.999, 0.998, and 0.998, respectively. For more information, please see (Christensen and Winkler 2009).

C.5 Condenser Heat-Regenerated Desiccant Wheel Dedicated Outdoor Air System (Solid Desiccant Dedicated Outdoor Air System)

This system is a commercially available product that is included in the manufacturer's selection software. In order to model this system, we used the manufacturer's software to generate a performance map that covered all inlet conditions to be expected in the current study. This performance map also considered an input of the number of compressors in operation. We obtained the control strategy that dictated this variable from the manufacturer and implemented it into the model. This strategy inputs room conditions and outdoor conditions and outputs the number of compressors in operation. The following graphs show the comparison between the model developed for the purposes of the current study and the manufacturer's software for compressor power, fan power, moisture removal rate, and temperature change over the entire system. Because the desiccant wheel is regenerated with condenser heat only, there is no additional energy used to regenerate the desiccant and the performance is completely defined by these four variables.

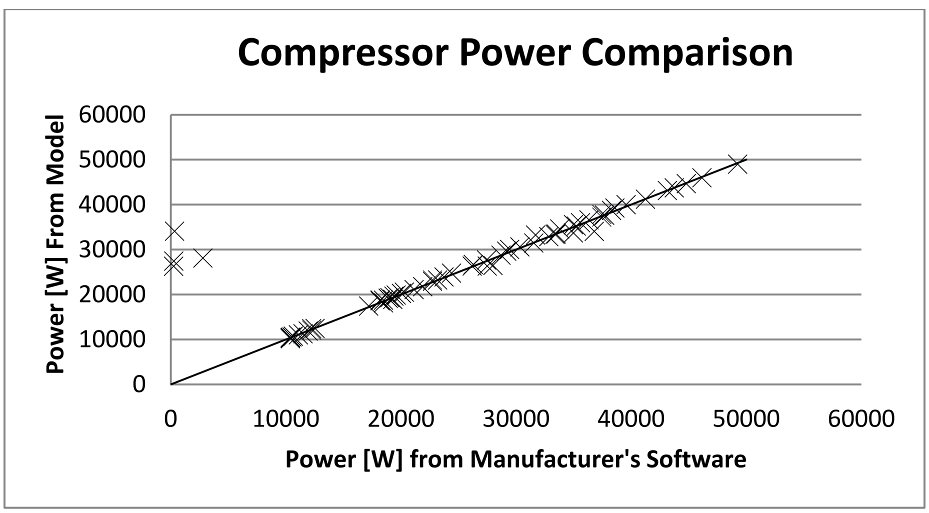


Figure C7. Comparison between manufacturer's software and model prediction of compressor power

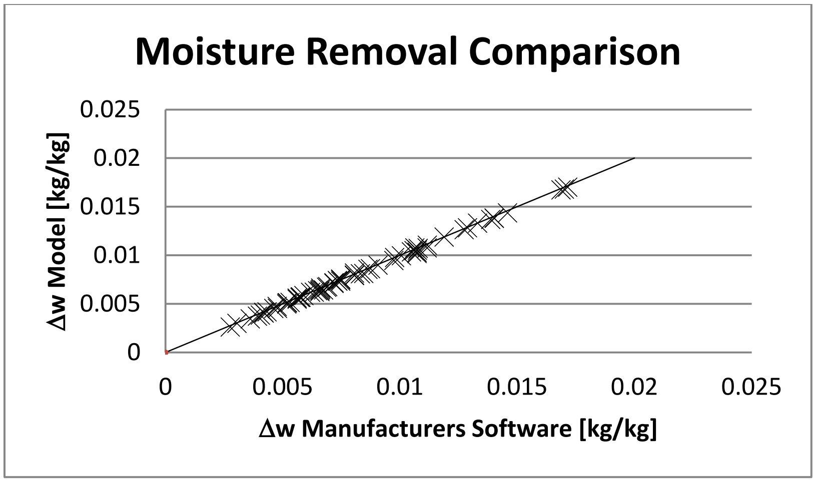


Figure C8. Comparison between manufacturer's software and model prediction of moisture removal

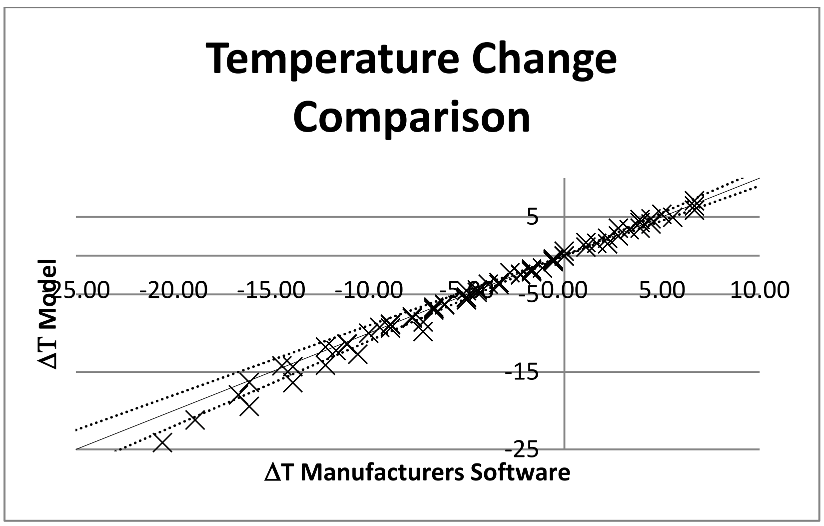


Figure C9. Comparison between manufacturer's software and model prediction of temperature change across system

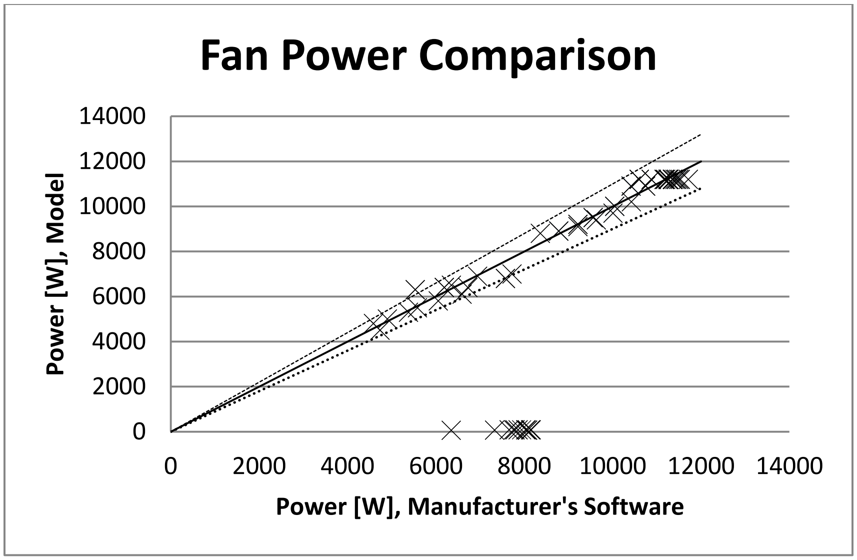


Figure C10. Comparison between manufacturer's software and model prediction of fan power

Appendix D. EnergyPlus-Modelica Comparison

This study employed three different simulation programs. While we made every effort to ensure that they produced comparable results and we made a successful comparison between EnergyPlus and Modelica, some discrepancies might be present between the various simulations. A third simulation platform was used for the system energy calculations of the Variable Capacity DX System and Adaptive Multi-Path System. This platform is proprietary and developed by the system's manufacturer. Because this simulation program was more specialized than the other two programs used, was tailored to represent the Adaptive Multi-Path System, and did not affect the refrigeration energy, it was used in place of the models built from manufacturer software and empirical models used for other systems.

A few important distinctions exist between simulations done with Modelica and those done with EnergyPlus. The software that implements the Modelica language uses a fully dynamic solver that assumes the vast majority of the problem variables are continuously varying. EnergyPlus, on the other hand, uses a quasi-steady state assumption over each time step. This results in a few differences:

- Modelica requires a time constant to be specified for each component. This is due to the fact that the objects, including cooling coils, are modeled as continuously stirred reactors with a finite volume and flow rate passing through, with an energy and moisture input that accounts for the effects of the cooling coil on the airstream. Selection of this time constant is nontrivial because a too-small constant will cause a stiff problem and instabilities in the solution algorithm, and a large time constant will cause nonphysical delays in the cooled air moving down the duct to the building. None of this is an issue in EnergyPlus because of the quasi-steady state assumption. Time constants were chosen to most accurately predict performance while providing for a stable problem.
- When reheat is important, as in supermarkets, the timing of when the reheat can be used is an issue. In Modelica, reheat is only available when the cooling coil is in operation, as in real-world operation. In EnergyPlus, as long as there is some cooling during a certain time step, 30% of the cooling energy used in the time step is available for reheat even if the run-time fraction for the time step was small and the heating is not necessarily needed at the same time as the cooling. This discrepancy was alleviated in large part by adding reheat energy at all times when cooling coils were in operation, which was what EnergyPlus effectively did. However, some small discrepancies may exist.
- The start-up/shut-down inefficiencies are not accounted for in the Modelica models as they currently exist. This is because there is no part-load factor because there is no run-time fraction because of the different solution algorithm. This omission introduces only a small error into the simulations.

In order to validate the method used to model the HVAC systems in Modelica, we modeled the baseline system in both EnergyPlus and Modelica for a supermarket in Miami and compared the outputs. It is impossible to model the system in Modelica in a way that is identical to EnergyPlus. This is true because the EnergyPlus algorithms used to calculate DX coil operation, reheat energy added to the air loop, and furnace gas energy all employ a runtime fraction approach to account for situations in which the coil does not need to run at full capacity for the entire EnergyPlus time step. EnergyPlus pre-calculates the percentage of the timestep during which the coil(s) must be in operation, based on the loads calculated for the space. This is done to expedite the solution process and alleviate the need for a fully dynamic solution algorithm. In contrast, Modelica models the dynamics of the physical process, including coils turning on and off in response to space temperatures and humidity. This method more closely approximates the functioning of the actual system control, which has no knowledge of loads in the space, but only responds to signals from a thermostat and a humidistat.

The following describes the control strategy we implemented in Modelica:

- Modeled a two-stage DX coil with a 50/50 capacity split
- For sensible cooling, the first stage turns on when space DBT is 0.5°F above cooling SP. If the space DBT continues to rise, the second stage turns on at 1.5°F above SP.
- As the space cools, the second stage turns off when the space is 0.5°F above SP; the first stage turns off at 0.5°F below SP.
- For dehumidification, the first stage turns on when space DPT is at 50.5°F and the second stage when space DPT is at 51.5°F. The second stage turns off when the DPT is at 50.5°F; the first stage turns off when the DPT is at 49.5°F. In all simulations, the space DPT never reaches the threshold for the second stage to be activated; therefore, only the first stage is used for dehumidification.
- When in dehumidification mode, the desuperheater reheat coil turns on at 30% of (energy removed by the DX coil + compressor power) when the space DBT is 0.5°F below the heating SP and kicks off when the DBT is 0.5°F above the heating SP.
- The gas heater turns on, delivering 95°F supply air, at 0.5°F below the heating SP and off when the DBT goes above 0.5°F above SP.

This strategy results in three unavoidable discrepancies between EnergyPlus and Modelica. First, Modelica must be supplied with a control algorithm that includes a deadband around SP temperatures and humidity that determine the cycling on and off of the coils. Therefore, SP temperatures are not met precisely, but rather space temperatures fluctuate between the SP plus one half of the deadband and the SP minus one half of the deadband. As is shown subsequently, this results in very little annual discrepancy between the two programs. Second, the operation of the desuperheater used to reheat air during dehumidification in EnergyPlus is affected somewhat by the solution algorithm. In EnergyPlus, the amount of reheat added by the desuperheater is a function of the runtime fraction of the DX coil during the time step. Because the solution algorithm is not fully dynamic, the amount of reheat available may be more or less than what is shown in a fully dynamic solution, but the discrepancy is minimal, as shown in Figures D1 and D2. Lastly, because of the dynamic solution algorithm used in Modelica, each coil requires a time constant, which determines the time it takes to change the temperature of the air leaving the coil. If this time constant is too large, a "lag" in the leaving air temperature occurs, whereby the air leaving the coil continues to be affected by coil operation even after the coil shuts off. If the time constant is too small, the numerical problem becomes very stiff and the solution will not converge. EnergyPlus, in contrast, models quasi-steady state coils and this issue does not arise. We took steps to minimize this effect and find a good time constant.

The outputs for the baseline system from both EnergyPlus and Modelica are displayed in Figures D1 and D2. The cumulative amount of electricity supplied to the compressor, pump, and condenser fan for the DX Coil over 1 year in Miami is also shown. As can be seen, the two simulations provided very comparable results. Total annual electricity use as calculated in Modelica is within 2.5% of the EnergyPlus prediction.

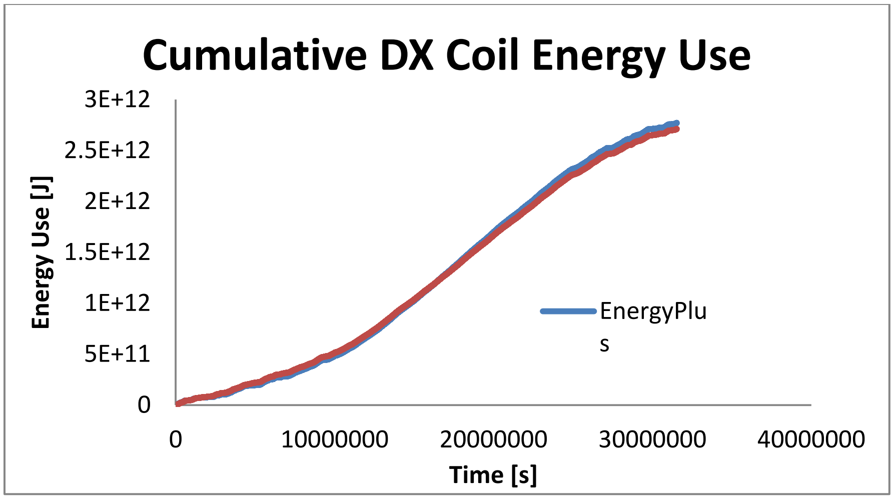


Figure D1. Cumulative DX coil electricity modeled with two programs

The comparison of gas use between the two programs is shown in Figure D2. It can be seen that the gas used in the summer time, which is almost exclusively used for the purposes of reheat during a dehumidification call, is somewhat greater in the EnergyPlus simulation than in the Modelica simulation. This is due to the differences between the solution algorithms discussed previously. However, annual gas used for Miami in Modelica is within 10% of the EnergyPlus prediction.

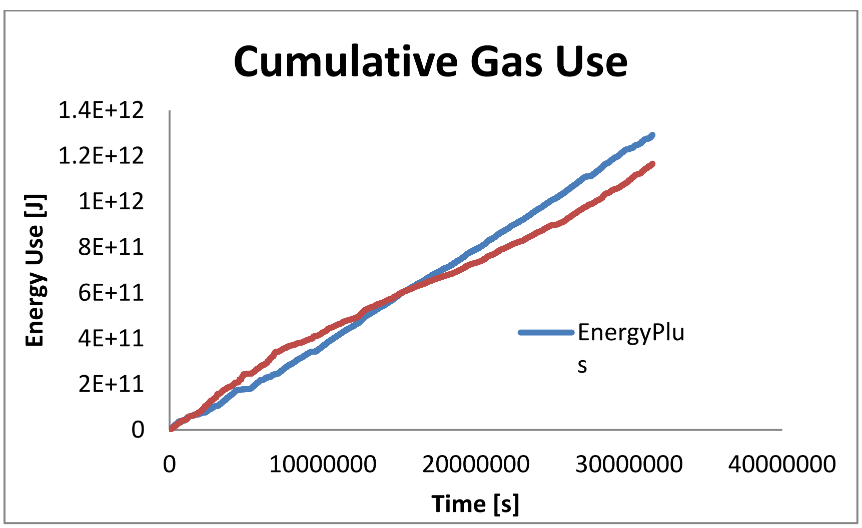


Figure D2. Gas use modeled with two simulation programs

In Figure D3, modeled supply fan power use over a year in Miami is shown for the two simulation programs. As can be seen, there is close agreement between the two programs.

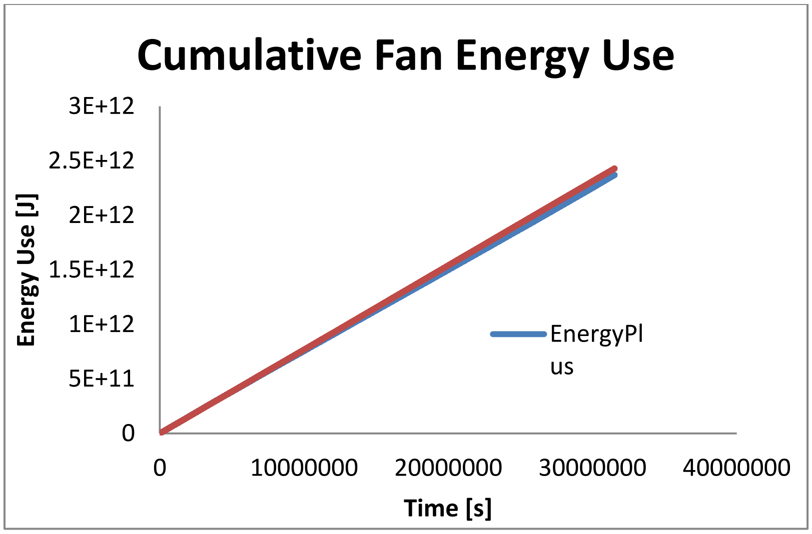


Figure D3. Supply fan power use modeled with two simulation programs

In Figure D4, the space temperature control for typical operation is shown. Figure D4 shows that the Modelica simulation can maintain temperatures between the heating SP and cooling SP with relatively good precision.

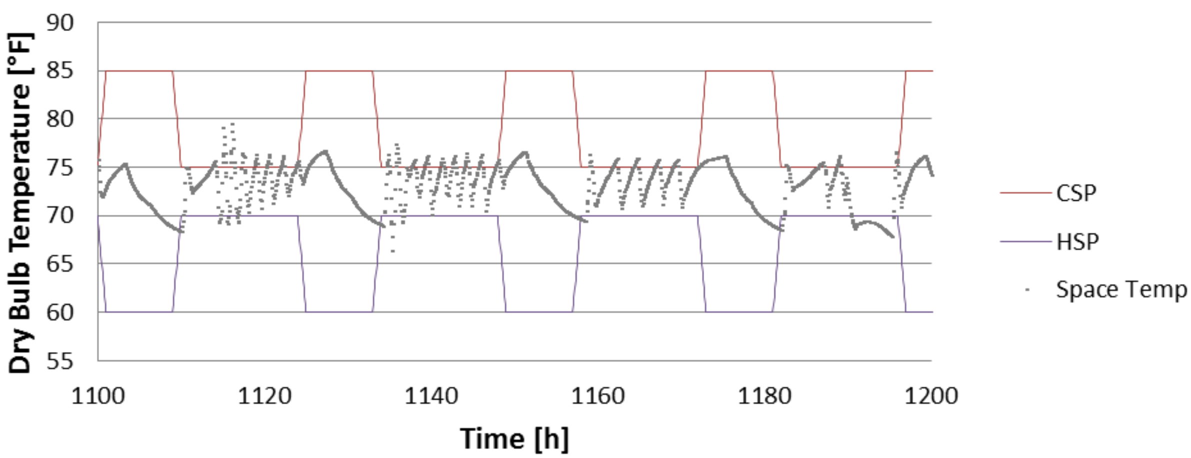


Figure D4. Typical control of space DBT as modeled with Modelica, showing cooling SP and heating SP with night setback and fluctuation of space temperature

Lastly, in Figure D5, the space DPT control over the course of the year in Miami is shown. It can be seen that Modelica is able to maintain space humidity within the deadband of the DPT SP. Again, this differs slightly from EnergyPlus outputs, because the EnergyPlus algorithm maintains the DPT SP exactly, based on pre-calculated moisture loads at each time step.

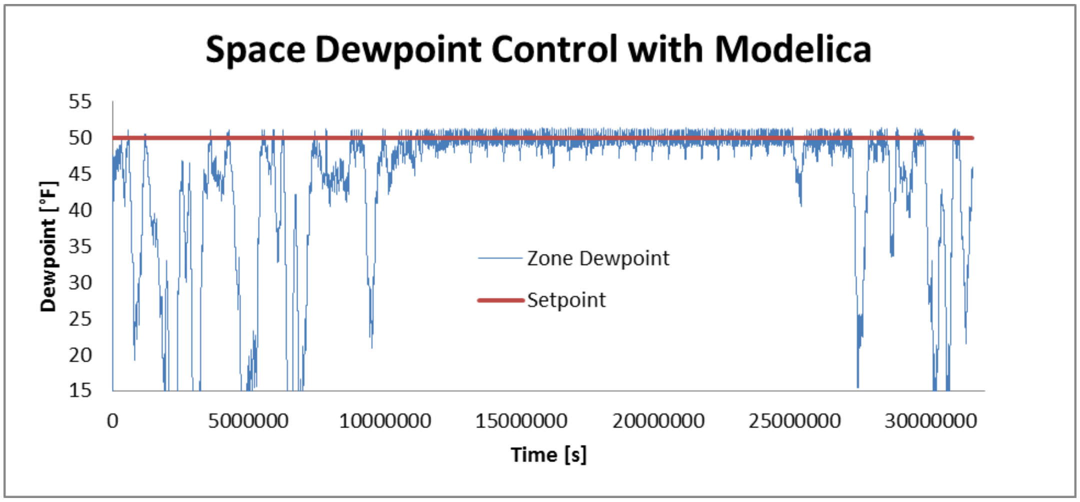


Figure D5. Space DPT throughout the year as predicted by Modelica simulation

Modelica was able to closely approximate the modeling of energy consumption and control of space conditions shown in EnergyPlus. The candidates for alternative cooling and dehumidification are modeled and their outputs are compared to Modelica outputs of the baseline to determine their potential benefits.

Appendix E. Advanced Dehumidification Strategies Not Used

Because of time constraints and lack of applicability in some cases, a few systems that may be beneficial in supermarket HVAC were not included in this study. The paragraphs below discuss a few of these systems.

E.1 Direct Expansion System With Wrap-Around Heat Pipes

When a DX-based system is used for dehumidification, air must be cooled well below its DPT in order to remove moisture. This approach results in a relatively cold exiting temperature, which is either delivered to the space directly, possibly causing thermal comfort problems, or is reheated, often using additional energy, in order to meet the space SHR. This problem is especially acute in supermarket applications in which large refrigerated cases are present in the zone being conditioned, because the space SHR in these zones can approach 0 or even be negative, while the baseline system SHR often cannot go below 0.7 without additional reheat energy. One way of combating this problem is by using a passive WAHP (shown with its corresponding psychrometric processes in Figure E1), which first precools the air before the DX coil, and then reheats the air after the coil with no additional energy input other than the fan power required to move the air through the heat exchanger. This method shifts the system SHR toward that needed in low-SHR applications.

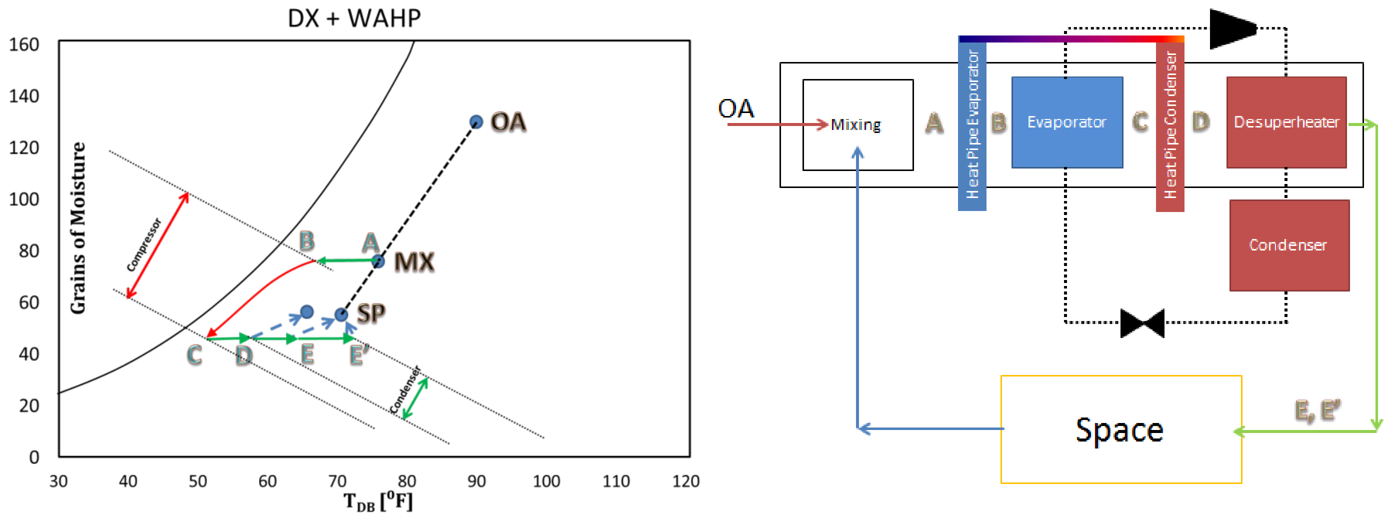


Figure E1. WAHP system schematic and psychrometric processes

An efficient way of lowering the SHR is with a heat pipe heat exchanger. This concept has been discussed since 1939 (Gatley 2000). A heat pipe is a passive device with no moving parts in which refrigerant with a boiling point below that of the entering air is boiled in the heat pipe evaporator, travels by gravity or by capillary action to the heat pipe condenser and is condensed there. This process removes the latent heat of vaporization of the refrigerant from the air before it enters the DX coil and adds it back to the air after it is dehumidified in the DX coil. This process can work over long distances, with small temperature differences (Yau and Ahmadzadehtalatapeh 2010). The heat pipe usually has a long life, requires little maintenance, and has no moving parts (Cooper 1996) and no external connections (Gatley 2000).

The WAHP system offers several advantages over the baseline. By removing a large portion of the sensible load on the DX coil, the heat pipe can increase the latent capacity (Yau and Ahmadzadehtalatapeh 2010; Keebaugh et al. 2004), reduce the need for reheat, and change the SHR of the system (Yau and Ahmadzadehtalatapeh 2010). In a 1992 study, heat pipe retrofits were shown to reduce the needed capacity by 14%–23% and fan power was actually reduced, owing to the smaller DX coil needed (Flannick 1992). The SHR of the system can be shifted from 0.75 to 0.5 at rating conditions with minimal loss in system COP (Kosar et al. 2007). Alternatively, a designer may choose to downsize the DX system and eliminate active reheating, because some of the sensible cooling is performed by the heat pipe evaporator and reheat by the heat pipe condenser (Cooper 1996). These benefits are especially pronounced in hot-humid climates, where heat pipes were shown to have less than a 1-year payback period (Mathur 1996). Humidity control is also improved by the shift in load from sensible to latent (Witte and Henninger 2006).

WAHPs do include some potential disadvantages. Because the device is completely passive, control is very difficult (Gatley 2000). Some control can be achieved by tilting a gravity-driven heat pipe, but this may not be cost-effective. Installation also must be done very precisely in order for the heat pipe to function properly (Gatley 2000). Adding the heat pipe also results in an additional pressure drop, albeit not as large as when other types of heat exchangers are added. This addition may result in higher power usage if the DX system is not down-sized when the heat pipe is added (Witte and Henninger 2006). One study found that WADWs and dual-path systems offered better humidity control (Witte and Henninger 2006). To the author's knowledge, two commercially available systems utilize the wrap-around heat exchanger configuration.

E.2 Wrap Around Desiccant Wheel (Cromer Cycle)

A somewhat newer means of gaining the benefits of the WAHP is the WADW configuration, or the Cromer Cycle, shown in Figure E2. Like the WAHP configuration, the Cromer Cycle can lower the SHR of the system by first passively cooling and humidifying air before it reaches the coil, then allowing the coil to remove more moisture than the baseline, before dehumidifying more deeply after the coil by means of a desiccant wheel. This dehumidification is usually done with a Type III desiccant wheel, which transfers humidity effectively at RH above 85% (Cromer 2000). This configuration allows for DPTs exiting the system to be 2°F –5°F lower than those exiting the baseline system (Cromer 2000). These often approach DPTs possible with actively regenerated desiccants (Kosar et al. 2007). Inclusion of WADW also allows for a higher evaporator temperature by moving the DPT lower chemically with the desiccant. The required reheat energy is also reduced by adding the heat of vaporization in the post-coil portion of the desiccant wheel. Unlike the WAHP, little is known about the performance of the WADW configuration, because the concept was just introduced 14 years ago (Cromer 2000). A ScienceDirect search for Cromer Cycle returns no results at the current time.

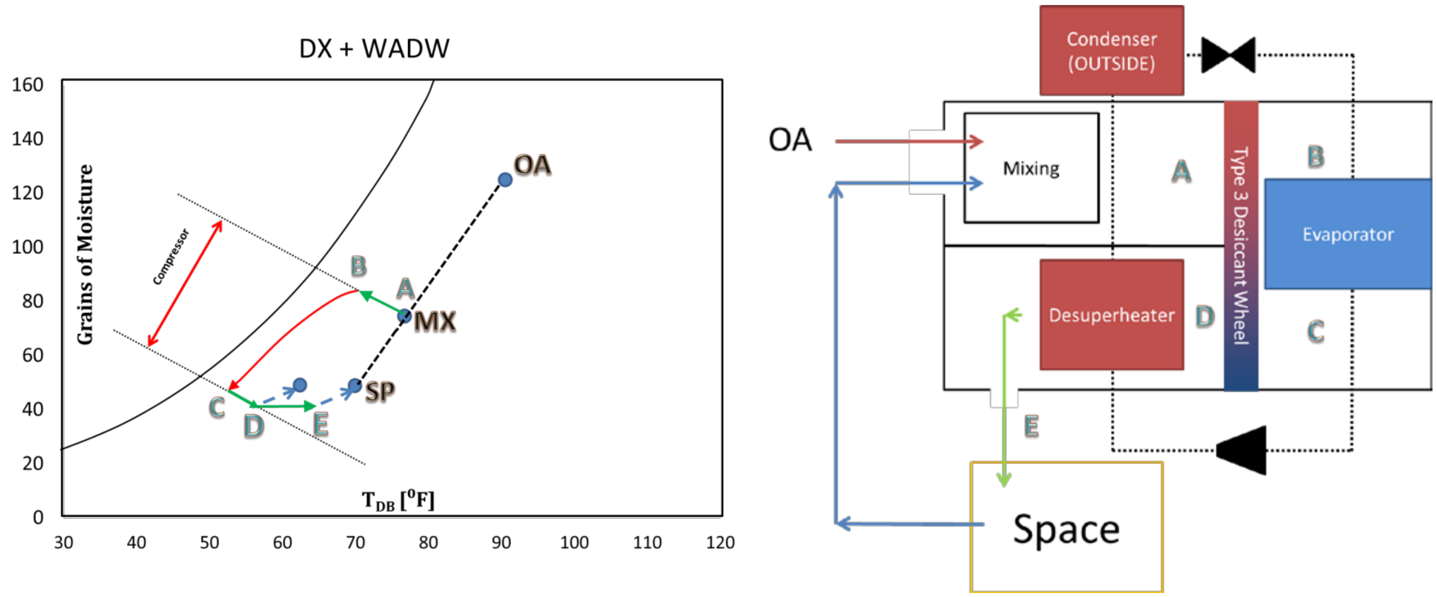


Figure E2. WADW system schematic and psychrometric processes

Several benefits of the Cromer Cycle configuration have been suggested. Because the desiccant wheel provides some dehumidification and evaporator temperature is increased, a greater capacity can be achieved at the same input power (Cromer 2000; Trane 2010). A 24% increase in capacity at design conditions has been calculated (Cromer 2000). Alternatively, a designer may choose to downsize the coil 15%–35% when the WADW is added. This results in lower energy use in the compressor (Cromer 2000; Trane 2010); a 10%–30% reduction has been reported (Trane 2008). The SHR and moisture removal capacity of the system are also altered with the addition of the Cromer Wheel. At design conditions, a 153% increase in moisture removal and shifting of the SHR from 0.74 to 0.47 is possible (Cromer 2000).

The Cromer Cycle also offers ancillary benefits. Control of the system is easy, because simply adjusting the speed of the wheel controls the amount of moisture transferred by the wheel. (Cromer 2000). No EA is needed for regeneration, which is helpful in applications, such as supermarkets, where quantities of EA available for regeneration may be insufficient (Trane 2010). Supply air is also moved away from the saturation line where it is supplied in the baseline system, reducing condensation and mold in ducts and terminal units (Cromer 2000). No refrigerant needs to be charged for the wheel, and the desiccant material will not freeze under cold conditions (Gatley 2000).

Some criticisms of the Cromer Cycle have been raised. First, in the Cromer Cycle, air must pass through a desiccant wheel with a relatively large pressure drop twice, resulting in higher fan power (Trane 2010). This configuration often results in a full system COP lower than that of the WAHP configuration (Kosar et al. 2007). The Cromer Wheel also dictates the configuration of the air handling unit, requiring the air to make a 180° turn in the middle of the air handling unit. Additional maintenance of the desiccant wheel is required and degradation from leaking seals, clogging, and reaction on the desiccant surface can occur (Gatley 2000). The Cromer Cycle has also been criticized for its inability to reduce the system SHR, compared to actively regenerated desiccant systems (Kosar et al. 2007)

E.3 Direct Expansion Dedicated Outdoor Air System With Wrap-Around Heat Pipe

This strategy is identical to the WAHP strategy discusses previously, except that the heat pipe system conditions 100% outdoor air and a parallel sensible device is operated, which is controlled by the thermostat in the space. Fricke and Sharma (2011) reported a 42% increase in dehumidification efficiency over the baseline with this strategy. The system is shown in Figure E3.

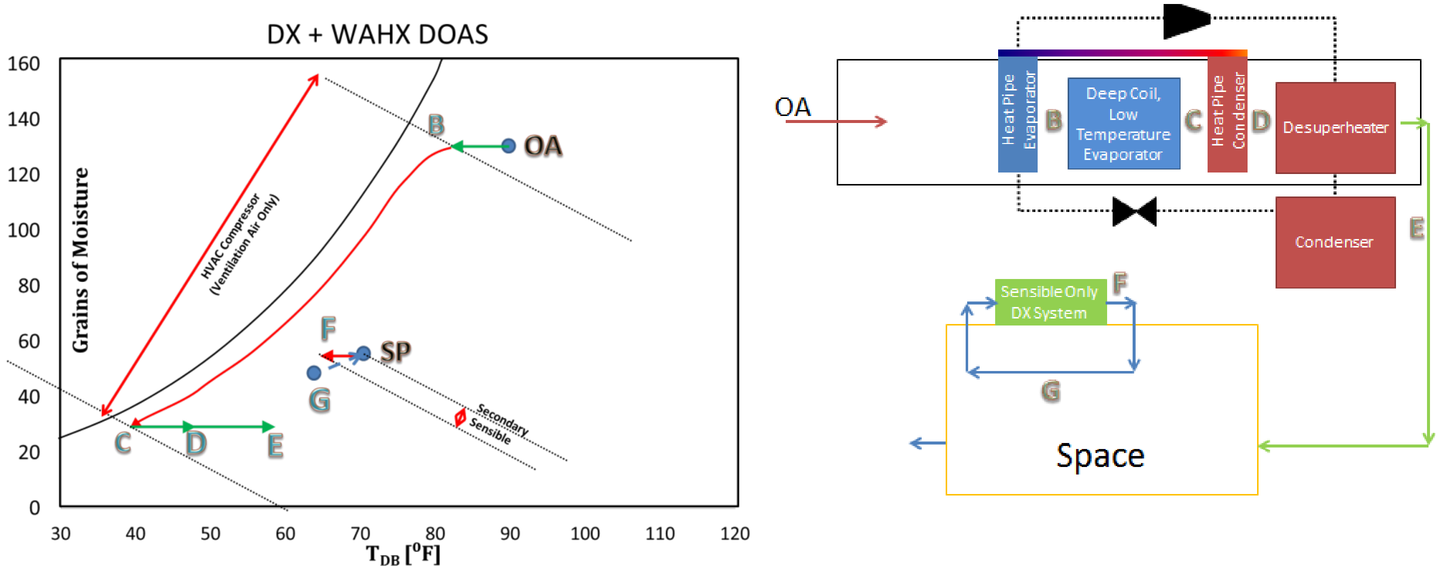


Figure E3. WAHP DOAS psychrometric processes and system schematic

E.4 Direct Expansion Dedicated Outdoor Air System With Wrap-Around Desiccant Wheel

This strategy, shown in Figure E4, is identical to the WADW strategy, except that the WADW system conditions 100% OA; again, a parallel device is operated to provide sensible cooling only. Also, a condenser desuperheater is often installed upstream of the first desiccant wheel section in order to preheat the air entering the wheel. This increases the driving force for moisture transfer and allows dried air to be delivered (Mumma 2007). If possible, this can be eliminated with a total enthalpy wheel preconditioner. This strategy is billed as the number one most promising for DOAS systems (Mumma 2007) when used with a total enthalpy recovery wheel.

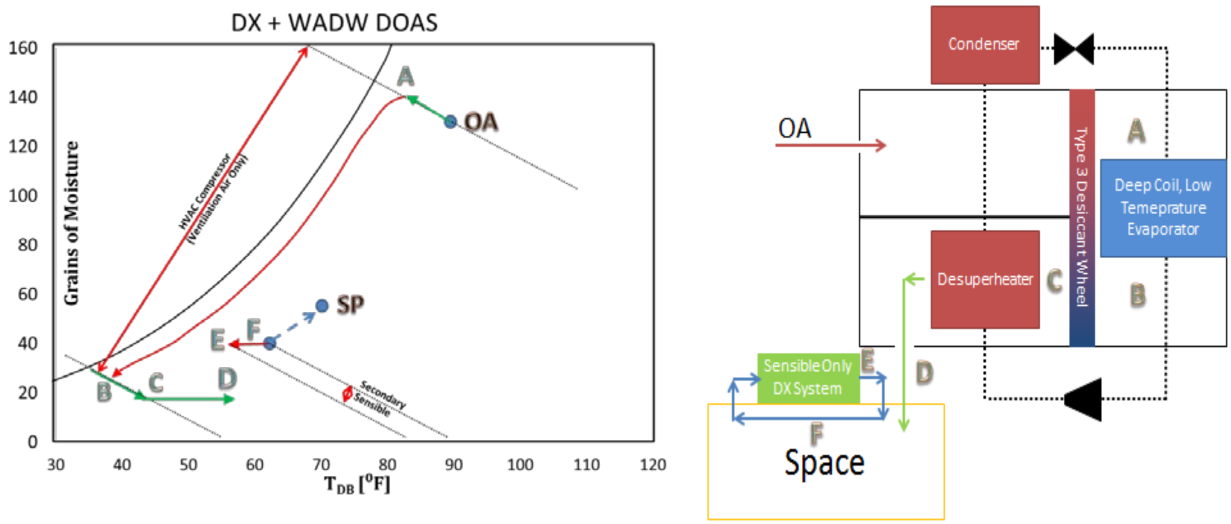


Figure E4. Psychrometric processes and system schematic for WADW DOAS system