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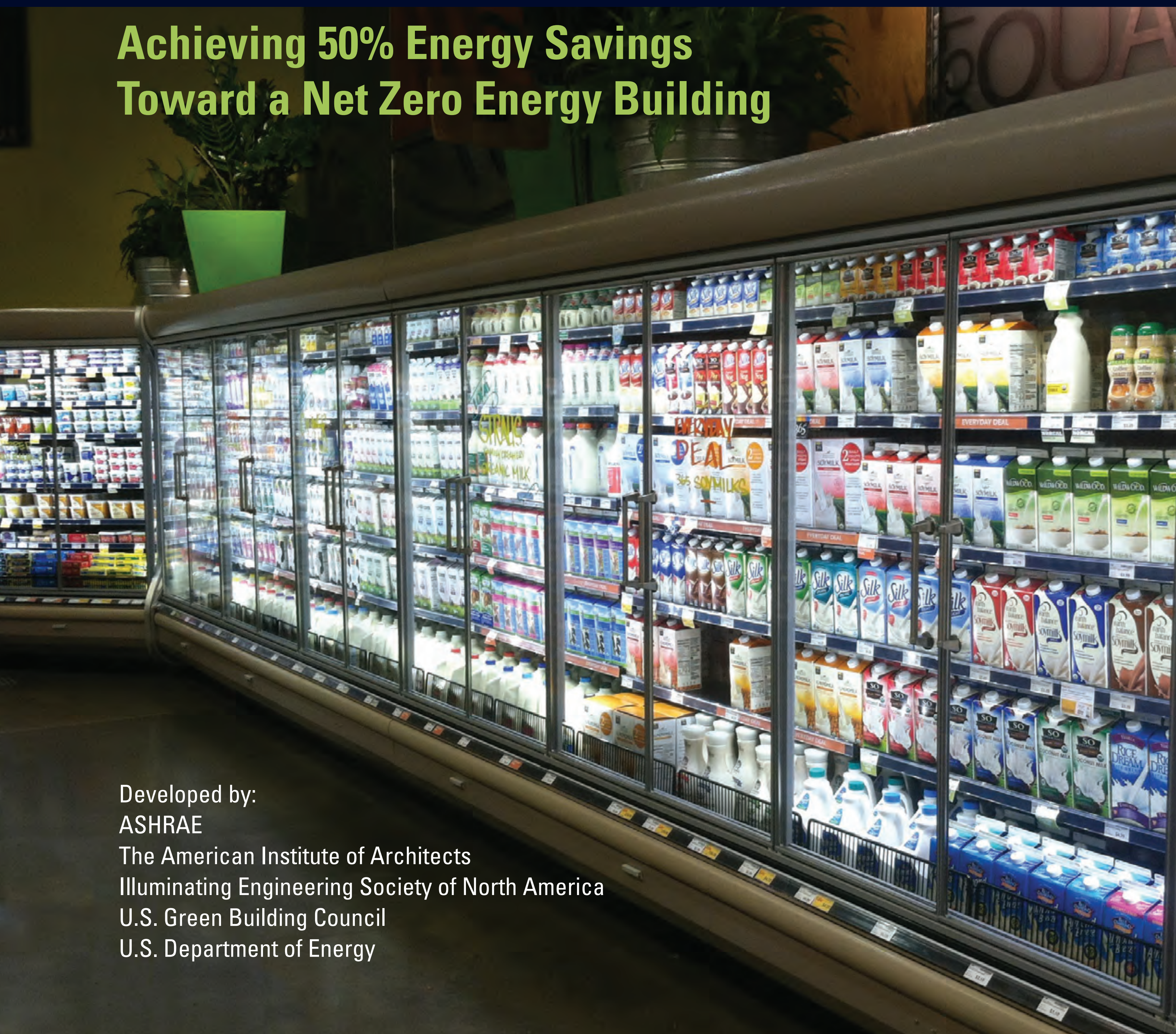
Advanced Energy Design Guide for Grocery Stores



Achieving 50% Energy Savings Toward a Net Zero Energy Building

Developed by:
ASHRAE
The American Institute of Architects
Illuminating Engineering Society of North America
U.S. Green Building Council
U.S. Department of Energy

Advanced Energy Design Guide for Grocery Stores

This is an ASHRAE Design Guide. Design Guides are developed under ASHRAE's Special Publication procedures and are not consensus documents. This document is an application manual that provides voluntary recommendations for consideration in achieving greater levels of energy savings relative to minimum standards.

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Advanced Energy Design Guide for Grocery Stores

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Acknowledgments

Advanced Energy Design Guide for Grocery Stores is the fifth in a series of publications designed to provide strategies and recommendations for achieving 50% energy savings over the minimum code requirements of ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings*. Grocery stores are a critical part of the economy and part of a larger group of closely related buildings that include convenience stores and larger retail establishments with food sales. This Guide is the result of the dedicated, collective efforts of many professionals who devoted countless hours to develop guidance that will help grocery stores use less energy.

The primary authors were the 12 members of the ASHRAE Special Project 138 committee who represented the participating organizations—ASHRAE, The American Institute of Architects (AIA), U.S. Green Building Council (USGBC), Illuminating Engineering Society of North America (IES), and U.S. Department of Energy (DOE).

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The project committee's efforts were guided by the AEDG Steering Committee, composed of members from the partner organizations—ASHRAE, AIA, USGBC, and IES—with the additional support and participation of DOE. Its members provided direction and guidance to complete development of the Guide within 12 months. The Steering Committee assembled an expert team of authors and defined a scope that kept the project committee's task manageable and focused. The representatives from these organizations provided a broad prospective and challenged the project committee to bring advanced grocery store design to a solution that could be achieved by all.

In addition to the voting members of the project committee, a number of individuals played key roles in the Guide's success. Specifically, thanks to the ASHRAE and NREL meetings staff for serving as hosts.

Twenty-nine people participated in two peer reviews, providing more than 420 remarks that helped strengthen and clarify the Guide. Their thoughtful input helped steer the finished product and ultimately its impact in the marketplace.

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Paul Torcellini
Chair, Special Project 138

February 2015

Abbreviations and Acronyms

ACCA	Air Conditioning Contractors of America
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
AIA	The American Institute of Architects
ASTM	ASTM International
ANSI	American National Standards Institute
BAS	building automation system
BAU	business as usual
BoD	Basis of Design
Btu	British thermal unit
CD	construction document
c.i.	continuous insulation
Cx	commissioning
CxA	commissioning authority
CFL	compact fluorescent
cfm	cubic feet per minute
CMH	ceramic metal halide
CO_2	carbon dioxide
COP	coefficient of performance, dimensionless
CV	constant volume
db	dry bulb
DCKV	demand-controlled kitchen ventilation
DCV	demand-controlled ventilation
DL	Advanced Energy Design Guide code for “daylighting”
DOAS	dedicated outdoor air system
DOE	U.S. Department of Energy
DX	direct expansion
E_c	combustion efficiency, dimensionless
E_t	thermal efficiency, dimensionless
EC	electronically commutated
ECM	energy conservation measure
EER	energy efficiency ratio, Btu/W·h
EEV	electronic expansion valve
EF	energy factor
EL	Advanced Energy Design Guide code for “electric lighting”
EN	Advanced Energy Design Guide code for “envelope”
ESP	external static pressure
EUI	energy use intensity

fc	footcandle
FC	filled cavity
FPI	fins per inch
Guide	*Advanced Energy Design Guide for Grocery Stores*
GWP	global warming potential
HC	heat capacity, Btu/(ft^2·°F)
HID	high-intensity discharge
HV	Advanced Energy Design Guide code for “HVAC systems and equipment”
HVAC	heating, ventilating, and air-conditioning
IAQ	indoor air quality
IEER	integrated energy efficiency ratio
IES, IESNA	Illuminating Engineering Society of North America
in.	inch
IPLV	integrated part-load value, dimensionless
kBtu/h	thousands of British thermal units per hour
KE	Advanced Energy Design Guide code for “kitchen equipment”
kW	kilowatt
kWh	kilowatt-hour
LCCA	life-cycle cost analysis
LED	light-emitting diode
lf	linear feet
LPD	lighting power density, W/ft^2
LPW	lumens per watt
Ls	liner system
M&V	measurement and verification
MA	mixed air
NEMA	National Electrical Manufacturers Association
NFRC	National Fenestration Rating Council
NREL	National Renewable Energy Laboratory
O&M	operation and maintenance
OPR	Owner’s Project Requirements
PL	Advanced Energy Design Guide code for “plug loads”
ppm	parts per million
PWM	pulse width modulating
QA	quality assurance; Advanced Energy Design Guide code for “quality assurance”
RE	Advanced Energy Design Guide code for “renewable energy”
RF	Advanced Energy Design Guide code for “refrigeration”
RTU	rooftop unit; rooftop air-conditioning unit
SAT	supply air temperature
SCT	saturated condensing temperature
SEER	seasonal energy efficiency ratio, Btu/W·h
SET	saturated evaporator temperature
SHGC	solar heat gain coefficient, dimensionless
SRI	Solar Reflectance Index, dimensionless
SST	saturated suction temperature
SWH	service water heating
SZCV	single-zone constant volume
SZVAV	single-zone variable air volume
TD	temperature difference (also referred to as *approach*)
USGBC	U.S. Green Building Council
VAV	variable air volume
VOC	volatile organic compound
VSD	variable-speed drive
VT	visible transmittance
W	watts
wb	wet bulb
in. w.c.	inches of water column
WH	Advanced Energy Design Guide code for “service water heating”
WSHP	water-source heat pump

Foreword: A Message for Building Owners and Developers

Advanced Energy Design Guide for Grocery Stores is a continuation of the series of Advanced Energy Design Guide (AEDG) publications designed to provide recommendations to achieve 50% energy savings when compared with the minimum code requirements of ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings* (ASHRAE 2004). This Guide applies to grocery stores with gross floor areas between 25,000 and 65,000 ft^2 with medium- and low-temperature refrigerated cases and walk-ins; however, many of the recommendations may also be applied to smaller or larger grocery stores. When combined with the 50% AEDG for Medium to Big-Box Retail Buildings, the combination can be used for larger stores that consist of both groceries and general merchandise.

Use of this Guide can help in creating a cost-effective design for new construction and major renovations of grocery store buildings that will result in buildings that consume substantially less energy compared to the minimum code-compliant design, resulting in lower operating costs. Also important is that through use of an integrated design process, an energy-efficient building offers a great possibility to enhance the shopping and working environment with respect to indoor air quality (IAQ), thermal comfort, and the visual effects of merchandise display.

For successful designs of "brand right" energy-efficient buildings, owners and designers must consider the following as they collaborate on the designs for their buildings:

- Meets all brand requirements of the food retailer.
- Creates a healthy and inviting indoor environment.
- Addresses where energy is used.
- Minimizes oversizing and unnecessary redundancy to reduce capital costs.
- Creates an appealing environment for the effective display of merchandise.
- Uses construction practices such as low-emitting materials and oversight to avoid moisture intrusion.
- Ensures operations will maintain clean, dry buildings with reduced sources of contaminants to help ensure good IAQ.
- Optimizes the ventilation requirements through a performance-based approach such as demand control or the IAQ Procedure of ASHRAE Standard 62.1-2013 (ASHRAE 2013) or meets other functional requirements such as makeup air for exhaust.
- Allows for downsized HVAC systems due to better envelope, optimized ventilation, more efficient lighting, and reduced and better-managed miscellaneous electric loads.

- Provides ongoing monitoring and oversight to ensure operational efficiency is maintained.
- Leverages successful efficiency measures into chain-wide rollouts to drive down existing building energy use.

Many food retailers are in the unique position of having a portfolio of buildings with similar designs across many climate zones. This portfolio of existing buildings can provide the basis to understand where energy is used and where opportunities exist for high-performance improvements. Focusing on the design of prototypical grocery stores allows best practices to be applied to site-specific building locations across the chain. One way to use this Guide is as a reference point for construction parameters in the prototypical design, as it is easily interpreted for different climate zones.

The ability for a prototypical design to achieve and maintain 50% energy use reduction in any climate and on any site requires more than just project design team agreement on any specific set of building energy systems. Maintaining a brand is critical for food retailers, as is the need for flexibility in merchandizing. It is more difficult to significantly vary the shape or orientation of site-specific buildings to account for location-specific parameters such as impact on solar energy. To achieve the highest performance in all locations, food retailers will need to accept more flexibility in design features to take advantage of climate-specific measures that make economic sense in some areas but not others. For example, indirect evaporative cooling may make sense in hot, dry climates but may not in colder climates. When designs do vary from the prototype, close attention must be carried through implementation and operations to ensure long-term success and true high-performing operations.

Understanding the risks and rewards of design decisions is crucial to achieving high performance. This understanding could be part of the owner's internal design expertise or that of a consultant with long-term, detailed knowledge of owner's needs. In any case, a partnership must be established amongst the members of the design team to encourage calculated design risk (such as not oversizing HVAC equipment). Owners will benefit from reduced capital costs and improved energy efficiency if systems are optimally designed without excessive safety factors. Designers should communicate opportunities in high-performance design to owners along with the risks/rewards for the design. When using new technology, building owners should consider options for modifications that could be made if performance isn't as intended, particularly in early stages of adoption for the new technology.

It is also worth noting that operation and maintenance (O&M) staff need to be provided with the tools, training, and information to keep the building running at the high level of efficiency designed into the project. Without this buy-in, the efforts of the design team can be futile. This includes assumptions made concerning equipment, maintenance, calibration, and replacement of critical building systems. Ultimately, the design team should take into consideration the O&M procedures used by the building owner and ensure that system design can be effectively incorporated into standard processes or inform the owner of special training and procedures that must be used for the high-performance building. Measurement and verification of system performance of the actual building is an ongoing process and is critical to sustained high performance of the building.

This Guide presents a broad range of subject matter, including concepts such as the integrated design process, multidisciplinary design strategies, and design tips and good practices. The focus of this Guide is to provide specific recommendations that are easily implemented in order to achieve 50% energy savings.

ENHANCED SHOPPING ENVIRONMENTS AND HUMIDITY CONTROL

In addition to high energy efficiency, good design practice focuses on creating healthier building environments, including visual comfort, acceptable acoustic comfort, thermal comfort, and good IAQ for shoppers and employees. The goal of food retailers is to create an inviting environment for customers visiting for a quick grab-and-go shopping experience and for those shopping for an extended time. The focus within a grocery store is on the products being sold and avoiding conditions that negatively impact the customer or employee experience. Energy-saving designs and products should be considered in light of their impact on the occupant experience and store brand in order to be accepted by grocery store operations staff and customers.

The grocery industry has grappled with humidity control for many years and has identified only a few efficient approaches. Grocery stores are highly sensitive to humidity, as it impacts refrigerated case performance and fresh food perishability. The key is to balance latent and sensible loads. This balance becomes more important as stores become more efficient and doors are placed on refrigerated cases. System designs should first address humidity by ensuring ventilation rates are adequate for good IAQ. The design should also specify systems that adapt to changing humidity conditions and ensure that the appropriate amount of humidity is removed to maintain good refrigeration system performance and prevent product loss.

LOWER LIFE-CYCLE COSTS

In some cases, the capital cost of energy-efficient building system technology cannot be justified by the energy cost savings alone. However, energy-efficient grocery stores can cost less to build than traditional store constructions through use of an integrated design process and project delivery. For example, installing refrigerated cases with glass doors can dramatically reduce refrigeration compressor and condenser capacity requirements while lowering space-heating requirements on the sales floor. Because food retailers directly purchase many of the energy-using components of a building, the owner can include energy-efficiency features in the selection process for those components.

Driving down capital costs through good design practices and procurement efforts on top of energy cost savings and maximizing incentives will help meet energy savings goals while maintaining the fiscal responsibility required for a successful commercial building owner.

REDUCED OPERATING COSTS

Designing for energy efficiency isn't enough to ensure a building will actually save energy. Ensuring that systems are installed and operating as intended is critical. Grocery store chains can take advantage of prototypical processes to incorporate commissioning-related activities into design and through building construction and then monitor operations through centralized monitoring and control systems. It is likely that when food retailers better understand where the energy is being consumed within their stores, they will find opportunities to reduce operating costs. Measurement and verification processes will help achieve and maintain the high performance goals of the store. Learning from ongoing monitoring will also provide information valuable to allow continuous improvement in design and operation. Participation in a benchmarking program such as the U.S. Environmental Protection Agency's ENERGY STAR® Portfolio Manager (EPA 2015) or a performance rating system such as ASHRAE's Building Energy Quotient (bEQ) (ASHRAE 2015) provides an opportunity for retailers to benchmark their store energy use against other stores in their own portfolio and against other retail buildings.

PARTNERS IN THE COMMUNITY

A grocery store is an integral part of the community in which it resides. Customers and employees typically live near the store, and the operations of that building can directly impact the surrounding community. Energy-efficient buildings draw fewer resources from the community and can assist the power grid in a community through programs such as electrical demand control. Measures such as reduced building lighting during peak power demand reduces the risk that homes in the community will experience power interruption. Energy efficiency measures can also help a grocery store be a better neighbor. For example, reduced operation of parking lot lighting and building equipment when not needed reduces light and noise pollution in the surrounding neighborhood. Likewise, grocery stores are very public facing, and customers and store employees who see products such as solid-state lighting (SSL), often referred to as *light-emitting diodes* (LEDs), being used successfully in their stores will more likely be comfortable using those products in their homes and in their workplaces.

Food retailers compete for product sales, but when it comes to sustainable design and energy efficiency, there is a strong common partnership among stores to share information that will drive down the energy footprint of commercial buildings. The U.S. Department of Energy (DOE) sponsors the Better Buildings Alliance (BBA), which involves many major grocery chains coming together to share best practices in partnership with U.S. national laboratories and organizations such as ASHRAE that promote research on products and processes to drive energy efficiency into the market (EERE 2015). BBA members participated in the development and review process of this Guide to share best practices with other food retailers. The common voice of the BBA is also helping equipment manufacturers better understand needs to economically incorporate high-efficiency products into the market. Taking advantage of the varied and valuable resources offered through these alliances and industry groups can dramatically increase the ability to implement the recommendations of this Guide.

CLOSING

Energy-efficient grocery store design can add value in addition to direct expense reduction, including the ability to publicize a corporate commitment to sustainability, linking to a corporate sustainable mission, higher employee morale, and maintenance cost savings when properly implemented.

Hopefully the contents of this Guide will enable retailers to incorporate energy-efficient design practices into their buildings. It should also challenge owners and designers to look beyond recommendations in the Guide to find additional efficiency and cost savings measures that are more unique to their specific building designs and operations.

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Introduction 1

Advanced Energy Design Guide for Grocery Stores (the Guide) provides user-friendly, how-to design guidance and efficiency recommendations for grocery stores with gross floor areas between 25,000 and 65,000 ft^2 with medium- and low-temperature refrigerated cases and walk-ins. Application of the recommendations in the Guide can be expected to result in facilities that consume 50% less energy when compared to those same facilities designed to meet the minimum code requirements of ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings* (ASHRAE 2004). This Guide contains recommendations to design a low-energy-use building and is *not* a minimum code or standard. A voluntary guidance document, this Guide is intended to supplement existing codes and standards and is not intended to replace, supersede, or circumvent them. Even though several design packages are provided in the document, this Guide represents *a way*, but *not the only way*, to build energy-efficient grocery stores with 50% energy savings.

The intended audience of this Guide includes, but is not limited to, building owners, architects, design engineers, energy modelers, general contractors, facility managers, and building operations staff. Specially, Chapter 2 is written for a target audience of all design team members, whether they are design professionals, construction experts, owner representatives, or other stakeholders. Chapters 3 through 5 are intended for design professionals to pursue sound design advice and identify interdisciplinary opportunities for significant energy reduction. The focus of this Guide is to identify proven concepts that are feasible to implement and benchmark necessary energy performance criteria for 50% energy savings. The Guide requires retail leaders and design professionals to be very intentional about the goals of their project and possibly to think differently about their processes and operations.

The mission of a grocery store is to facilitate the delivery of goods and services to the public. The performance requirements of a building intended to serve these needs will be the driving force behind most design decisions for the building, and the benefits of some energy-saving measures could compromise the fundamental goal. The energy-saving measures in this Guide are intended to complement, or at least to avoid compromising, the main goal of these buildings.

The energy savings projections of this Guide are based on site energy consumption rather than source energy. *Site energy* refers to the number of units of energy consumed on the site and typically metered at the property line. *Source energy* takes into account the efficiency with which raw materials are converted into energy and transmitted to the site and refers to the total amount of energy originally embodied in the raw materials. For example, it is generally accepted that site electrical energy is 100% efficient, but in fact it takes approximately 3 kWh of total energy to produce and deliver 1 kWh to the customer because the production and distribution of electrical energy is roughly 33% efficient.

This Guide was developed by a project committee that represents a diverse group of professionals and practitioners. Guidance and support were provided through a collaboration of ASHRAE, The American Institute of Architects (AIA), Illuminating Engineering Society of North America (IES), U.S. Green Building Council (USGBC), and the U.S. Department of Energy (DOE). In essence, this Guide provides design teams a methodology for achieving energy savings goals that are financially feasible, operationally workable, and otherwise readily achievable. Because technology to conserve and generate energy is growing rapidly, it is clear that innovation is an important ingredient to the success of reducing energy consumption in retail facilities, and it is the hope of the authors that this publication will expose other existing best practices and lead to new concepts.

GOAL OF THIS GUIDE

This Guide strives to provide guidance and recommendations to reduce the total energy use in grocery stores by 50% or more, on a site energy basis, using a building that complies with ASHRAE/IESNA Standard 90.1-2004 as the minimum code-compliant baseline (ASHRAE 2004). The energy savings goal is to be achieved in each climate location rather than at an aggregated national average. The 50% savings is determined based on whole-building site energy savings, which includes process and plug loads.

SCOPE

This Guide is intended for grocery stores ranging in size from 25,000 to 65,000 ft^2 with medium- and low-temperature refrigerated cases and walk-ins, but it also applies to smaller or larger stores with similar space types. Space types covered by the Guide include dry goods, meat/dairy, produce, deli, bakery, restrooms, mechanical rooms, meeting/break rooms, offices, corridors, vestibules, and back-of-house storage, including receiving areas. This Guide does not cover parking garages, campus utilities such as chilled water and steam, water use, or sewage disposal.

The primary focus of this Guide is new construction, but recommendations may be equally applicable to stores undergoing complete renovation and in part to many other renovation, addition, remodeling, and modernization projects (including changes to one or more systems in existing buildings).

Included in the Guide are recommendations for the design of the following:

- Building opaque envelope and fenestration
- Lighting systems, including electrical interior and exterior lights and daylighting
- Heating, ventilation, and air-conditioning (HVAC) systems
- Building automation and controls
- Outdoor air requirements
- Service water heating
- Plug and process loads
- Commercial kitchens, including cooking appliances and exhaust hoods
- Commercial refrigeration systems, including refrigerated display cases, walk-ins, compressor systems, condensers, heat recovery systems, and related control systems
- Quality assurance, including commissioning and measurement and verification

Additional savings recommendations not necessary for 50% savings are discussed in the "Additional Bonus Savings" section of Chapter 5.

The recommendation tables in Chapter 4 do not include all the components listed in ASHRAE/IESNA Standard 90.1-2004 (ASHRAE 2004). Though this Guide focuses only on the primary energy systems within a building, the underlying energy analysis assumes that all the other components and system comply with the minimum design criteria in ASHRAE/IESNA Standard 90.1 (ASHRAE 2013a) and ASHRAE Standard 62.1, *Ventilation for Acceptable Indoor Air Quality* (ASHRAE 2013b).

In addition, this Guide is not intended to substitute for rating systems or references that address the full range of sustainability issues in food retail design, such as acoustics, productivity, sales rates, indoor air quality, water efficiency, landscaping, and transportation, except as they relate to energy use. Nor is it a comprehensive design text. The Guide assumes good design skills and expertise in grocery store design.

WATER AS A RESOURCE

Water is an important natural resource. Though this Guide deals only with direct building-related energy conservation measures, including service water heating, all water savings result in related energy savings. Water savings from low-flow fixtures and reduced water use from efficient landscaping result in indirect energy savings from pumping and waste disposal. Potable water savings also result in water supply and processing energy savings of 10–25 Btu per gallon of water saved (AwwaRF 2007). Water is also used to produce electricity and to extract and process fossil fuels. Saving energy saves water and saving water saves energy. Both water and energy savings should be integrated into high-performance grocery store design.

REDUCED GREENHOUSE GAS EMISSIONS

According to some estimates, buildings are responsible for nearly 40% of all carbon dioxide (CO_2) emissions annually in the United States (EIA 2011). Carbon dioxide, which is produced when fossil fuel is burned, is the primary contributor to greenhouse gas emissions. Retailers can be a part of the solution when they reduce their consumption of fossil fuels for heating, cooling, and electricity. Grocery retailers, even more so, have a big opportunity to make positive changes due to their opportunity to reduce direct greenhouse gas emissions in the form of refrigerant leak reductions. Historically, grocers have seen annual leak rates averaging 25% of the total charge with standard systems (EPA 2011a). Though designers are using less harmful refrigerants that are less efficient and requiring systems that are less efficient in attempt to reduce this direct impact, these approaches may not make an overall positive impact due to the indirect emissions connected to energy consumption. For this reason, the total equivalent warming impact (TEWI) or life cycle climate performance (LCCP) of a system provides a measurement to ensure that a net positive climate impact is realized. Carbon offsets can also be considered for grocery stores by minimizing the footprints of building and parking areas while maximizing low-maintenance native vegetation. Customers, employees, and communities where stores reside will appreciate this forward-thinking leadership.

ENERGY MODELING ANALYSIS

To provide a baseline and quantify the energy savings for this Guide, a prototypical grocery store was developed and analyzed using hourly building simulations. This building model is a 45,000 ft^2 grocery store carefully assembled to be representative of construction for buildings of that class. Information was drawn from a number of sources, including the Commercial Buildings Energy Consumption Survey (CBECS) (EIA 2003), Dodge Construction Data (Dodge 2009), and various grocery store templates from around the country. Since no refrigeration baseline information was available from ASHRAE/IESNA Standard 90.1-2004, baseline assumptions for refrigeration systems represent an average of nationwide standard practice

Refrigeration Baseline Assumptions

- Air-cooled condensers sized at 10°F temperature difference (TD) for low-temperature systems and 15°F TD for medium-temperature systems.
- Evaporative-cooled condensers sized based on design ambient wet-bulb temperature (wb) as follows: <68°F wb: 25°F TD, 68°F to 75°F wb: 22°F TD, >75°F wb: 18°F TD.
- Condensers controlled with fan cycling for air-cooled condensers and two-speed fan control for evaporative condensers, with fixed head pressure control set at 85°F saturated condensing temperature (SCT).
- Condenser-specific efficiencies of 50 and 140 Btu/h·W for air-cooled and evaporative-cooled condensers, respectively, using a specific efficiency rating point for air-cooled condensers of 10°F TD and for evaporative-cooled condensers of 70°F wb and 100°F SCT.
- Compressor selected with two low-temperature suction groups and two medium-temperature suction groups. Compressors were assumed to be reciprocating semi-hermetic type.
- Mechanical subcooling on low-temperature systems only, to 50°F liquid temperature, using a 20°F suction group to provide subcooling.
- Direct-expansion (DX) cooling design with individual piping circuit runs to compressor racks.
- Display case cooling loads based on 2004 vintage case design, using T8 lights (three rows of lights on open upright cases), shaded pole fan motors, and low-temperature door cases with 215 W/door anti-sweat heaters using modulating heater control based on relative humidity.
- Low-temperature cases assumed to use glass doors.
- Medium-temperature cases assumed to be all open cases with the exception of 15 dairy case or point-of-sale walk-in doors.
- Display case lights operating 24 hours.
- Walk-in evaporator coils equipped with shaded pole motors, except for permanent split capacitor motors on low-profile coils in prep areas.
- Electric defrost on low-temperature cases, meat cases, and walk-ins.
- Walk-ins equipped with strip curtains.
- Temperature control for cases and walk-ins using mechanical evaporator pressure regulators (EPRs) with no floating suction pressure.
- Service water heating heat recovery with average leaving temperature of 100°F.
- No space heat recovery.

design in 2004. See the Refrigeration Baseline Assumptions sidebar for the actual assumptions used for the baseline model. The space types included in the prototype design are as follows:

- Sales areas
- Produce
- Deli
- Bakery
- Entrance/exit vestibule
- Stocking room
- Mechanical room
- Refrigeration rack areas
- Medium- and low-temperature walk-in boxes
- Corridor/transition spaces
- Restrooms
- Enclosed offices
- Break rooms
- Conference/meeting rooms

Two sets of hour-by-hour simulations were run for the prototype. The first set meets the minimum requirements of Standard 90.1-2004, and the second uses the recommendations in this Guide. Each set was simulated in the eight climate zones adopted by the International

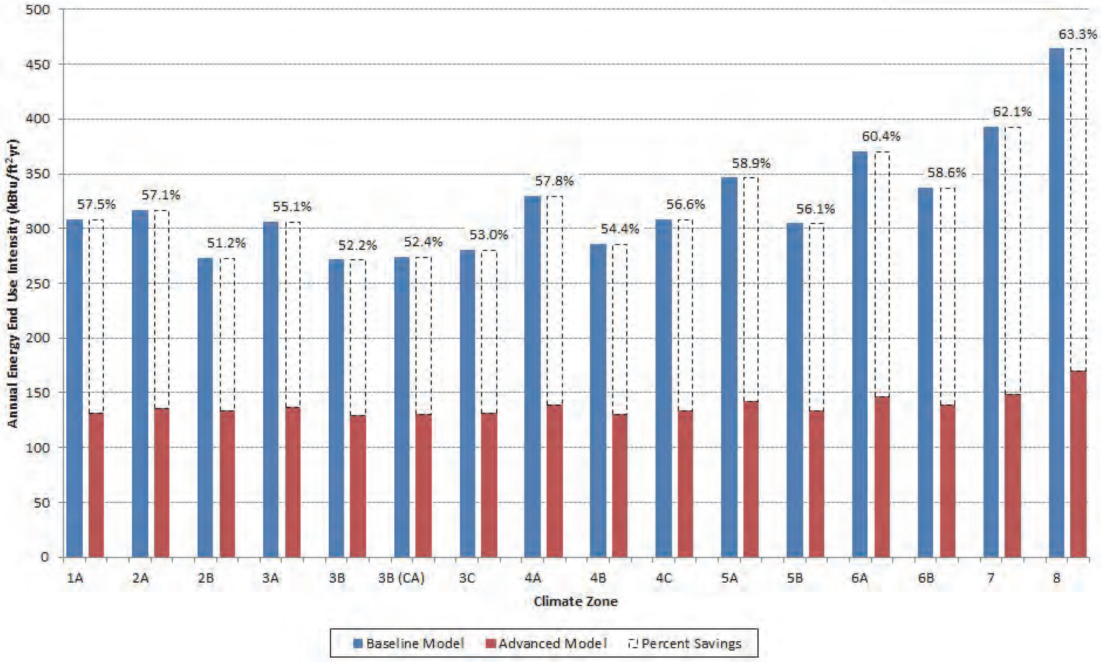

Figure 1-1 Energy Savings by Climate Zone

Energy Code Council (IECC) and ASHRAE in development of the prevailing energy codes and standards. The climate zones were further divided into moist and dry regions, represented by 16 climate locations. All materials and equipment modeled in the simulations are commercially available from two or more manufacturers.

Energy savings for the recommendations vary depending on climate zone, HVAC system type, and store type, but in all cases are *at least* 50% when compared to Standard 90.1-2004. The recommendations are based on the committee's expertise and high-performance best practices. Where appropriate, the recommendations are consistent through all climate zones. As a result, rather than ignore some best practices because they aren't needed to reach 50% savings, some climate zones have a greater than 50% energy savings. This is demonstrated in Figure 1-1, which shows the energy savings for one of the recommended HVAC system types for all 16 climate zones. Additional detail on the energy savings is provided in Chapter 3. Details for recommended system designs are shown in the Chapter 4 recommendation tables and the Chapter 5 how-to tips.

Overall, the savings when compared to Standard 90.1-2004 range from 51% to 63%. Previous guides included comparisons to other editions of Standard 90.1 (e.g., 1999 or 2010). These editions of Standard 90.1 did not contain refrigeration efficiency requirements, which are a significant portion of the grocery load and energy use. Due to this, overall savings comparisons with other editions of Standard 90.1 are not included in this Guide. Preliminary energy savings analysis approaches, methodologies, and complete results of the prototype building simulations are documented in a technical report that was published by the National Renewable Energy Laboratory (Leach et al. 2009).

ACHIEVING 50% ENERGY SAVINGS

Meeting the 50% energy savings goal is challenging, and it requires more than doing business as usual. Here are the essentials:

1. *Obtain building owner buy-in*. There must be strong buy-in from the owner and operator's leadership and staff. The more they know about and participate in the planning and design

process, the better they will be able to help achieve the 50% goal after the store becomes operational. The building owner must decide on the goals and provide the leadership to make the goals reality.

2. *Assemble an experienced, innovative design team*. Interest and experience in designing energy-efficient buildings, innovative thinking, and the ability to work together as a team are all critical to meeting the 50% goal. The team achieves this goal by creating a building that maximizes daylighting; minimizes process, heating, and cooling loads; and has highly efficient lighting and HVAC systems. Energy goals should be communicated in the request for proposal (RFP), and design team selection should be based in part on the team's ability to meet the goals. The design team implements the goals for the owner.
3. *Adopt an integrated design process*. Cost-effective, energy-efficient design requires trade-offs among potential energy-saving features. This requires an integrated approach to grocery store design. A highly efficient lighting system, for instance, may cost more than a conventional one, but because it produces less heat, the building's cooling system can often be downsized. The greater the energy savings, the more complicated the trade-offs become and the more design team members must work together to determine the optimal mix of energy-saving features. Because many options are available, the design team will have wide latitude in making energy-saving trade-offs.
4. *Understand the performance of your current prototype*. For retailers that already use a predetermined prototype, understanding energy performance and actual operational trends in their existing portfolio will provide a valuable starting point for applying the Guide's recommendations. Benchmarking actual energy use by climate, understanding actual occupancy density and outdoor air requirements, and measuring unoccupied load profiles can provide valuable insight into the most effective strategies to meet 50% savings.
5. *Consider energy modeling*. This Guide provides design packages to achieve energy savings of 50% without having to invest in early design energy modeling, but whole-building energy modeling programs can provide additional flexibility and optimization to evaluate the energy-efficient measures on an individual project. These simulation programs have learning curves of varying difficulty, but energy modeling for grocery store design is highly encouraged for achieving energy savings of 50%. See DOE's Building Energy Software Tools Directory at www.eere.energy.gov/buildings/tools_directory for links to information about energy modeling programs (DOE 2015). To properly model a grocery store, it is important to choose a simulation program that has component-level refrigeration system capability and addresses the interaction between refrigerated display cases and sales-area HVAC systems. Part of the key to energy savings is using the simulations to make envelope decisions first and then evaluating heating, cooling, refrigeration, and lighting systems. Developing HVAC load calculations is not energy modeling and is not a substitute for energy modeling.
6. *Use building commissioning*. Studies verify that building systems, no matter how carefully designed, are often improperly installed or set up and do not operate as efficiently as expected. The 50% energy savings goal can best be achieved through building commissioning (Cx), a systematic process of ensuring that all building systems—including envelope, lighting, and HVAC—perform as intended. The Cx process works because it integrates the traditionally separate functions of building design; system selection; equipment start-up; system control calibration; testing, adjusting and balancing; documentation; and staff training. The more comprehensive the Cx process, the greater the likelihood of energy savings. A commissioning authority (CxA) should be appointed at the beginning of the project and work with the design team throughout the project. Solving problems in the design phase is more effective and less expensive than making changes or fixes during construction. See the "Using Integrated Design to Maximize Energy Efficiency" section of Chapter 2 and the "Quality Assurance" sections of Chapters 3 and 5 for more information.
7. *Train building users and operations staff*. Staff training can be part of the building Cx process, but a plan must be in place to train staff for the life of the building to meet energy savings goals. The building's designers and contractors normally are not responsible for

the store after it becomes operational, so the building owner must establish a continuous training program that helps occupants and operation and maintenance (O&M) staff maintain and operate the building for maximum energy efficiency. This training should include information about the impact of plug loads on energy use and the importance of using energy-efficient equipment and appliances.

8. *Monitor the building*. A monitoring plan is necessary to ensure that energy goals are met over the life of the building. Even simple plans, such as recording and plotting monthly utility bills, can help ensure that the energy goals are met. Buildings that do not meet the design goals often have operational issues that should be corrected.

CONDITIONS TO PROMOTE THE GROCERY STORE FUNCTION

A grocery store is a vehicle to present goods, many of which are perishable, to the buying public in an environment that encourages selection and purchase. Inherent to this function is the creation of an appealing ambience for the consumer and provisions to present and preserve the displayed goods. Inadequate protection of perishable goods results in increased overhead costs due to product shrinkage. Throughout a project, the design team should continuously discuss how energy-saving measures will impact the overall function of the facility. The design and construction of a high-performance grocery store requires an integrated approach where these factors remain a priority and are not adversely affected when striving for energy reduction.

For specific guidance regarding the interaction of thermal comfort, indoor air quality, sound and vibration, and other factors, refer to ASHRAE Guideline 10, *Interactions Affecting the Achievement of Acceptable Indoor Environments* (ASHRAE 2011).

SALES ENVIRONMENT

Maintaining space ambient humidity levels is a primary strategy for energy efficiency in grocery stores. Reconciliation of this strategy with maintenance of ideal human comfort conditions can be challenging. Low space dry-bulb temperature is a frequent by-product of aggressive humidity control coupled with open refrigerated cases. The grocery store, however, is not a sedentary environment. Occupants typically are both physically and mentally active, walking through the store and weighing purchasing decisions. As a result, acceptable dry-bulb temperatures in the space may be lower than in office or other commercial building spaces. The guideline for space conditions is that they should not be so extreme as to encourage customers to avoid certain areas in the facility.

INDOOR AIR QUALITY (IAQ)

ASHRAE Standard 62.1, *Ventilation for Acceptable Indoor Air Quality* (ASHRAE 2013b), defines minimum requirements for the design, installation, operation, and maintenance of ventilation systems, but IAQ encompasses more than just ventilation. For more information, refer to *Indoor Air Quality Guide: Best Practices for Design, Construction, and Commissioning* (ASHRAE 2009), which provides specific guidance for achieving the following key objectives:

- Managing the design and construction process to achieve good IAQ
- Controlling moisture in building assemblies
- Limiting entry of outdoor contaminants
- Controlling moisture and contaminants related to mechanical systems
- Limiting contaminants from indoor sources
- Capturing and exhausting contaminants from building equipment and activities
- Reducing contaminant concentrations through ventilation, filtration, and air cleaning
- Applying more advanced ventilation approaches

While meeting these standards for IAQ, however, it should be remembered that product aromas are a frequent component of the grocery store ambience, and their complete elimination may not be desirable.

THERMAL COMFORT

ASHRAE Standard 55, *Thermal Environmental Conditions for Human Occupancy* (ASHRE 2013c), defines the combinations of indoor thermal environmental factors and personal factors that will produce conditions acceptable to a majority of the occupants.

According to ASHRAE Standard 55, six primary factors must be addressed when defining conditions for thermal comfort: metabolic rate, clothing insulation, air temperature, radiant temperature, air speed, and humidity. Appropriate levels of clothing, the cooling effect of air motion, radiant cooling or heating systems, and personal environmental control, for example, can increase occupant comfort efficiently.

In many occupancies, a wide dead band that does not compromise comfort conditions is one means of increasing energy efficiency. For grocery stores, strict maintenance of space dew-point temperature upper limits results in lower energy use because of avoided refrigerated case defrost. Pushing the lower limits of allowable space dry-bulb temperatures may also increase energy efficiency by avoiding reheat necessitated by dehumidification-driven minimum airflow and supply air temperature.

VISUAL COMFORT

The environment of a grocery store, except for back-of-house spaces, is primarily a sales environment rather than a work environment. Conventional visual comfort standards may need to be modified for this environment. Display lighting that highlights specific goods requires contrast ratios that create visual impact. Electric lighting levels and contrast ratios should be designed to meet Illuminating Engineering Society of North America (IES) recommended light levels (IES 2012).

Direct sun penetration should be minimized in work areas because the resulting high contrast ratio will cause discomfort issues. However, direct sun may be used for creating a specific sales ambience.

ACOUSTIC COMFORT

Proper acoustics must be a priority when considering all design decisions and must not be adversely affected when striving for energy reduction. Addressing acoustics during the design phase of a project, rather than attempting to fix problems after construction, will likely minimize the impact on costs.

Recommendations on acoustic comfort can be found in the Sound and Vibration chapter of *ASHRAE Handbook—Fundamentals* (ASHRAE 2013d).

HOW TO USE THIS GUIDE

- Review Chapter 2 to understand how an integrated design process is used to maximize energy efficiency. Review the integrated design strategies for design professionals to develop and evaluate alternatives.
- Review Chapter 3 for information on design concepts and practices and their application. This chapter includes recommendations and methods for choosing between options for refrigeration heat rejection, direct versus indirect systems, and other common site-specific decisions.
- Use Chapter 4 to review climate-specific design strategies and select specific energy-saving measures by climate zone. This chapter provides prescriptive packages that do not require modeling for energy savings. These measures also can be used to earn credits for

Steps for the Building Owner to Follow when Using the Advanced Energy Design Guide

Project Phase	Actions	Outcomes
Project Conception	□ Select the AEDG(s) for your building type from *www.ashrae.org/aedg.* □ Learn about the business case for advanced energy design in the Foreword. □ Review the case studies for similar projects.	➢ Project-appropriate AEDG selection ➢ Project-specific energy performance goals
Team Selection	□ Incorporate AEDG recommendations into RFPs. □ Ask proposers how they used AEDG recommendations and made the business case for energy savings in past projects.	➢ Team with AEDG experience ➢ Team committed to using AEDG
Conceptual Design	□ Require design teams to implement AEDG recommendations. □ Learn about integrated design in Chapter 2. □ Review site-specific costs and benefits of the AEDG recommendations.	➢ Understanding and application of AEDG recommendations ➢ Awareness of cost impacts of AEDG recommendations
Design Development	□ Include AEDG recommendations in the Owner's Project Requirements (OPR). □ Integrate AEDG recommendations into project tracking and status meetings.	➢ Design that incorporates AEDG recommendations
Construction	□ Request regular updates on progress toward AEDG goals. □ Ensure that late project modifications to not compromise AEDG goals.	➢ Verification that AEDG recommendations are installed as designed (through commissioning process)
Operation	□ Verify that AEDG-recommended systems function as intended (through commissioning). □ Leverage the one-year warranty period to address outstanding issues.	➢ High-performance building incorporating AEDG recommendations ➢ Achievement of design energy goals

the Leadership in Energy and Environmental Design® (LEED®) Green Building Rating System (USGBC 2015) and other building rating systems.

- Use Chapter 5 to apply the energy-saving measures in Chapter 4. This chapter has suggestions about best design practices, how to avoid problems, and how to achieve additional savings with energy-efficient appliances, plug-in equipment, and other energy-saving measures.
- Refer to the Appendices for additional information:

  Appendix A—Envelope Thermal Performance Factors

  Appendix B—International Climatic Zone Definitions
- Review case studies and technology example sidebars interspersed throughout the Guide for examples of energy-efficient technologies in grocery stores.
- Note that this Guide is presented in Inch-Pound (I-P) units only; it is up to the individual user to convert values to metric as required.

The recommendations in this Guide are based on typical prototype operational schedules and industry best practices as well as typical costs and utility rates. The operational schedule, actual costs, and utility rates of any one project may vary, and it is encouraged that each specific project perform life-cycle cost analysis (LCCA) for key design considerations to properly capture the unique project costs and operational considerations**.**

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Whole Foods Market—A Case Study

The Whole Foods Market store in Raleigh, North Carolina (Climate Zone 4A), opened in March 2011. The 40,000 ft^2 single-story, stand-alone store includes fresh and packaged food items, general merchandise, an in-house bakery, a deli, and food preparation areas.

The store was the first constructed under Whole Foods Market's goal of reducing annual energy consumption by 50% as part of DOE's Commercial Buildings Partnership program (EERE 2015a). The criteria used to evaluate energy-efficient measures for the project include payback period (≤ 5 years), tax incentives, and utility rebates, as well as costs for capital, installation, O&M, and energy.

Interaction between the building's systems was a key design consideration. The design team estimated energy savings for combined packages of energy efficiency measures rather than for individual measures. This allowed for the consideration of individual energy efficiency measures that may have been deemed too expensive had they been evaluated individually as well as for the determination of savings for the entire system and not just for components.

Refrigeration

Refrigeration is a major portion of the energy use in a supermarket. Strategies employed to reduce energy usage of the refrigeration systems include the following:

- Case doors are included on medium-temperature dairy, deli, and packaged produce cases to reduce load.
- Night curtains are used on open medium-temperature cases to reduce load during unoccupied hours.
- Anti-sweat door heaters were installed to minimize condensation. A lower sales floor dew point is maintained in conjunction with the anti-sweat control strategies for the refrigerated case doors.
- LED lights are used instead of T8 fluorescents in all low-temperature and medium-temperature refrigerated cases and walk-in freezers to reduce both lighting power usage and additional load from lamp and ballast heat.
- Electronically commutated (high-efficiency) evaporator fan motors were installed in refrigerated cases.

Refrigerated Cases with Doors

LED Case Fixtures

Photographs reproduced with permission of Whole Foods Market and courtesy of NREL, credit Jennifer Scheib

- Variable-speed fans are used to cool the low-temperature condensers.
- Electronic expansion valves (EEVs) maintain temperature control while the store uses a lowered minimum SCT (55°F versus 75°F).
- Waste heat is captured for both air and service water heating.

HVAC

The design of the HVAC systems needs to be integrated with the refrigeration and kitchen systems in the store. Energy-efficient measures include the following:

- Sales floor humidity is controlled via air-handling units equipped with a solid desiccant wheel. Instead of using natural gas, the wheel is reactivated with waste heat from the condenser coil.
- Total airflow rate in the store was decreased from 1.0 to 0.6 cfm/ft^2 to reduce fan power consumption.

Kitchen

There is significant interaction between the kitchen and store areas. Measures employed to maintain IAQ and contain odors include the following:

- Side panels were installed on all the exhaust hoods, which allows for a lower exhaust flow rate and improved capture of the exhaust fumes.
- Demand-controlled ventilation sensors and controls are included on the exhaust hoods to reduce the exhaust flow and required makeup air when there is no cooking. The sensors allow the hood to operate in response to hot temperatures or smoke from the cooking surface.

Lighting

A number of considerations must be balanced when designing a lighting system for a grocery store, including efficacy, controllability, color rendering, and cost. The measures selected for this project include the following:

- The total installed lighting was reduced to 1 W/ft^2. Ambient lighting is provided with a combination of linear fluorescent, metal halide, and LED fixtures.
- Lighting is controlled in the dry goods section with a bi-level daylighting strategy that allows for OFF, 50% ON, and 100% ON lighting.
- Skylights were installed and their distribution optimized to improve control in response to daylight.

Solid Desiccant Wheel

Photograph reproduced with permission of Whole Foods Market and courtesy of NREL, credit Mike Farish

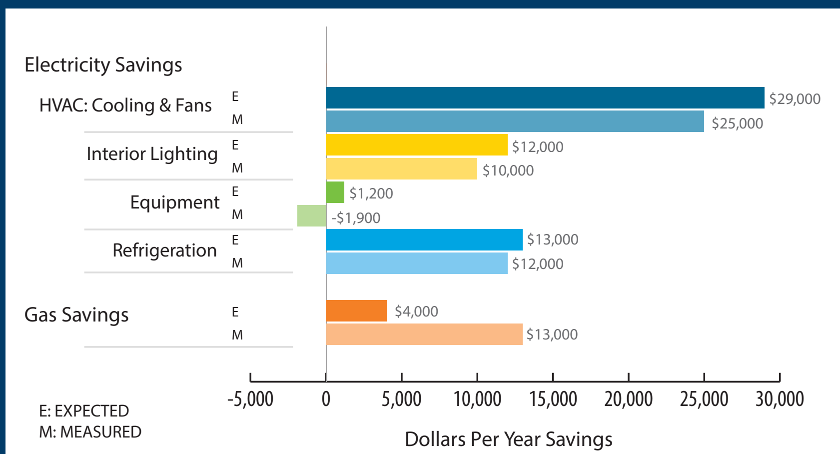
Energy Savings

Data from June 2012 to July 2013 were used to evaluate the store's performance and showed a total annual energy use intensity (EUI) of roughly 290 kBtu/ft^2, which represents considerable energy cost reductions, as illustrated in the graph below.

Additional information on the Whole Foods Market Raleigh store project can be found in the Buildings Database on the EERE website: https://buildingdata.energy.gov/project/whole-foods-market-commercial-building-partnerships-new-construction (EERE 2015b).

Deli and Food Preparation Areas

Photograph reproduced with permission of Whole Foods Market and courtesy of NREL, credit Ian Doebber (left) and Jennifer Scheib (right)



Expected and Measured Energy Cost Reductions

Source: EERE (2013)

Integrated Design Process

2

This chapter discusses the principles of integrated design and demonstrates how they can be used to implement energy efficiency strategies. For the most part, these strategies apply to initial prototype development and one-off stores. The final section in this chapter discusses how the integrated design process can be modified for the future refinement of existing prototype designs.

Some grocery stores are one-offs (unique buildings), while others are built from prototype designs. A one-off grocery store design effort results in detailed design for a specific site and climate, for a specific program, so that the integrated design process can be carried to final systems and equipment selections and final details. For the prototype grocery store, the integrated design process results in the identification of specific selections for some elements of the design that will not change with different sites and a set of alternatives for elements that are specific to the site and the exact program for the project. When the prototype design is adapted to a site with a final program, energy analysis and life-cycle costing may be used to select from the identified alternatives using applicable utility rates. Both traditional and prototypical integrated design processes are further described in this chapter.

Understanding the building owner's needs is a critical part of integrated design. The Owner's Project Requirements (OPR) should be part of early design discussions to allow for optimal system design that meets the current and future needs of the building owner without unnecessary assumptions leading to system and equipment oversizing that can reduce operational efficiency. Applying the integrated design process to grocery stores includes incorporating these programmatic requirements into the design, evaluating design alternatives for life-cycle costing, identifying opportunities for integration between HVAC and refrigeration systems, and verifying energy savings.

PRINCIPLES OF INTEGRATED DESIGN

Integrated design is a method for design and construction that considers whole-building system interactions and uses an interactive team approach for all phases of a project. Integrated design is necessary to achieve 50% or better energy savings. In this process, all parties work together through the phases of a project to maximize the efficiency. This results in a coordinated, constructible, and cost-effective design. Integrated design increases the productivity of the project process, provides higher-performing buildings and, because of better communication and coordination between disciplines, protects the construction budget by reducing construction change orders.

Integrated design fosters unique opportunities to build connections for the grocer's corporate sustainability mission. Saving energy not only results in bottom-line profit but also can become a visible symbol of a grocer's commitment to sustainability and community well-being.

First and foremost, a grocery store creates a customer experience as a means to encourage purchasing. Some elements of the experience, such as the display or preparation of products, may have significant implications for the energy consumption of the facility. Energy conservation strategies must be consistent with the creation and maintenance of that experience.

One of the most important concepts of integrated design is that some of the simplest, most up-front architectural decisions about a building's form, orientation, window placement, and even internal layout can have significant impacts on the efficacy of refrigeration, HVAC, or lighting conservation strategies. Integrated design encourages the team to take these issues into account from their earliest design decisions through active project team discussion and collaborative dialog between all parties on a project team. The efficiency and quality of the project's design and construction are obtained through the following team interaction and process recommendations:

- Issue a request for proposal (RFP) or request for qualifications (RFQ) to define energy design and performance goals and expectations and identify the project team and stakeholders. Procurement documents for an integrated design process should identify design elements critical to brand identity and customer experience while specifying goals, and possibly incentives, for meeting and exceeding performance metrics for the store.
- Establish early involvement of all design and construction team members.
- Establish initially-agreed-upon and documented common goals, including operational baseline performance benchmarks.
- Consider new and different methodologies.
- Establish open communication with early input from all parties on project strategies.
- Provide life-cycle costing and factor in value-added benefits to determine the feasibility of project systems.

A key difference between integrated design and conventional design is the consideration of life-cycle costs in making project decisions. This requires a holistic approach at the beginning of the project to input relevant programming, design, construction, and operations information to the greatest extent possible. This means that corporate representatives, architects, engineers, contractors, commissioning authorities, building operators, and other integrated design team members must work together from the outset of a project to accumulate the necessary information to make data-driven decisions. For grocery stores, it is important that technical expertise on the required refrigeration systems and, if applicable, kitchen equipment be present from the beginning of the design process so that the implications of these systems can be integrated with the developing HVAC and lighting designs. In a design-bid-build project, contractor input is needed to ensure that needed construction information is available in a design-build or construction management delivery.

Consider a design team that includes a strong corporate or owner's advocate who understands the risks and rewards of design decisions when building energy-efficient grocery stores. This could be an internal person with design expertise or a consultant with long-term, detailed knowledge of the building owner's needs. In any case, the design team must establish a partnership to evaluate both the risks and operational implications of design alternatives.

It follows that the integrated design process requires the formation of the project team early in the process. Early collaborative goals and metrics of performance and open, inclusive participation contribute to trust among team members and to the overall success of the project. The inclusion of all the project team members benefits the project by allowing all the participants to provide their expertise throughout the process. Successful implementation of this process should provide the owner with a built project that achieves the desired level of energy conservation with the desired level of programmatic flexibility and known requirements for operation and maintenance.

USING INTEGRATED DESIGN TO MAXIMIZE ENERGY EFFICIENCY

Integrated design establishes key collaboration agreements to remove barriers between parties and to encourage early contributions of wisdom and experience. This section provides best practice guidance to achieve 50% or better energy efficiency than ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings* (ASHRAE 2004), in the building design.

PROJECT KICK-OFF

The project kick-off meeting is the most important meeting of the entire project with regards to establishing the OPR. This exercise may be led by the commissioning authority (CxA) or the owner's representative, and it allows the corporate personnel and stakeholders to define what a successful project means for them. A well-defined OPR at the beginning of the project will help ensure that the energy goals are integrated into the design and considered throughout the project. Inclusion of the stakeholders (corporate representatives, store managers, maintenance staff, store employees, customers, etc.) will produce more creative and integrated solutions, which are key to the project's success. The requirements can cover construction costs, serviceability, operating costs, required spaces and adjacencies, functional aspects, specific maintenance or system preferences, frequency of use, and other owner priorities. It is strongly recommended that the traditional OPR from the commissioning (Cx) process be augmented to include the following information:

- Corporate brand requirements that might restrict some design measures
- The owner's goals for creating a customer experience
- Targeted building energy and sustainability ratings, such as those offered by the Leadership in Energy and Environmental Design® (LEED®) Green Building Rating System (USGBC 2015), Green Globes (GBI 2015), Building Energy Quotient (bEQ) (ASHRAE 2015), ENERGY STAR® (EPA 2015a), etc.
- Life-cycle costs of systems and cost transfer
- Quantification of value-added benefits of efficiency measures
- Ownership/leasing arrangements, including renewable energy credits and utility back-charging or metering
- Prioritization of requirements
- A clear delineation of the owner's chain of command and communication protocol for decision making, including that for change requests and expenditure approvals
- Constraints imposed by the site, code, or planning agreements with the city, preexisting standards (if any), corporate sustainability policy statements, etc.
- Site-based measurements of existing equipment or similar equipment at another store to determine actual plug-load usage

The OPR is necessary to ensure that all parties on the design and construction team are equally aware of the owner's priorities. While there may be multiple systems that can meet the criteria, the specific systems to be considered can be narrowed or identified early in the project. This lowers risk for all parties and provides a reference document to guide future decisions when budget pressures challenge system selections. It is acknowledged that the OPR may grow (or change) during the course of the project to accommodate corporate preferences and take advantage of new technologies or opportunities. Nevertheless, it is good practice to keep a comprehensive list of all OPR changes.

PROGRAMMING AND CONCEPT DESIGN

During the programming and concept design phase, the specific parameters for the project are compiled and organized. This phase also incorporates a series of brainstorming sessions that assist the integrated design team in evaluating the OPR for opportunities and risks in the context of the site. A key conceptual exercise usually covers a series of holistic site investigation and building massing studies to see which strategy best addresses the following issues:

- Site conditions (e.g., existing shading from adjacent buildings or landscaping, outdoor air quality, outdoor ambient noise environment, outdoor ambient temperature and humidity profiles, site surface material)
- Orientation and availability of natural resources (e.g., sun, wind, geothermal, climate, bodies of water)
- Prevailing climatic conditions that will impact the dehumidification requirements for the refrigerated cases and the required conditioning for the kitchen exhaust makeup air
- Local material availability or reuse
- Site hydrology, including storm-water management and regulatory implications of any designated wetlands
- The utilities available on the site, which will dictate choice of energy source, potable water source, and disposition of wastewater
- Status of surrounding buildings and review of code/planning regulations that may create obstructions to natural resources in the future or otherwise limit the design
- Hardscaping or landscaping potential to reduce a heat island effect or provide natural shading
- Security concerns
- Accessibility to transportation
- Sustainability opportunities
- Environmental risks or challenges

The goal of the programming and concept phase is to review a number of schemes and identify appropriate strategies and resources. It may be difficult to correctly estimate final costs for each model or to determine the final systems, but past experience should provide the integrated design team the ability to rank schemes qualitatively against the OPR. The exercise should result in the integrated design team coming to consensus on the major site parameters and identifying the major design strategies to meet the OPR.

For grocery stores, configuration of refrigeration, kitchen, and exhaust systems is much more important to the design of HVAC systems and resulting energy efficiency than is the form and envelope of the building. During the programming process, the extent of refrigerated storage; the approach to refrigerated display of goods; and the extent, type, and location of in-house cooking—along with the desired flexibility for future reconfiguration of these facilities—should be established.

Space adjacency inside the building is critical for grocers, so early discussion in the programming phase of the impact of space location relative to energy efficiency is important. Space programming inside the building might change the building shell or orientation requirements. Adjacency can reduce the cost of implementing energy efficiency measures and improve their effectiveness. For example, the location of open refrigerated cases with respect to entrances or kitchen exhausts can affect the optimal design of HVAC systems for the space.

At the end of programming and concept design, a consensus should be reached among the integrated design team regarding program design concepts or parameters including (but not limited to) the OPR, site, building envelope, orientation, possible systems, and so on. However, the owner must confirm the basic strategies for a positive and productive workplace environment, including the extent of visual connection with the outdoors. A grocery store may have limited options for site orientation and placement due to standardized design or coordination with surrounding buildings, but considerations should be discussed when possible.

SCHEMATIC DESIGN

The schematic design phase is for developing programming ideas and concepts into diagrammatic plans. At this phase, the architectural team members identify where the various program occupancies are projected to be in the building as a test exercise. The schematic design plan and proposed building massing provide opportunities to study and identify envelope strategies for fenestration and opaque surfaces to carry out program parameters and enhance the indoor environment for the occupants. This building massing and programmatic building layout becomes the basis for the initial energy model on which schematic energy conservation measures (ECMs) will be tested.

Important characteristics of grocery stores are the thermal balance between internal heat gains such as lighting and equipment and uncontrolled internal cooling from refrigerated cases and the airflow balance between required occupant ventilation and required kitchen exhaust. During the schematic design phase, the range of these balances and how they are influenced by both operational and weather variations should be explored. This investigation will provide important information for conditioning requirements to maintain both human comfort and the required ambient conditions for refrigerated display cases. These requirements are the primary determinants for HVAC system type and components and are critical to achieving both energy efficiency and the desired customer experience.

As a start, requirements and locations should be generated for alternative mechanical and electrical systems strategies as input for construction and life-cycle cost comparisons in subsequent phases.

Once a baseline building is created, its costs are estimated and compared to the OPR to ensure that even the most "standard" of the available designs meets the first-cost and program requirements. It is often useful to perform a preliminary energy analysis by zone to determine approximate annual operating costs for different design alternatives. The results of this analysis should be compared with energy bills for similar stores to establish that the model is a reasonable representation of real-world performance. During schematic design, this level of calculation is usually adequate to confirm trends in energy savings associated with design decisions.

The last necessary work for the schematic design phase is to identify ECMs that might be applied to the baseline case or alternate schemes that improve energy efficiency while maintaining operational and customer experience performance of the baseline case. This is the point at which a thorough discussion of trade-offs and cost transfers should be documented. Typical exploratory interdisciplinary discussions during this phase include the following:

- Identification of structural and cladding systems and implications for envelope thermal performance
- Identification of internal finish strategy (exposed structure, suspended ceiling, etc.) and implications for lighting and air distribution strategies
- Selection of façade type and orientation and each face's relative proportion and performance (properties of glazing and opaque wall insulation)
- Configuration of roofing shape/slope/direction and applicability of cool-roofing materials, clerestories, and skylights and/or installation of photovoltaic (PV) panels
- Commitment to ENERGY STAR equipment for reduction in energy consumption for internal appliances
- Review of kitchen functional requirements, proposed kitchen equipment, and resulting exhaust/makeup air systems
- Review of alternative refrigeration system designs (such as distributed or secondary systems)
- Review of alternative HVAC systems

As a conclusion to the discussions, the design team usually identifies a certain number of optimal ECMs that they wish to append to the baseline. Through a life-cycle cost analysis (LCCA), a more complete energy model is created to evaluate ECM payback. A matrix of options is developed to assess each ECM against a common set of criteria, including but not limited to the following:

- Additional first-cost investment
- Anticipated annual energy cost savings
- Anticipated annual maintenance costs savings or additions
- Return on investment
- Energy reduction
- Carbon emissions savings
- Potential additional LEED points for Energy and Atmosphere Credit 1 (USGBC 2015)
- Thermal comfort maintenance for customers
- Flexibility to accommodate future changes to program or to customer experience strategy
- Value-added benefits not related to energy use

There may be other project-specific owner requirements that should be incorporated into the matrix. The key point is that it is important for all parties to understand the whole view of any ECM application so that a balanced decision can be made inclusive of all impacts on the desired goals of the project, including operations. The goal is to pick a selection of ECMs to pursue during the design development phase. These decisions are quite crucial before design work and calculations are begun in design development. In many cases, while energy savings may not be enough to justify the cost of a more efficient strategy, that strategy may still be desirable based on other value-added benefits. For example, a measure that reduces energy use may also improve occupant comfort, resulting in reduced operational costs, an improved customer experience, and increased sales. Reconsideration of selected ECMs during value engineering in subsequent design phases should recognize the synergistic behavior of some ECMs rather than addressing their "economic performance" in isolation.

DESIGN DEVELOPMENT

The design development phase establishes the final scope of ECMs into the architectural scheme for the project. The final energy models are usually used for submission to code authorities to show compliance with ASHRAE/IES Standard 90.1 (ASHRAE 2013a) and may be used for submissions for LEED (USGBC 2015). The development of the scheme includes further design, calculation, and documentation of the building envelope, lighting, and mechanical/plumbing systems that are regulated by code as well as corporate-set limits on plug load densities. A whole-building energy model is the best way to address the estimate of savings by various ECMs because it captures the synergistic behavior among certain ECMs. Additionally, there is often a financial investment/LCCA of the ECM components. These cost estimates are more detailed at this phase. The cost analysis justifies the ECMs to be maintained as goals in the next phase of design. During design development, the focus is on documentation of the design intent, and most energy-related disciplines will write a Basis of Design (BoD) report explaining the design intent. This BoD is then compared to the OPR by the CxA during a peer review process to ensure that the owner's goals are met by the design development model.

Design development is the phase of the project where the original OPR is confirmed through the project scope. Safety factors, diversity assumptions, and redundancy and critical service requirements are identified as a team exercise during design development, and these decisions should be reflected in the system design and component sizing. These decisions may have significant impacts on the estimated cost for the design. The cost analysis provides the basis for value engineering and decisions regarding first costs and operations. If a prototypical design is used by the grocer, the prototype documents may be incorporated into design development with many of the typical design decisions addressed.

During design development, strategies for cooking and refrigerated display will be finalized so that the range of conditioning requirements to meet human comfort criteria can be defined. Participation by the providers of these systems is important at this point so that not only their requirements for design conditions but also their operational characteristics in off-peak periods can be determined.

CONSTRUCTION DOCUMENTS

The construction documents (CDs) phase is the final detailing of all systems, inclusive of sustainability features and ECMs. The mechanical, electrical, and plumbing systems incorporate system drawings, specifications, BoD reports, controls drawings, controls points lists, and sequences of operation. The CxA reviews all of the documents and the updated BoD for compliance with the OPR.

At this point, it is important for the team to review and confirm project constructability, cost-optimization and waste-reduction techniques, necessary documentation, and acquisition of materials to meet the performance requirements for each of the ECMs, the control strategies, and the design intent of the specifications. Since many grocers purchase components of their buildings directly, any modifications to typical materials should be coordinated with the grocer's procurement group to ensure appropriate materials will be procured.

Some of the systems for the grocery store may be procured through a separate design-build procurement process, no matter the process by which the construction of the balance of the project is procured. In that circumstance, the design-build procurement documents must be configured with sufficient performance requirements and coordination language to ensure that the system thus procured will meet the energy-efficiency expectations for the project.

BID PHASE

Most grocery store projects will use a design-bid-build delivery. Projects using other delivery methods may bypass the bid phase. For projects that include a bid phase, the following measures by the design team are essential to achieve 50% savings or better:

- Acquire timely and appropriate construction information and expertise in the early phases of the project to ensure constructability; availability of equipment; materials; necessary skilled labor; and quality, accurate estimating and cost control.
- Provide a thorough and interactive prebid conference to discuss the desired corporate goals, including ECMs, the Cx process, and contract documents.
- To ensure that corporate goals are achieved, consider adding general performance specification requirements, including contractor experience on high-performance buildings.

CONSTRUCTION ADMINISTRATION

During the construction administration phase, the CxA and the design professionals on the integrated design team will review submittals and construction performance to ensure compliance with the contract (construction) documents. In accordance with the contract documents, the integrated design team will ensure that construction meets all regulatory requirements and is in compliance with manufacturer performance and warranty standards. Design professionals on the integrated design team are responsible for reviewing the construction, reporting any deficiencies of installed work, and requiring remedial efforts to correct the work if necessary. Any deviation from the CDs must be approved by the integrated design team, and the documentation must prove that the substitution will not adversely affect energy efficiency (among other things).

After all the equipment is installed and the building is enclosed, equipment manufacturers perform testing procedures during start-up and confirm that the equipment is operating correctly. A testing, adjusting, and balancing (TAB) contractor adjusts the settings on the equipment to achieve the water flow and airflow as required in the CDs.

The contracting team and manufacturer representatives are responsible for producing a set of operation and maintenance (O&M) manuals and performing the specified hours of training for the building's personnel. It is recommended that the training sessions be recorded and kept on file for future operators. It is further recommended that key technical facility operators be on site during the final month of contractor operations prior to final inspections and that checklists are started to familiarize these operators with the equipment and design intent. When final inspection begins, building operators should accompany the contractor and design team during testing of the equipment.

Start-up and operating performance tests for refrigeration systems may be on a different timeline and performed by different personnel than those for building HVAC systems, especially if they are provided outside the contract for the general construction of the building. The owner likely will have refrigeration specialists on staff who should attend these activities. They should also attend training sessions offered by the equipment vendor to learn special characteristics and operational requirements for the specific equipment installed in the store.

COMMISSIONING

Commissioning activities should be performed prior to the store opening to ensure that all building systems are functioning properly in all modes of operation. Commissioning in this stage is essential to the overall integrated design process, as it confirms that the performance of all building systems, considered together, is consistent with the design strategy for 50% energy savings and meets all the requirements within the OPR.

Commissioning activities in this phase, such as performance testing or prefunctional testing, bridge the gap between construction and start-up and involve different building teams and contractors that are responsible for correct installations. This process will typically be managed by the CxA. For example, the CxA ensures that the construction team completes any checklists developed in earlier phases and may confirm the findings through random sampling. The CxA should also supervise the contractors as control system performance tests are carried out, verifying that the system adequately reacts to artificially applied inputs. If the team has executed the Cx plan and is aligned with the quality assurance (QA) goals, the performance testing will occur quickly and only minor issues will need to be resolved. Likewise, the CxA assists with the supervision of the formal training of the owner's operations personnel. The training ensures that service technicians and store personnel can operate and maintain the systems properly and can make corrective actions should the system performance degrade. Performance testing and training can be conducted jointly: the functional testing process can double as a training tool to educate O&M staff on how the systems operate as well as on system orientation prior to training.

When all performance testing and other Cx activities are complete, the CxA issues a report to the owner confirming that the performance goals of the project have been met. At that time, the owner can take over the building operations from the contractor with confidence that the design and construction are operating as intended, meeting the 50% energy savings goal.

START-UP AND OPERATIONS

Operation and maintenance (O&M) of the equipment after the contractor's transfer of the building to the owner are essential to achieving and maintaining 50% energy savings. It is recommended that personnel responsible for building systems operations secure certification as certified energy managers. It is often the case that the first year of occupancy will reveal a truer nature of how the building will actually perform. The actual occupancy patterns may be differ-

Refrigeration Commissioning Guide

Commissioning of refrigeration equipment is a very specialized process that is thoroughly described in *Refrigeration Commissioning Guide for Commercial and Industrial Systems* (ASHRAE 2013b). Procedures described in this publication may not be familiar to building Cx specialists, so the owner may wish to engage a separate party familiar with Cx of commercial refrigeration systems.

ent from the original design assumptions in the energy model. Often, the CxA will have a service extension for 11 to 18 months after the building has been transferred to review the status of operations. The CxA may recommend adjustments to system setpoints to optimize operations. Occasionally, a second measurement and verification (M&V) exercise is performed at this time to benchmark the energy use of each piece of equipment. If there are extreme seasonal differences in the particular climate, a second set of benchmarks may be established in an alternate season.

It can be beneficial for the design and construction team to host a seminar for the initial occupants (taped, if possible, for future occupants) to describe the building's design intent and sustainability features. This is an ideal way to introduce the occupants to the building and to allow the store leadership to state their support for the energy efficiency initiatives in the building. It is important to caution occupants that a low-energy building sometimes takes a period of operation to reach its optimized control strategy. All occupants are invited to submit and share energy performance comments as additional inputs for improving the operations. This type of personal engagement will encourage positive-impact behaviors with the occupants. These first adopters can act as efficiency coaches for future occupants.

Many multisite owners provide remote monitoring and oversight of building operations. In this case, it is important that the remote operators of building systems are aware of any unique aspects of the energy efficiency measures of a specific building so they can help ensure appropriate operations. This remote monitoring also provides a means of ongoing verification of operations and can alert staff to faults that may impact performance.

For grocers using a prototypical design for building construction, a similar procedure can be used to provide a consistent approach to operating the building. Standardized training manuals, videos, and classes can be provided for on-site building operators and outside service vendors that work on building systems. Very specific operational materials focused on the systems found in the store without extraneous information will help maintain high-performance operations.

PROTOTYPE DEVELOPMENT—CONTINUOUS IMPROVEMENT

Grocery stores differ from many other building types in that they often have a prototype design that is constructed at many different locations across the country. The integrated design process discussed in this chapter should be followed for initial prototype development. If a prototype design already exists, owners are encouraged to reevaluate the design using the full integrated design process. However, this exercise may not be feasible considering how much time, effort, and money has already been invested in the design of the existing prototype. In cases where an existing prototype design is used, it is possible to incorporate energy efficiency into a portfolio of buildings through the following:

- Take advantage of the portfolio of buildings to continually optimize design based on an understanding of how existing similar buildings operate.
- Be prepared to vary the prototype by climate to achieve 50% energy savings.
- Design in possible future retrofit opportunities by allowing for future flexibility and upgrade potential, such as increased structural capacity for heavier HVAC equipment.
- Review site-specific opportunities for energy improvement that may not be part of the prototype.

Even though complete overhaul of the prototype design may not be feasible for marketing or cost reasons, there are some advantages that a prototype design has over a typical one-off grocery store design. The prototype often has been constructed in many different locations and has been in operation for many years, which means that there are hard data (utility bills) that can be used to benchmark its energy performance. Many portfolios also have detailed submetered data that provide even more information for benchmarking. Using these data can help immensely in evaluating current prototypes and existing stock for energy improvement opportunities, both as retrofits to existing stores and improvements to the existing prototype design.

INTEGRATED DESIGN PROCESS STRATEGIES

The most important characteristics of the integrated design process are sharing of information and quantitative evaluation of design alternatives. The following sections provide descriptions of strategies by which these goals can be achieved.

CHARETTES AND DESIGN REVIEWS

It is very desirable that design teams pursuing significant energy savings engage in the habit of early-phase design charettes involving all team members, followed by periodic design reviews. This process of holding one another accountable throughout the design process helps ensure that unintentionally myopic thinking on the part of any one team member doesn't accidentally propagate into a vulnerability not identified until commissioning. The entire team must understand the multidisciplinary multilateral agreements as noted herein, must acknowledge and support the achievement of stated energy-use goals, and must review the documents as they grow to ensure that the holistic system survives through detailing and value engineering processes.

Typically, kick-off charettes are convened by a named facilitator who sets the ground rules of the brainstorming session to encourage people to contribute and, most importantly, to listen. There should be agreed-upon time limits to each person's speaking length during the brainstorming time and time limits on the brainstorming period. All ideas are welcome and can be raised without judgment during the brainstorming period. It is often useful to start with a brainstorming period related to project and team goals, followed by a discernment session that allows the "brain-dump" list to be ordered with prioritization for time and cost investment. This can then be followed by a "blue-sky" type of brainstorming related to energy efficiency measures. During this brainstorming session, it is necessary to refrain from actually starting to design or the value of the limited-time creative output from all team members may be diminished. There will be months to design using the great ideas thereafter.

Design reviews can benefit from reviewers that are both internal and external to the team. Internal reviewers are intimately aware of all of the step-by-step decisions that led to the current state. External reviewers provide a level of objectivity and can offer advice from past experience on similar challenges. The CxA's job is to review the content for commissionability and impact on energy use. Again, a facilitator may be necessary to ensure that all reviewers have time to speak without their suggestions being immediately contested by those parties with an investment in the status quo of the current design or who are biased for other reasons. It is often beneficial to capture in writing all of the comments in an objective manner so that they can be respectfully addressed in sequence and a resolution on change of or continuance of design direction can be achieved and shared by the entire team.

USE OF ENERGY MODELING AS DESIGN GUIDANCE

Energy modeling is a powerful design tool for reviewing the relative energy savings of various ECMs. It can be further exploited when coupled with thorough financial analysis to ensure that the investment in initial cost will pay for itself in annual energy savings. The whole design team should understand that it is difficult for the current state of energy modeling software to predict actual energy use of a building; it is best suited to compare options to each other. As noted previously, whole-building energy models can capture the synergistic effects of multiple ECMs and thus are a good tool for evaluating the impact of selective removal of ECMs in any cost-cutting or value-engineering exercise. There is no federal standard verifying the absolute accuracy of an energy modeling engine as compared to real life in uncontrolled circumstances. As a rule of thumb, results of at least a 5% relative energy savings arising from comparative energy models with ECMs applied to the same source file are probably a true indicator of measurable savings in real life. Anything less should be reviewed by the design team in a careful risk management process.

For a grocery store, the selected energy modeling platform should have the ability to assess the impact of open refrigerated cases on the HVAC system in the building. It also should be able to assess the impact of the most common ECMs for commercial refrigeration systems. The interplay between HVAC and refrigeration components in the space can have a significant impact on building energy use, and the selected energy analysis technique should be sensitive to these issues.

The energy modeling process involves a very large amount of data input, and in some software programs it is extremely difficult to change geometry after the fact without rebuilding the entire model. As such, the design team should understand how it will choose to spend its limited energy modeling fee from the start and ensure that each model run is absolutely necessary to confirm beneficial direction. There are many aspects of this Guide that have proven energy reduction benefits and are now considered best practice that do not need to be analyzed individually for cost-effectiveness. For instance, any nontechnical person acknowledges that reductions in plug and lighting loads reduce ultimate energy use given equivalent schedules of usage. To achieve 50% reduction in building energy use as compared to ASHRAE/IESNA Standard 90.1-2004 (ASHRAE 2004), energy modeling should be confirming the relative size of an already known benefit, not proving a bad position to resistant team members.

MULTIDISCIPLINARY COORDINATION FOR ENERGY EFFICIENCY

Integrated design strategies require significant multilateral agreement on design intent from a variety of stakeholders. The following tips are provided to identify a series of items for which a direction and agreement must be achieved. Truly holistic low-energy design solutions are not achieved solely through the optimization of each component but rather by exploiting the mutually beneficial synergies between design strategies.

Define Business as Usual and Baseline Buildings

One of the very first things that the design team must define is what the business-as-usual (BAU) design solution would be. The BAU case is typically defined as a simple, box-shaped building that fills the site and is minimally compliant with ASHRAE/IESNA Standard 90.1 (ASHRAE 2004). For a grocery store, identifying the extent of refrigerated cases and food preparation is a prerequisite for defining the BAU case. The energy use of this building typically defines the upper bound for allowable energy use and sets the comparative standard against which absolute savings are measured on the road toward net zero energy use. As each ECM is applied, the design team should track incremental victories. Because comparison to the BAU case truly reflects all design decisions, including those related to building form and orientation, this comparison is the real measure of project success; this is especially true with respect to cost justification.

The second key item that the design team must define is what the baseline design solution would be once the preferred building configuration's design is completed. The baseline, which defines the energy use goal that 50% savings represents, is very different from the BAU case. Standard 90.1 requires that all proposed and baseline energy models have identical shapes, footprints, and occupancies. Thus, the baseline does not reward fundamental building configuration decisions for their positive effect on energy use. Again, while such decisions do not contribute to energy savings as defined with respect to a Standard 90.1 baseline, it is important to remember that they do contribute to overall project success and can be quantified using the BAU case.

It is important for the design team to agree to move away from both the BAU case and the baseline in making proactive design decisions. It is also important that there be no shifting benchmark of success.

Benchmarking

While the BAU case represents the highest allowable energy use intensity (EUI) on site by calculation methods, there are other energy-use benchmarks that can be used, including the following:

- U.S. Environmental Protection Agency's (EPA) ENERGY STAR Portfolio Manager (EPA 2015b)
- The grocer's existing portfolio
- U.S. Energy Information Administration's Commercial Buildings Energy Consumption Survey (CBECS) (EIA 2003)
- California Energy Commission's California Commercial End-Use Survey (CEUS) (CEC 2006)

It is possible to benchmark the proposed design against the BAU case and against its preexisting peers to demonstrate that substantial steps have been taken toward energy use reduction. Designers often successfully compare their designs to the "typical" equivalent building in the preexisting stock or to the number of houses that could be powered on the energy savings to make it easier for laypeople to understand the magnitude of energy savings.

Historic data, however, are not the inspiration for good design in the future. This is where more aspirational benchmarking can benefit the team. The most frequently used benchmarks are as follows:

- Energy savings as designated by percentage annual cost savings as compared to ASHRAE/IES Standard 90.1, Appendix G (ASHRAE 2013a) (typically used by codes and policies, but also used by LEED [USGBC 2015])
- Absolute EUI definitions (occasionally used by campuses, regularly used by the General Services Administration; easiest to measure and verify after construction)
- Net zero energy definitions

As noted previously, it is important for the design team to agree to move away from the design practices that have led to poor performance in the past and toward a quantifiable target that is consistent with the available funding.

Cost Management

In the grocery sector, energy is typically the second highest cost of operation (depending on the operating hours), so cost is very important. Some of the strategies that can be used to control costs in energy-efficient grocery design include the following:

- Offset costs of added insulation by reducing the HVAC loads and the number or size of rooftops.
- Offset the additional investment in a high-performance lighting system with reduced cooling capacity.
- Consider fewer, larger rooftop units (RTUs) that can more cost-effectively incorporate advanced HVAC recommendations such as energy recovery and economizers. Overall HVAC system costs may drop even if additional ductwork is required to maintain good air distribution.
- The owner sets the expectation for peak sizing and occupancy loading based on an understanding of peak loading in previous prototype buildings; therefore, mitigate design team sizing risk by considering realistic peak occupancy and internal demands. Often, sizing HVAC systems based on actual loading in previously built prototypes can result in both significant first-cost savings and energy savings.
- Use highly visible, but also effective, "green" measures such as daylighting to provide added benefit as symbols of a grocer's commitment to sustainability and community well-being.

- Achieve lower lighting power densities (LPDs) through careful integration with interior finishes while maintaining desired illuminance levels (determined specifically for the tasks to be performed)—white or brightly colored ceilings, walls, and floors result in better distribution of electrical lighting in the space, which can allow for less overall installed electrical lighting.
- Use light-emitting diode (LED) lights, which have benefits beyond energy savings that can offset additional capital investment required; benefits include increased comfort through reduction of "hot spots" near lights, flexibility of application according to display needs, increased focus of light for improved contrast in merchandise display, and reduced maintenance costs.
- Streamline the envelope construction process through use of modular wall sections constructed off site; additional costs of modular constructions can be offset by construction cost savings.
- Cost-justify shaded overhangs for fully glazed entrance façades by the added value created by a shaded, sheltered entryway that protects customers from the elements.
- Design vestibules to provide a covered placed to store shopping carts, increase employee comfort in point-of-sale areas, and create opportunities for targeted merchandising and advertizing.
- Balance and understand actual maintenance costs with energy costs.
- Leverage purchasing power and direct purchase of specific cost-effective equipment that meets the efficiency requirements in this Guide.
- Use tax and utility incentives and rebates.

Budget Sharing

One oft-heard but fundamentally unnecessary question is "whose budget pays for improved energy efficiency?" The answer should always be, "the owner's budget!" When a team commits itself to delivering low-energy, holistic solutions, it is virtually impossible to distinguish for the accountants how much energy efficiency each trade or discipline "purchased" on behalf of the project through its respective design decisions. A classic example is the cost of building shading: there are increased structural and façade costs, but these may be offset by reduced capital costs for window glazing and air conditioning. These trade-offs are absolutely necessary to explore in consideration of the particular goals and context of the building.

So long as the overall building construction budget remains consistent with the OPR, it doesn't matter where the money was spent as far as energy efficiency measures are concerned. Early understanding of additional capital costs (if any) required for energy measures will allow the owner to consider increases to the building capital budget that will reduce long-term energy use and the life-cycle cost of the building.

Therefore, discipline-based construction budget allocations might be inappropriate for the integrated design paradigm and should be reviewed early in the project. Similarly, it might be argued that traditional fee percentages may also be unintentionally preventing the disciplines most capable of proposing and proving energy reduction techniques from applying their analytical technologies and abilities to the solutions.

Lastly, the EUI budget itself must also be equitably shared. In most grocery stores, refrigeration is by far the largest single energy end use. Improvement of other systems, to reduce space humidity and thus defrost, or to reduce lighting heat gain inside refrigerated cases, can reduce the refrigeration energy component. Legislation and ingenuity have brought us to the point at which most electrical, mechanical, and lighting equipment have been optimized for the current state of technology. Therefore, it is important for design teams to carefully review the relative proportion of energy use by discretionary design choice and collectively attack those portions of the pie chart that represent the largest users. If the team knows that it is accountable to share the responsibility for the end-use energy budget, it sets the tone for sharing the energy savings as well.

Investment Financial Analysis

Many of the examples so far have discussed trade-offs made by the design team to reduce total building energy use. To confirm that each decision contributes to affordable energy savings, energy modeling can be coupled with a series of financial analyses to show which ECM gives "the biggest bang for the buck." The three most typical tools include the following:

- *Life-cycle cost analysis (LCCA)* is a calculation method that adds first cost to a selected number of years of annual energy and maintenance costs, inclusive of equipment replacement costs and an estimate on inflation. The option that has the lowest life-cycle cost is usually chosen if the capital budget allows. LCCA is the financial tool most often used by institutional owners planning to hold and operate a building through a few generations of equipment technology.
- *Simple payback period* is a calculation method that divides the incremental first cost by the net annual operational savings (energy savings and maintenance impact) to determine how long it will take to break even on the investment. The simple payback method is most often used by developers looking to recoup costs before divesting of a property or by long-term building owners with limited funding for retrofits.
- *Return on investment (ROI)* is a calculation that takes the ratio of the energy savings over a predefined number of years minus the capital costs divided by the capital costs:

$$\frac{\text{Energy savings over X years} - \text{Capital costs}}{\text{Capital costs}}$$

where X is a predefined number of years.

This essentially asks "what is my rate of return on the investment?" and it allows a somewhat parallel comparison to the rate of return used in the financial markets. The ROI method is usually used by wealth-holding clients comparing relative opportunity costs when looking to invest in stable profit growth. In downturn economies burdened with the ever-rising cost of energy, some financial institutions have begun to provide financing for energy efficiency upgrades based on projected ROI.

It is important in all of these financial comparisons that the team agrees on what inflation and depreciation rates are appropriate to be used. It is also important to consider the financial implications of potential value-added benefits unrelated to energy use.

Building Configuration and Floor Area Minimization

For first-cost reasons, there is obviously a drive toward the minimization of built square footage, consistent with maintaining the required sales floor area and required back-of-house storage of goods. Compact configuration of back-of-house spaces can minimize circulation area and thus overall building area. In locations with harsh climates, it may be desirable to locate cart storage inside, and extensive vestibules can be used for that storage in addition to providing an "air lock" area to reduce the impact of the harsh climate on interior comfort and energy consumption.

Another major item to be addressed is the architectural configuration of the building. Façade square footage represents a source of conductive heat loss or heat gain as the outdoor air temperatures fluctuate; the larger the amount of façade area, the greater this impact.

Design for Maintainability and Replacement

For a facility to continue saving energy over the building's lifetime, the systems must be properly serviced and the components replaced if necessary. In order for them to be properly serviced, there has to be adequate access to the equipment. Space allocation and system layout should recognize accessibility requirements for both routine and remedial maintenance.

Safety Factors and Diversity Factors

An important part of the integrated design process is the explicit and detailed calculation of loads for sizing of HVAC, refrigeration, and service hot-water equipment. These calculations should incorporate known parameters of the building design and operation with safety and diversity factors to allow for the unknowns of future operation and use. Detailed load calculations are important, both to control first costs of the equipment and to ensure efficient part-load operation. Safety and diversity factors should be determined with the participation of the entire design team, not only to bring the best information to the process but also to ensure that the building will conform to the owner's expectations.

Safety factors are included in load calculations to cover uncertainty with respect to design assumptions. Occupant density, power density of owner-installed equipment, concurrency of peak loads and, for grocery stores, the extent of open refrigerated display cases are the design variables with the most uncertainty. Grocery stores typically have minimal plug loads but may have some special lighted displays. Allowances for these energy consumers and potential sources of heat gain to the space should be identified early in the design process. Store interior reconfiguration occurs occasionally, and allowances should be made for the heating or cooling load implications of a defined range of interior reconfiguration. Reconfiguration will likely only have a significant impact on the internal thermal balance of the space if it involves the extent of refrigerated cases, of food preparation, or of heated ready-to-eat food displays. Measuring actual plug load power density in the existing stock of a grocer's portfolio provides a good basis for estimating this design parameter, and a safety factor can be added to cover future additions or modifications. Loads should be calculated with the best possible specific information and safety factors applied only to cover uncertainty related to specific assumptions.

Diversity factors are very different from safety factors in that, rather than making an allowance for unknown variation, they reflect the fact that individual load peaks are typically not concurrent. For instance, in HVAC and electrical design, it is quite common to find the following diversity factors applied:

- Diversity assumptions about occupancy
- Diversity assumptions about the fraction of time that doors of refrigerated cases are open
- Diversity assumptions about utilization of food preparation and cooking equipment

It is important to note that diversity factors are independent of schedules and as such must be reviewed with the schedules to ensure that the appropriate level of fluctuation is accounted for only once (especially when the schedule is a percent of load type schedule). It is important to understand the impact of schedule and diversity factors on the internal thermal balance of the store. During high-traffic periods, intensive access to refrigerated cases may actually decrease space temperature and increase space heating required to maintain comfort. Sizing of cooling requirements in grocery stores is mostly linked to ventilation and dehumidification requirements and to heat gain from people during densely occupied periods.

The fact that most of the façade of a grocery store is opaque and that, other than cooking, there is little internal heat gain eliminates most of the opportunities for overstating internal cooling loads. With HVAC equipment selected with good part-load operating characteristics, the danger of oversizing is reduced. The ability to provide dehumidification during part-load conditions is the most important characteristic of the HVAC system in many climates.

Schedules of Occupancy and Use and Utility Rates

It is important for the team to map out the anticipated schedules of occupancy and use for each area of the building. This information is crucial to the energy modeling and can greatly affect the outcomes with regard to estimated energy savings over a known benchmark or through LCCA. It is important to note that most energy models run the same schedule week after week, so schedules not only should be configured to cover typical weeks but also should be changed to account for any known periods of building closure or times of higher than normal building occupancy, such as the holiday shopping season.

Another item to bear in mind regarding scheduling is whether a standardized schedule will be imposed on the energy model through regulatory requirements. For compliance modeling, in particular, some codes such as California's *Building Energy Efficiency Standards for Residential and Nonresidential Buildings* require that prescribed schedules are used instead of a schedule grounded in a realistic review of actual prototype use (CEC 2013). It is important for the entire team to be aware of any such constraints ahead of time. If the likely schedules for the proposed facility differ markedly from those prescribed by the local energy code, life-cycle costing exercises should be performed using the likely schedule, while code-compliance modeling should use the prescribed code schedules.

It is essential that the team understand the applicable utility rates, especially any embedded demand charges and on/off/high/low/seasonal peak-period definitions local to the site and its service utility, as well as any service riders that might affect the choice of systems. Energy rate structures are important because the prevailing benchmark for energy savings in Standard 90.1 and most energy codes are based on annual cost, not absolute energy savings. While the intent of this Guide is to assist owners in achieving 50% energy savings with respect to ASHRAE/IESNA Standard 90.1-2004, energy cost savings are most important to the owner. Some energy cost reduction measures, especially those that may mitigate electric peak demand charges, may actually result in a slight increase in energy consumption. Familiarity with how the owner will purchase energy for the store will better enable the designers to find energy cost reduction opportunities.

Standby Capacity Sizing Protocols

To ensure service continuity during component failure or routine maintenance, standby capacity for the facility should be thoroughly discussed by the owner and the design team early in the design process. Specifically, the systems requiring continuous service, the potential threats to that continuity, and the means for ensuring continuity should be identified. Often continuity of service is ensured by providing duplicate components or spare capacity within a system so that it can continue to operate at an acceptable level despite the failure of some portion of the system. The type of continuity should also be identified. For some systems, operation should be seamless across a failure while, for other systems, rapid restoration of service at the required level is acceptable.

Provision of redundant capacity sometimes improves energy efficiency during normal operation, especially if components, such as air-cooled chillers, operate more efficiently at part load than at full load. The designer should take care, however, to ensure that the enlarged system can operate efficiently at low part loads. Methods for providing redundancy for refrigeration systems can be somewhat different and sometimes integrated into the design of how the loads are distributed. For example, owners may choose to distribute refrigerated case loads across two or more systems so that if one system goes down, the product can be migrated to freezer boxes or other frozen areas with available storage space. Where there is a requirement for excess capacity (possibly for future expansion), designers should design carefully so that the systems can still effectively and efficiently meet the *actual* load. Also, designers should note that safety factors are applied during component and equipment sizing and that some of these are iterative. When determining the extent of required redundancy, these safety factors should be considered as preexisting redundant capacity to avoid gross oversizing of systems.

In grocery stores, the system with the greatest need for continuity of service is food refrigeration, which does not require seamless operation across an emergency but must avoid inoperability for extended periods of time to minimize product loss. The ability to restore service rapidly may not require significant redundancy or oversizing of equipment but may be achieved by provision of spare refrigeration compressors or storage of critical spare parts on site to ensure that the loss of service from equipment malfunction is brief. Many stores use standby electric generators to provide power to portions of the refrigeration systems during utility outages. In some jurisdictions, these standby generators may be operated in parallel with the utility electric service during peak demand periods to reduce the building electric demand and the resulting demand charges. Review of service continuity requirements with a mind toward energy efficiency and energy cost reduction often results in achieving both goals.

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Walmart—A Case Study

The Walmart in Centennial, Colorado, was retrofitted as part of the U.S. Department of Energy (DOE) Commercial Buildings Partnership program (EERE 2015a). Planning began in 2009, and energy efficiency measures were installed in late 2012 and early 2013. The 213,000 ft^2 food store includes grocery, general merchandise, service deli, and restaurant areas. The store has refrigerated cases in both the grocery area and the stockrooms.

The project was guided by the overall goal of implementing energy efficiency measures that save energy for this store and that can be replicated by other stores. Another goal of the project was to integrate the operation of the building systems to optimize for cost and performance. Energy modeling was used during the design process both to verify targeted energy savings and to screen efficiency measures. Measures that would negatively impact customer experience were not considered feasible.

Refrigeration

The energy savings from the refrigeration energy efficiency measures were due to the measures themselves as well as a reduction in the heating energy required for the sales area. These measures include the following:

- Glass doors and LED lighting were added to medium-temperature dairy, deli, and beer vertical multishelf cases.
- Electronically commutated (high-efficiency) evaporator fan motors were installed in all low-temperature and medium-temperature walk-ins.
- The anti-sweat heater controls for low-temperature cases were upgraded to cycle based on dew-point temperature. Previously, the controls operated on full power at all times.

Lighting

Parking lot lighting was upgraded to LED fixtures, but the most cost-effective lighting upgrades were the interior lighting measures, including a reduction in the overall lighting power density (LPD) to 0.73 W/ft^2. Additional measures include the following:

- In the produce area, 48 100 W metal halide fixtures were replaced with 96 12 W, 1000 lumen LED spotlights.
- Additional LED light upgrades include replacement of 70 W metal halide canopy downlights with LED fixtures and 175 W wall-mounted metal halide lamps with recessed LED fixtures.
- Perimeter lights in the general merchandise areas were delamped (half of the lamps were removed in each fixture) while maintaining brightly lit walls and signage.
- Occupancy sensors were installed in the back of house to turn off lights when areas are unoccupied.

Centennial, Colorado, Walmart Store

HVAC

While lighting improvements can lead to increased heating energy, this effect was offset by using refrigeration heat recovery. HVAC changes include the following:

- Waste heat from the two medium-temperature systems was used to preheat ventilation air for the grocery area of the store served by two air-handling units (AHUs).
- Direct-expansion (DX) condensing was combined with indirect evaporative precooling of the ventilation air on six of the 20 ton rooftop units (RTUs) serving the sales are of the store.

Energy Savings

The energy savings for this project were modeled in comparison to an ASHRAE/IESNA Standard 90.1-2007 baseline and showed a $258,000 expected annual energy cost savings as compared to that baseline.

Additional information on the Walmart Centennial, Colorado, store and the lessons learned from this project can be found in the Buildings Database on the EERE website: https://buildingdata.energy.gov/ (EERE 2015b).

12 W LED Spotlights

LED Parking Lot Lights

Waste Heat Recovery System Glycol Pump

Delivery of Glycol to AHU

Photographs reproduced courtesy of NREL, credit Dennis Schroeder

Design Concepts and Practices

3

INTRODUCTION

Grocery stores have long hours of operation and year-round refrigeration with the potential for heat recovery, which sets them apart from other commercial retail buildings. System interaction is more profound and more important to understand and account for in design. Many systems, including lighting, dehumidification, ventilation, hot water, and space, demand a closer look at the indirect impact they have on other systems in the building. In many cases, there are opportunities for energy savings when the systems are not looked at independently but rather designed to work together.

This chapter addresses specific methods and promotes several design philosophies that can be used to achieve more effective and efficient design of grocery stores. Much of Chapter 3 is devoted to refrigeration, as well as the interactions between refrigeration and HVAC systems. Predominantly, refrigeration systems are custom designed and built to react to all the countless ways of arranging and merchandizing refrigerated product on a sales floor. Additionally, technology is ever changing due to new energy and environmental requirements that can sometimes change design paradigms. Conventional rules of thumb may no longer be applicable and need to be challenged. The choice of refrigerant and of direct or indirect cooling systems is a significant challenge, with numerous trade-offs between energy efficiency, first cost, and environmental consequences.

While this Guide focuses on energy efficiency, this chapter also includes information on refrigerant global warming potential (GWP), the use of natural refrigerants, and refrigerant charge as they relate to direct greenhouse gas emissions. Because refrigeration systems must necessarily meet product temperature requirements 100% of the time, the effect of oversizing on efficiency is a potential concern. Safety factors and derating factors are examined and discussed, with examples to promote design that incorporates a deeper understanding of actual hour-by-hour operation of components and systems. Information is also included regarding how to evaluate and choose the condensing method for a particular store.

In addition, several design approaches and technology concepts are described as paths to more accurate design and greater understanding of system performance and energy efficiency through the year. While some approaches are constrained by the lack of information from original equipment manufacturers (OEMs) and may take time to develop, they have the potential for reduced capital cost over time as well as better performance and lower operating costs.

BUILDING SITE AND DESIGN INFLUENCES

CLIMATE CHARACTERIZATIONS

Understanding efficiency opportunities and challenges by climate zone is necessary for a grocer to reach advanced levels of energy savings. Store designers should recognize that energy efficiency strategies differ from climate zone to climate zone. The specific weather variables that affect the energy consumption of the store are expressed by annual or seasonal metrics as follows:

- The intensity and length of the heating season are represented by heating degree-days (HDDs) as shown in Figure 3-1.
- The intensity and length of the cooling season are represented by cooling degree-days (CDDs) as shown in Figure 3-2.
- The consistent intensity of the sun's energy is represented by the annual solar radiation as shown in Figure 3-3.
- The worst case for removal of airborne moisture (i.e., dehumidification) is represented by the design dew point as shown in Figure 3-4. Bin data for dew-point temperature or moisture ratio are necessary to fully evaluate annual dehumidification requirements.
- The potential for evaporative cooling or heat rejection is represented by the design wet-bulb temperature as shown in Figure 3-5.

In combination, these variables show that distinct patterns emerge with regard to climate types, as shown in Table 3.1, each of which has particular energy impacts on building design and operation. The U.S. has been divided into eight primary climate zones for the specification of design criteria in major energy codes such as ANSI/ASHRAE/IES Standard 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings* (ASHARE 2013a), and ANSI/

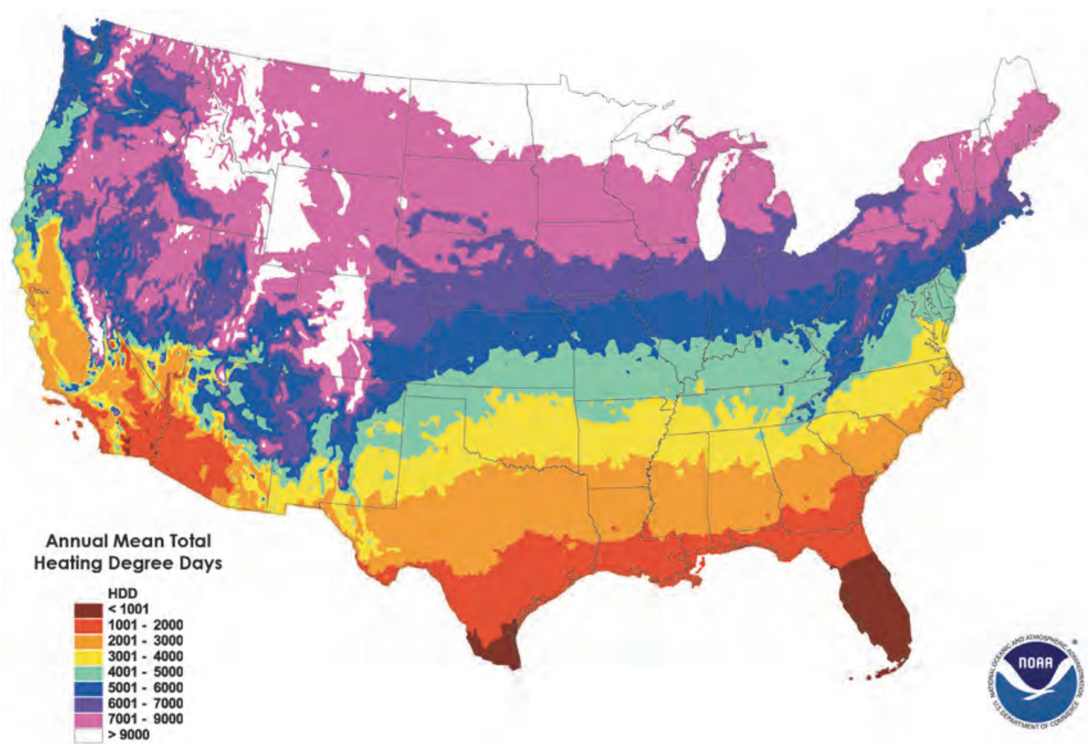


Figure 3-1 Heating Degree-Days
Source: NCDC (2005)

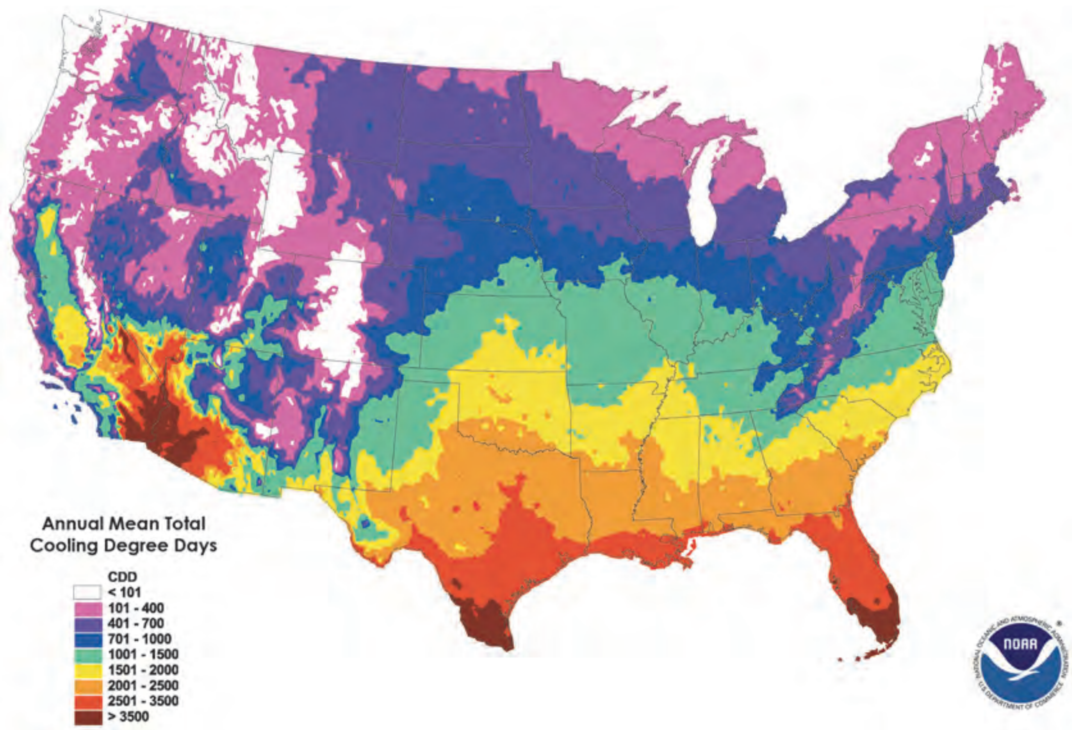


Figure 3-2 Cooling Degree-Days
Source: NCDC (2005)

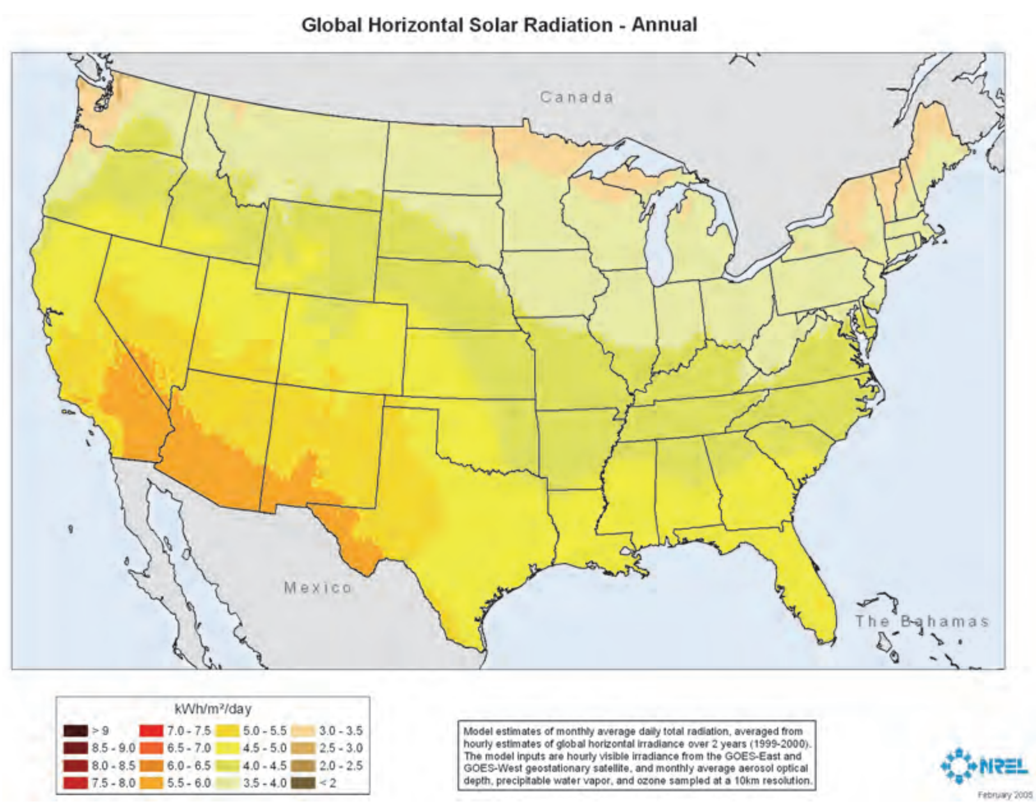


Figure 3-3 Annual Solar Radiation
Source: NREL (2005)

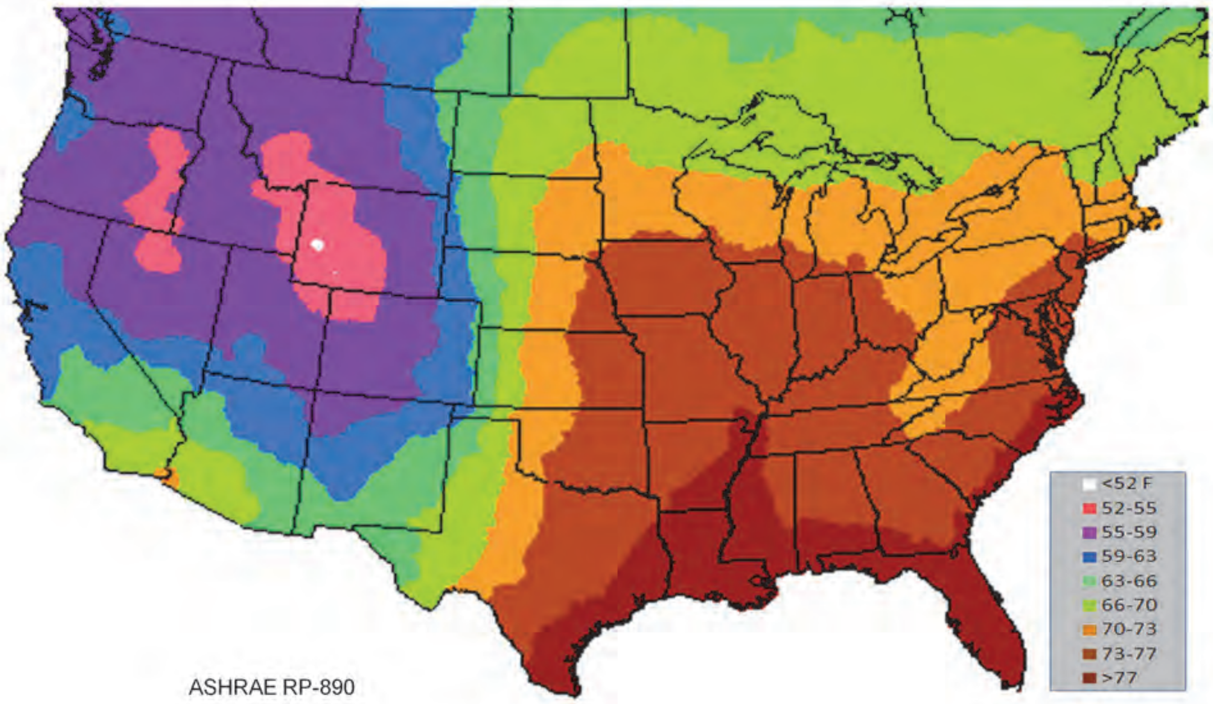


Figure 3-4 Design Dew-Point Temperatures
Source: Colliver et al. (1997)

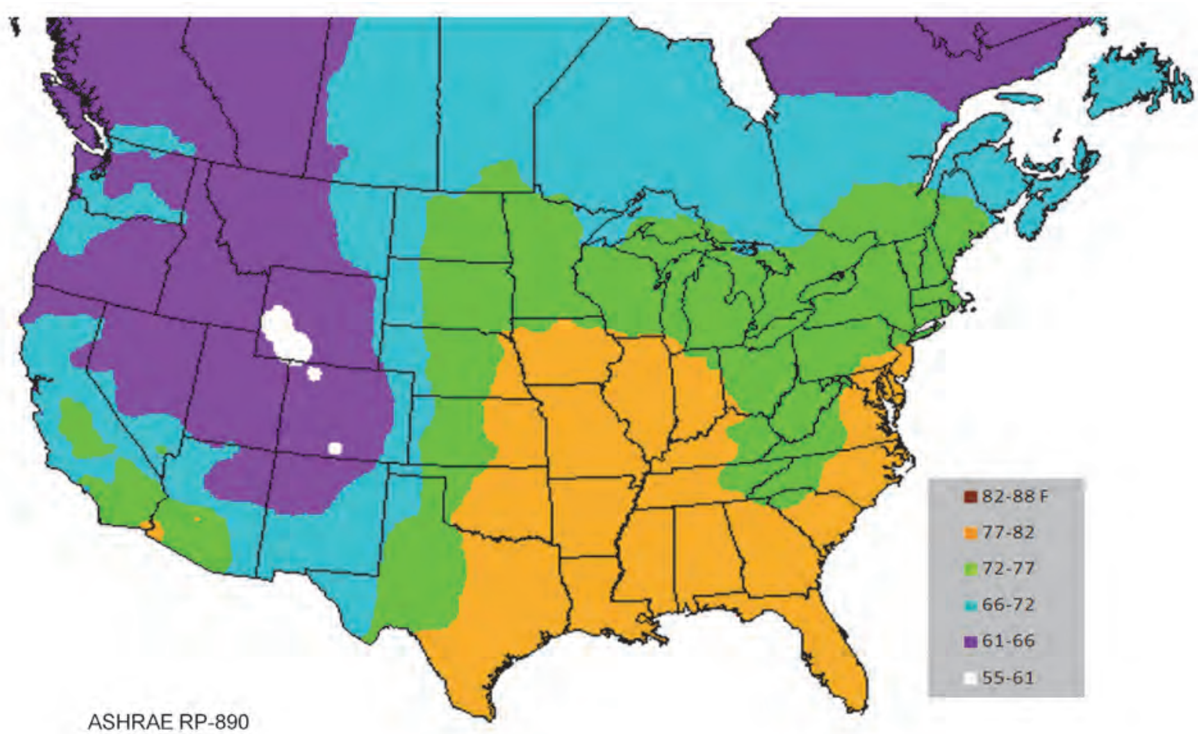


Figure 3-5 Design Wet-Bulb Temperatures
Source: Colliver et al. (1997)

ASHRAE/USGBC/IES Standard 189.1, *Standard for the Design of High-Performance Green Buildings Except Low-Rise Residential Buildings* (ASHRAE 2014). Figure 3-6 shows these climate zones as compared to CDDs and HDDs.

The characterization of these climate zones is based on seasonal performance metrics, not on peak or design values. Each climate zone is clustered by HDD65 for heating and CDD50 for cooling; these zones are further subdivided by moisture levels into humid (A), dry (B), and marine (C) to characterize their seasonal weather values (see Figure 3-6). Sixteen cities have been identified as sufficient to represent the total variation in climate, as shown in Table 3-1.

It is important for the design team to determine the unique characteristics of the climate closest to the site. Annual hourly climate data are usually used for energy modeling and are available from federal government sources (see the "Weather Data" heading at energyplus.gov for a complete list). In addition to the acquisition of local data, it is necessary to assess any local topography or adjacent properties that may cause reduction in access to sunlight for daylighting and passive solar heating.

Table 3-1 Cities Characterized by Climate Combinations

Climate	Hot	Mild	Cold	Very Cold	Extremely Cold
Humid	Miami—1A Houston—2A Atlanta—3A	Baltimore—4A	Chicago—5A Minneapolis—6A		
Dry	Phoenix—2B Los Angeles—3B (coastal) Las Vegas—3B (others)	Albuquerque—4B	Denver—5B Helena—6B	Duluth—7	Fairbanks—8
Marine		San Francisco—3C Seattle—4C			

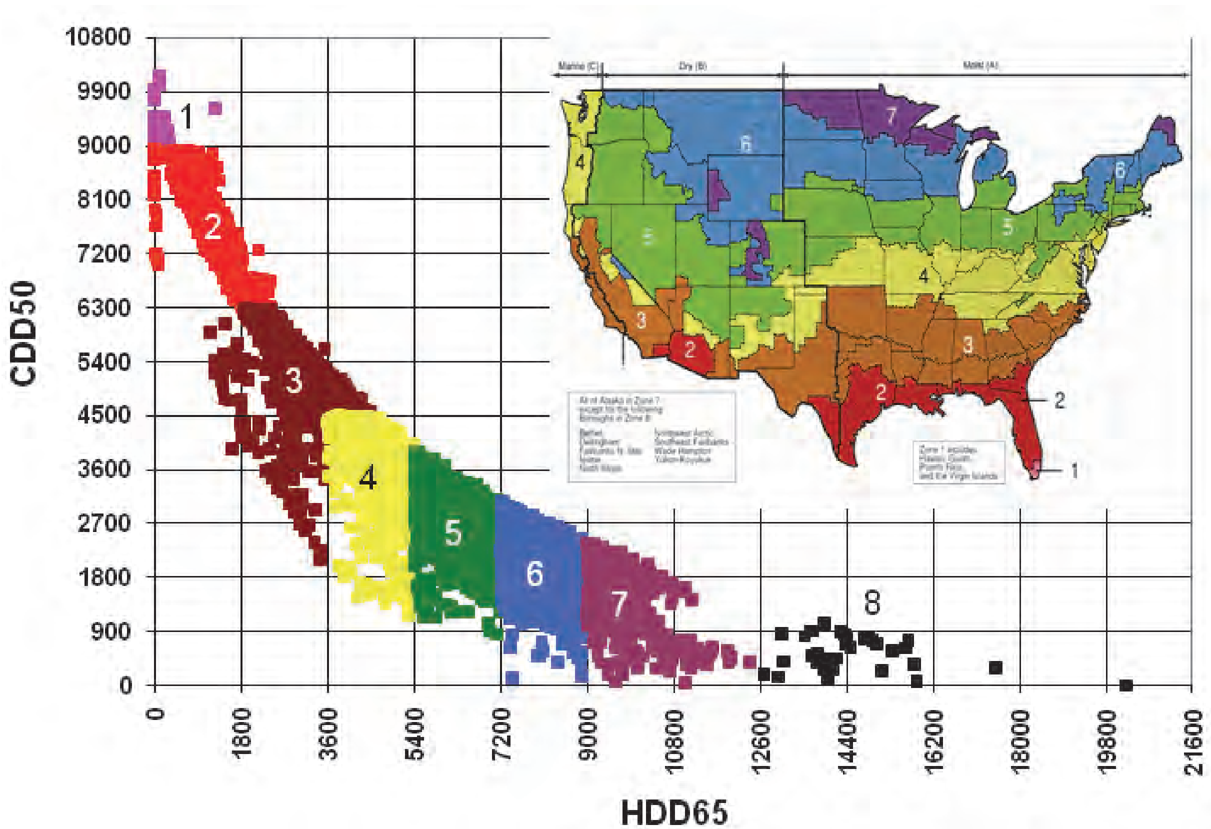


Figure 3-6 U.S. Primary Climate Zone Map

CLIMATE DEPENDENCE

Multiple combinations of climate conditions influence the energy performance of a building. Grocery stores differ from other building types in that the annual distribution of heating and cooling requirements for the building are more dependent upon the extent of open refrigerated cases than on the climate in which the building is located. The annual energy use for the facility will be a product of the extent to which outdoor air requires cooling for dehumidification and subsequent heating to maintain space minimum temperature conditions. Humidity control for the sales area requires that outdoor air be delivered to the space with a dew-point temperature no higher than 52°F, preferably lower, so that whenever the outdoor air has a higher dew-point temperature, it must be dehumidified to that dew point or below. To maintain comfort and adequate ventilation to the space, this cool, dehumidified supply air may require reheat. Stores in warm, humid climates with extensive open cases and with extensive cooking operations requiring large amounts of exhaust will have the highest energy consumption. The heat released by the interior lights, plug loads, and fans adds to the cooling load and diminishes the heating load, while the heat extracted from the space by the refrigerated display cases increases the heating requirements. In some cases, the heat gain removed from the space by improving the lighting system must be replaced by reheat within the air-conditioning units.

Figure 3.7 shows the energy end-use breakdown for the baseline case and the recommended design for one of the recommended HVAC system types in the 16 climate zones. Details for the recommended design are shown in the Chapter 4 recommendation tables and the Chapter 5 how-to recommendations. Figure 3.7 shows energy consumption for a store with mixed-air (MA), single-zone variable-air-volume (SZVAV), direct-expansion (DX) packaged rooftop air-conditioning units (RTUs) with an indirect gas furnace. Fundamentally, what can be seen in Figure 3-7 is as follows:

- Lighting, plug, and fan loads are constant inputs for the baseline case and therefore are very consistent in the energy use intensity (EUI) budget. Indeed, the only fluctuation most likely occurs from fan energy responding to on/off controls in response to climate.

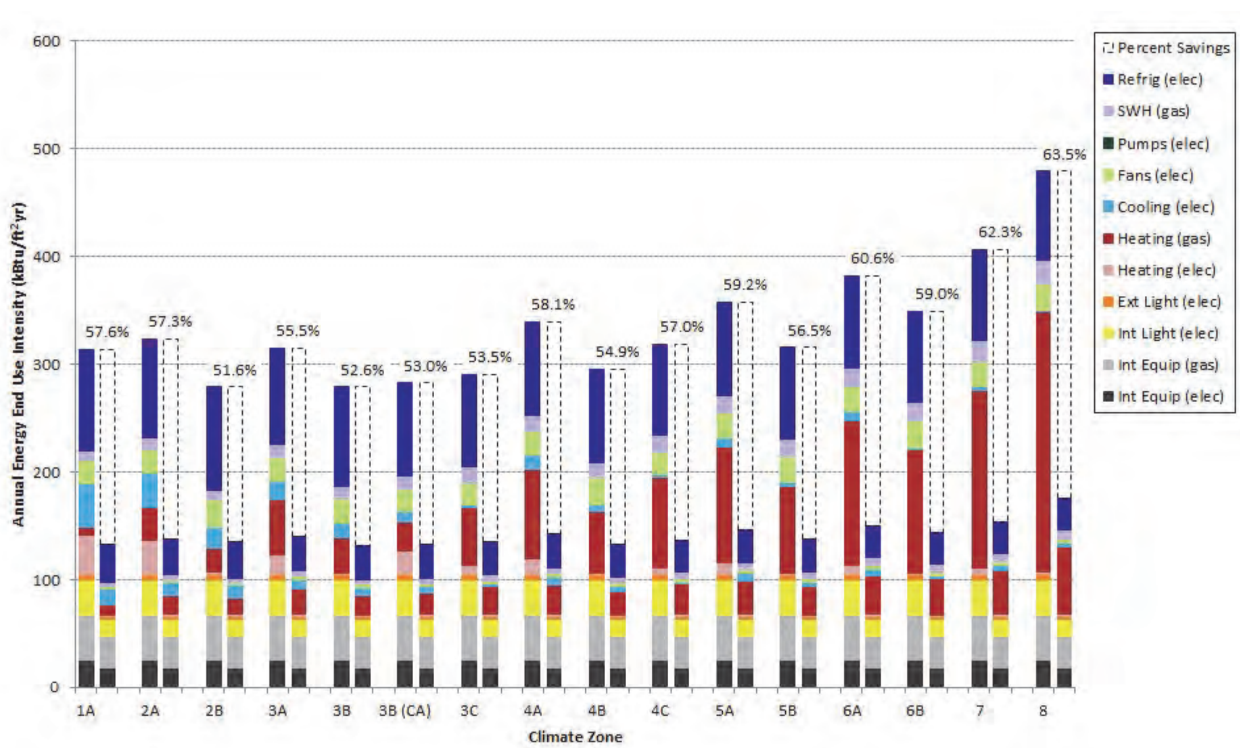


Figure 3-7 Energy Breakdown for Mixed-Air SZVAV DX Packaged RTU

- Heating energy for the baseline case increases in both warm humid climates and cold climates because of the need to heat supply air to the space, either because the air is refrigerated for dehumidification or because of the low temperature of the outdoor air. Lowest energy consumption is found in coastal zones 3C, 4C, and 5C, which also offer the greatest challenge for meeting the 50% energy conservation goal.

BUILDING ORIENTATION

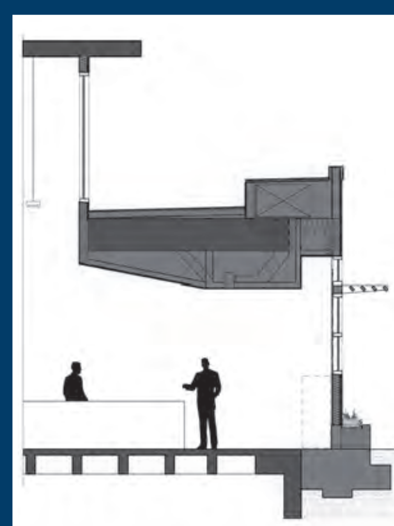
Typically, the only glazed façade of a grocery store is the entrance façade. This façade can provide daylighting for cart pick-up and check-out areas. Should the entrance façade be west-facing, canopies that might also provide rain or sun protection for customers entering or leaving the store can also provide sun and glare protection for those spaces immediately inside the glazed façade.

The configuration of most grocery stores minimizes the opportunity for daylight sidelighting through windows. Clearstory windows or windows at higher heights above shelving are one daylighting method. Since most stores are single story, daylighting with skylights is an effective energy-conservation strategy. As a result, wall orientation and massing may be less important for this building type than for some others. See DL1–DL3 in Chapter 5 for more information.

Daylighting in Grocery Stores

The Shattuck Avenue Safeway store in Berkeley, California, underwent a complete renovation and major expansion in 2012, increasing the original 25,000 ft^2 store to 45,000 ft^2. One objective of the project was to allow for increased daylighting elements in the store.

The existing store had an iconic barrel vault with a clerestory window. During the expansion, the barrel vault was lengthened and another clerestory window was added. A 13 ft square skylight was also added above the stair that connects the sales floor to the parking garage, providing shoppers with an appealing link through the vertical circulation. Openings from the stairwell into the store allow the sales floor to also benefit from this daylighting.



Corner Entry Showing Barrel Vault with Clerestory Window
Photograph courtesy of Nick Gomez, Lowney Architecture, and drawing courtesy of Ken Lowney, Lowney Architecture

The expansion of the existing store included a new segmented elevation along Shattuck Avenue leading into a corner entry. The design of this wall allows for floor-to-ceiling windows within the segmented sections and punched windows on the street side of the wall. The punched windows allow daylight into the Produce and Bakery departments, give pedestrians clear views into the store, and provide shoppers views out to the Berkeley hills. The sill height of the punched windows was coordinated to work with the product shelving on the interior.

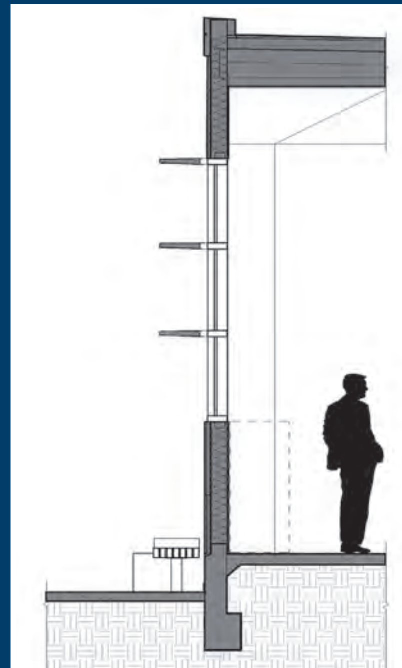
Corner Entry and Segmented Walls along Shattuck Avenue Sidewalk

Photograph courtesy of Nick Gomez, Lowney Architecture, and drawing courtesy of Ken Lowney, Lowney Architecture

Daylighted Stairwell

Full-Height Windows

Photographs courtesy of Nick Gomez, Lowney Architecture



Punched Windows with Sill Height Above Product Shelving
Photograph courtesy of Nick Gomez, Lowney Architecture, and drawing courtesy of Ken Lowney, Lowney Architecture

REFRIGERATION DESIGN PHILOSOPHIES

Refrigeration systems consume approximately half of the total energy consumed by a typical grocery store, and compressors are the largest energy-consuming component within a refrigeration system. Therefore, many recommendations are targeted at reducing compressor energy. Primarily, there are two ways to reduce compressor energy—by reducing system load and by reducing compressor lift.

Reducing system load is probably the simplest and sometimes easiest way to save energy. If the load can be reduced (either in operation or in design), the system simply won't need to run as many compressors to meet the load. Adding doors to open cases is a prime example of how to reduce system load.

Reducing compressor lift means reducing head pressure and/or increasing suction pressure. By bringing these two pressures closer together, compressors don't need to work as hard to "lift" the refrigerant gas from low pressure to high pressure. There are several design techniques and recommendations presented in this Guide that help to increase suction pressure, reduce head pressure, decrease load, or provide a blended benefit between all three.

IMPACTS OF REFRIGERANT SELECTION

Although the recommendations of this Guide do not include efficiency gains due to a refrigerant type, there are several refrigerant options to choose from that may affect system architecture and operation. Additionally, refrigerant choices can affect system capacity, which can create efficiency penalties if not accounted for in design.

Global Warming Potential

The biggest driver currently pushing users to change refrigerants is the direct affect the refrigerant has on greenhouse gas emissions. Since the total global warming potential (GWP)

Table 3-2 Common Refrigerant Classifications

Refrigerant	GWP	Class
R-404a	3920	A1
R-407a	2107	A1
R-407f	1824	A1
R-410a	1725	A1
R-134a	1300	A1
R-32	650	A2L
Propane	3	A3
CO_2	1	A1
Ammonia	0	B2L

Data source: Bitzer (2012)

of the system is determined directly through refrigerant leaks and indirectly through energy use, they both must be considered factors. Because of this dynamic, it is more justifiable to focus on the achievable total equivalent warming impact (TEWI) of an entire system and not just the GWP of the refrigerant used. TEWI calculations account for the direct and indirect impact to greenhouse gas emissions. Table 3-2 shows some common refrigerants with their associated GWPs and classes.

To make any sound design decisions based on TEWI, owners need to communicate the importance of a low-TEWI system as it relates to company goals, philosophies, and other weighted factors. A low-TEWI system may not be the *most* efficient system available, but it should be a very efficient system, nonetheless.

From an energy standpoint, some of these refrigerant types require specific system architectures that may carry efficiency implications. For refrigerants that must be utilized in a secondary system or even a cascade arrangement, there can be efficiency impacts due to the addition of refrigerant pumps or the introduction of additional steps of heat exchange, for example.

Traditionally A1 refrigerants (refrigerants with lower toxicity and no flame propagation) have been used in grocery stores. However, other classes, such as A2L, B2L, and A3 refrigerants (which may be toxic or have a lower or higher flammability) have been gaining more attention, due not only to reduced GWP but also to improved efficiency potential. As these reduced-GWP refrigerants are more frequently used, special attention should be paid to the achievable efficiencies realized in actual system operation. It's not hard to apply an "efficient refrigerant" to an inefficient system. Similarly, an efficient system can become inefficient if an "inefficient refrigerant" is used. Systems must be designed for the refrigerant used in order to maximize operational efficiency and reduce the indirect contributions to global warming as much as possible.

Designing with High-Glide Zeotropes

Synthetic refrigerants that exhibit significant degrees of temperature glide are becoming more common. Most of these refrigerants accepted today can operate with comparable efficiency levels to those being replaced. However, if the dynamics of temperature glide are not understood, it becomes easy to make design errors that can induce system efficiency losses.

Because compressor capacity ratings are determined and published at pressures expressed in dew-point temperature, many designers also size condensers using dew-point temperatures without significantly derating condenser capacity. For example, assuming 8 degrees of glide in the condenser, a derating factor in the range of 30% to 40% would be necessary, depending on the design temperature difference (TD) that is being targeted, to determine the actual heat rejection capabilities of a condenser when using dew-point temperatures to calculate condenser capacity. Design condenser TDs based on dew point cannot be maintained in actual operation since the condensing temperature will only decrease from dew point through the condensing process due to glide. The resulting average TD within the condenser will be much less than the design TD and, in addition to the obvious capacity issues, will result in an energy penalty. Depending on how the system is set up in the field, either condenser fans will run harder in attempt to achieve the condenser TD setpoint or the TD setpoint will be satisfied and the compressors will work harder due to an increase in head pressure.

On the low side of the system, more inefficiency is created if superheats are not adjusted properly. Refrigerant temperature rise at an evaporator outlet has always signified the presence of superheat with traditional refrigerants, whereas conditions may still be saturated with zeotropes. Liquid refrigerant that is allowed to exit the evaporator increases the required mass flow of the system and may result in an energy penalty at the compressor.

Compressor Systems with Ammonia Refrigerant

For those that seek to maximize system efficiency by leveraging more efficient refrigerants, ammonia (R-717, NH_3) is becoming a more frequent consideration due to its reputation as a highly efficient refrigerant. Some may be surprised to find that NH_3 has successfully been used in grocery stores as an efficient natural refrigerant option. In fact, the U.S. Environmental Protection Agency (EPA) Significant New Alternatives Policy (SNAP) program (EPA 2014) had approved the refrigerant for grocery store use well before it had been attempted for use in such stores. As an ASHRAE-classified B2L refrigerant (ASHRAE 2013b), R-717 is recognized as toxic and mildly flammable, so it must be used in an indirect system. As a result, an outdoor or rooftop system is the most feasible option.

There is obvious benefit in minimizing the ammonia refrigerant charge for safety reasons; however, designers should be careful not to sacrifice system efficiency or reliability in an attempt to minimize charge. Ammonia is best suited for, and operates most efficiently and reliably in, a flooded or overfed system. Most grocery stores could apply ammonia in this fashion and realize R-717 charges below 300 lb. Dry expansion systems that can maintain the reliability and inherent efficiencies of ammonia may be another attractive alternative due to their ability to achieve charge levels below 100 lb.

Rooftop Ammonia Refrigeration System
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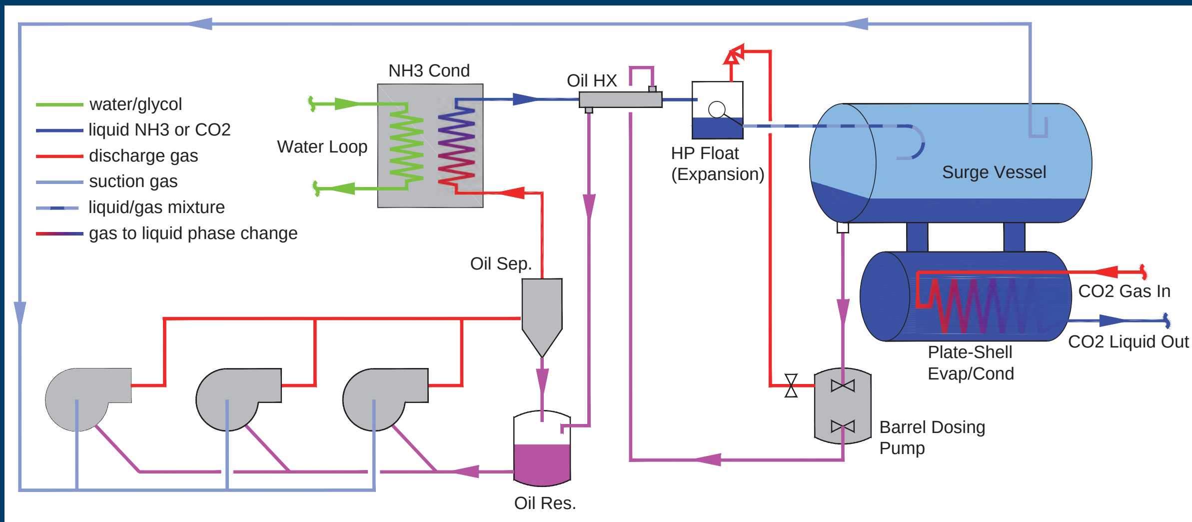

Ammonia Flooded System Type Schematic
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The schematic included here illustrates an ammonia-flooded system type that has been used successfully in a U.S. grocery store. Notice that there is no high pressure receiver and that condensed ammonia liquid can be expanded directly to the cascade heat exchanger. Since the compressors pull from the top of the surge vessel, which is maintained at saturated conditions, there is a negligible amount of superheat present at the compressor inlet. Additionally, the evaporator remains fully wetted, allowing for very effective heat transfer.

Due to the undesirable reaction NH_3 has with copper, standard compressors must be open drive. This difference actually results in additional energy savings for the system. Since motor heat can be dissipated to the air and not carried to the condenser by the cold suction gas, there is a slight efficiency gain at the condenser. Overall, due to the dynamics discussed here, ammonia compressor systems have the potential to operate up to 25% more efficiently than some of the more common hydrofluorocarbon (HFC) compressor systems used today. The biggest hurdles facing wider adoption of ammonia systems commercially are the refrigerant's safety classification and system cost. However, with more of these low-charge ammonia systems planned for installation, not only will the cost become more competitive, but confidence will grow in the ability to safely apply ammonia commercially.

For more information regarding the application of ammonia in U.S. grocery stores, refer to the International Institute of Ammonia Refrigeration (IIAR) technical paper "Feasibility of Ammonia in U.S. Supermarkets" by Caleb Nelson, presented at the 2012 IIAR Conference in Milwaukee.

SIZING CONCEPTS

One of the most important refrigeration design tasks is to correctly size systems for their intended function and purpose. To do this, rely on the compressor manufacturer's rated capacities as a foundation for equipment selections. Depending on the nature of the system, the intended function, and even the specific end user, factors for safety, derating, prorating, fouling, etc., may be applied to the sizing process as well. Although these factors are often necessary and appropriate, improper understanding and application can be detrimental to the system performance and efficiency.

Many times end users require systems to be intentionally oversized to accommodate possible future load increases. If systems must be oversized for this reason, special attention to part-load operation becomes even more critical. The use of electronic valves and linear compressor staging technology can be very effective in helping systems maintain proper operation and expected efficiencies at part-load—especially when oversized.

Standardized Design Concepts

Standardization in design offers many benefits but requires careful application. Benefits include faster construction, first-cost reduction, easier service, better understanding of equipment, and easier application of successive technical advancements. However, a one-size-fits-all approach to design can be inefficient and costly when applied too broadly or when resulting in oversized equipment and use of technologies that are not effective or are unnecessary in particular climates. With a single design for all situations, when something doesn't work, the design must be increased to meet the shortfall, even though it may only be applicable in certain locations.

The ideal approach includes standardized design elements that can be selected where they are most appropriate, retaining the benefits of standardization with even greater gains in performance and cost-effectiveness on a store-by-store basis. Achieving this goal can be a long-term effort with a wide range of relevant factors that varies from chain to chain, including store types, climates, utility rates, and operational requirements. A key tool in developing a standardized design approach is building modeling, which allows evaluation of design approaches and sizing concepts across multiple climates, utility rates, and operating assumptions to determine where design variants are necessary to meet performance and reliability needs or offer a sufficient advantage in operating costs.

How to Fine-Tune Refrigeration System Sizing versus Loads

Refrigeration system design in grocery stores can be fine-tuned over time by studying operating information from new and existing stores. Chain stores have multiple stores across varying climate zones, offering a resource for better understanding actual versus expected system operation during weather extremes and with varying operating conditions through the use of data from energy management systems (EMSs) and refrigeration control systems. With an understanding of compressor ratings and adjustments for actual operating conditions, compressors are a reasonable means of determining actual cooling load. Observed operation during available load conditions and weather examples may still require extrapolation to design conditions that occur infrequently. If the cooling system does not meet required operating conditions at peak loads and weather extremes, even though there is a safety factor in the design, an engineering study is needed to determine the system components that are limiting performance, which often involves a detailed understanding of the differences in operation at rated conditions and actual system conditions. For example, particularly for medium-temperature refrigeration and air-conditioning systems, at extreme conditions, a large fraction of the available capacity is unused. On the other hand, low-temperature systems are often observed with all compressors running, even when not at peak loads, even though the design includes a substantial safety factor. The objective should be to understand which components have excessive safety factors rather than simply downsize the entire system—and understand why components fall short of expected capacity and performance. Understanding is often gained by examining component performance separately. For example, compressors and condenser capacity can be considered separately as well as looking at the balance operating conditions.

In addition to observing system performance parameters, study of alarms (when available), service calls, and maintenance history is particularly valuable in considering design choices and safety factors. Any event that results in loss of adequate refrigeration affects product quality and store sales, particularly outages where product is lost or removed from the display case. This information can be studied to determine the root cause and whether it could be avoided with additional capacity, backup, bypass, or redundancy.

The effect of component failure, e.g., the loss of one compressor, should be evaluated explicitly. Rather than applying an overall safety factor that envelops all potential equipment failures, designing for outages at a component level may allow reduction in overall safety factor. For example, reserve compressor capacity for system reliability does not need to be counted in the heat rejection capacity.

Safety Factor and Equipment Derating

Much is made of the unnecessary and expensive safety factor common in refrigeration. One example seen in many stores is medium-temperature parallel racks that may never run more than 50% capacity even during peak conditions. Meanwhile, in the same store, the low-temperature parallel rack compressor systems may have a 25% safety factor on paper but have all compressors running, even at below-design weather conditions. Compared with HVAC chilled-water systems, where water flow and temperatures are easily measured, measuring refrigerant flow and enthalpy differences is quite difficult, to the extent that accurate measurements of refrigeration loads and equipment performance in grocery stores is virtually nonexistent.

Safety factors should be explicitly applied and observable. Explicit safety factors may mean deciding that condensers need a purposeful factor to allow for scaling or recirculation. Or, it may mean that a compressor system should meet design load with one compressor not in operation (if that is the owner's design expectation). An observable safety factor means that if a compressor rack is intended to have 20% safety at design conditions, this should be observable with operating history over a period of time.

Commercial refrigeration condenser and evaporator coils do not have published ratings that reference a rating or test standard. This is in contrast to HVAC equipment, which is commonly rated to Air-Conditioning, Heating, and Refrigeration Institute (AHRI) standards and moreover is certified to be accurate. Comparative differences between manufacturers' equipment are relatively obvious and tend to normalize, but the true performance of equipment is often far short of the catalog numbers, at least to the extent performance can be measured in the field. These factors are largely "baked into" grocery industry practice through the process of iterative adjustment.

Reducing oversizing should be undertaken carefully. First, it is important to understand equipment derating factors for more accurate engineering. These factors may apply to design conditions or to hourly operating conditions or to both. While hourly energy modeling has not been widely used in the refrigeration industry, it can help improve equipment design decisions. The following sections detail areas where performance derating may be considered.

Off-Design Conditions

Refrigeration equipment is selected first and foremost to maintain perishable foods at safe temperatures, even during the most stringent weather and operating conditions. Because these worst-case conditions may occur only for a few hours a year, operation during average conditions is far more important from an energy-efficiency standpoint. Most refrigeration equipment is not published or certified to a rating standard even at design conditions. Moreover, performance at off-design conditions is rarely provided, so assumptions are left to the design engineer.

Accurate field measurements of actual performance will improve understanding of off-design performance. Manufacturers may in the future provide more off-design information and provide greater focus on average operating conditions. In some situations, the manufacturer may only need to be asked to provide performance data at multiple conditions, which could lead to significant energy savings by optimizing efficiency at average conditions instead of peak conditions.

Part-Load Conditions

By definition, part-load conditions are of concern for annual energy analysis. Understanding of part-load performance is essential to selecting and designing energy-efficient refrigeration systems. Selection of compressors is an example: load varies continuously through the year and will presumably never be at 100%, assuming there is a true safety factor. Hence, mod-

Condenser Derating Example

As an example of how derating factors should be considered for modeling the performance of a particular type of equipment, the following derating comparison table considers the potential estimated adjustment factors for evaporative and air-cooled condensers in a grocery store. While none of the table values are available from manufacturers' catalogs or design guides, experience in measuring actual condenser operation shows that operating temperature differences (TDs) are often much higher than design TDs, particularly when adjusting for the actual part-load compressor operation. The first reaction is that there is something wrong with the condenser, when in fact there are numerous reasons for the difference between catalog values and real-world average operation. The derating comparison table outlines some of the reasons performance during a given hour in the year may vary from the idealized catalog rating for a condenser as well as how derating factors may vary between these two types of condensers. The value of considering individual factors is that some of these are minimized through improved design, control, and maintenance practices.

The table values are only estimates, intended to provoke thought and demonstrate the value of hourly analysis. To be effective, the analysis requires estimating operation during average hours rather than simply designing for peak conditions with catalog ratings.

Derating Comparison for the Purpose of Hourly Modeling

Catalog Capacity	Evaporative 100%	Air-Cooled 100%	Comments on Derating Factors
Applied vs. Catalog Adjustment	10%	10%	Applied conditions are likely less optimal than the assumptions made for catalog ratings, although since there are no rating standards referenced, all application points seem to have equal accuracy. Could be considered a commercialization factor.
Scale, Fouling, and Dirt Effects	20%	10%	Evaporator fouling is higher on average due to challenges with water treatment and the large effect of even a small amount of scale.
Non-Steady-State Effects	5%	5%	Can be much greater with fan cycling (causing unsteady, even reverse flow, particularly in evaporative condensers). Variable speed helps minimize this impact.
Field Installation Effects	10%	10%	Factors caused by location, air recirculation, etc. Evaporative condensers commonly located inside compressor room, with resultant heat gain and pressure drop. Evaporative condensers that suffer from backflooding (causing undesirable subcooling) during cool weather could realize a factor here several times higher.
Part-Load Effects	0%	0%	Aside from variable-speed fan principles, some part-load factors may reduce performance and some may increase performance. Heat exchange performance at part load is not published.
Net Capacity vs. Catalog	62%	69%	
Average Derating vs. Catalog	38%	31%	This considers all factors that may come into play to result in the effective capacity available *at a given hour*. This is only an example for understanding, not a recommendation.

eling of different compressor choices involves defining capacity and power at part load. Condensers are another example. At design conditions, the condenser is at full speed and full power. Only at part load does the value of variable speed versus fan cycling become apparent. Mechanical subcooling is an example where stable and accurate control is easy at design conditions but far more challenging at reduced load. After considering performance of components at part load, the next step is understanding how they interact as a system at part load, and this understanding drives the control strategies that yield the greatest energy savings.

Transient Conditions

The historical focus on selecting refrigeration equipment is not based on real-world operation but rather is based primarily on meeting peak conditions when the system is running at 100% capacity at design conditions, fully balanced, and at steady-state conditions. In reality, refrigeration system loads and operation are highly variable. Temperature is controlled second by second, and systems are in constant motion. In most systems pressures vary continuously, fans cycle, and compressors cycle and load or unload. This is a challenge to understand, to include in energy analysis, and to evaluate for better design, but is an important area in reducing energy use.

Generally, approaching steady-state operation results in more efficient performance of all system components. This starts with compressor selections and controls. Continuously variable capacity control allows the highest suitable suction pressure to be maintained, whereas compressor cycling results in an average suction pressure often much lower than design conditions.

Fouling and Age

Equipment performance can be affected by age and dirt, scaling, and fouling of heat-exchange surfaces. Making explicit allowances for age and fouling is a useful when trying to reduce generalized safety factors and better understand equipment performance. Condensers and other heat rejection equipment are the primary focus. Designers need to decide what scenarios and equipment require a fouling and age allowance in the form of system or equipment derating based on the equipment condition.

AIR VERSUS EVAPORATIVE CONDENSING

Identifying the best condensing means is important but is far from obvious in many areas, and thus often justifies an hourly analysis incorporating the hourly cost of electricity and calculated water consumption and water costs. Air-cooled and evaporative condensers have similar energy usage results in most climates. Evaporative condensers perform better in many warm and humid climates, and air-cooled condensers perform better in many hot and dry areas. First cost, control strategies, energy usage, energy cost, water usage, water cost, and water treatment all need to be evaluated to identify the best means of condensing.

Figures 3-8 through 3-10 help explain the comparative differences between air-cooled and evaporative condensing in terms of how each condensing method responds to weather. Note that these are examples for just one location and weather varies by location.

Figure 3-8 compares wet-bulb temperature and dry-bulb temperature for a peak day and an average weather day in Dallas, Texas. The important point is the much greater difference between dry-bulb temperature and wet-bulb temperature on the peak day compared with the average day. When there is a large difference between dry-bulb and wet-bulb temperatures, evaporative condensing has a greater advantage, by allowing lower condensing temperatures. However, the difference may be quite small for much of the year in most areas.

Again using Dallas, Texas, Figure 3-9 illustrates an entire year of weather with dry-bulb temperature and wet-bulb temperature data points for each hour of the year (8760 hours). The temperatures are arranged from highest to lowest dry-bulb temperature, lined up with the range of wet-bulb temperature values that coincide with the dry-bulb temperature values. The differ-

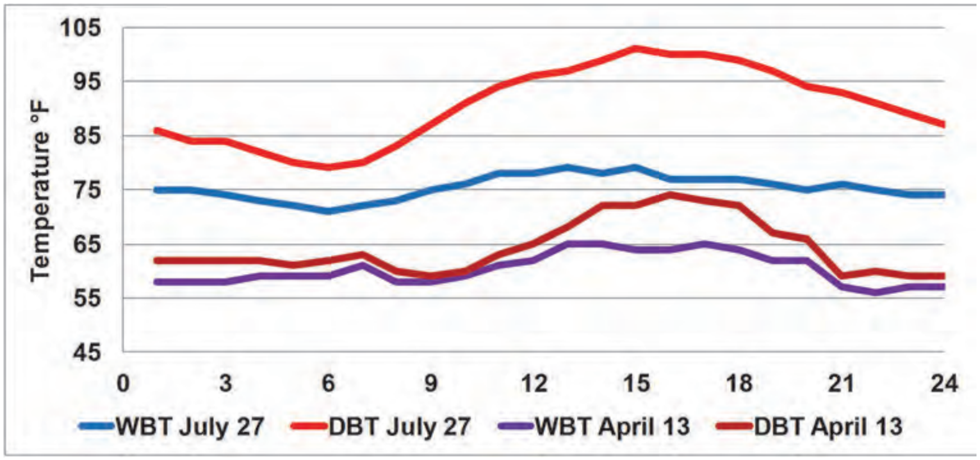

Figure 3-8 Peak Day and Typical Day Weather for Dallas, Texas

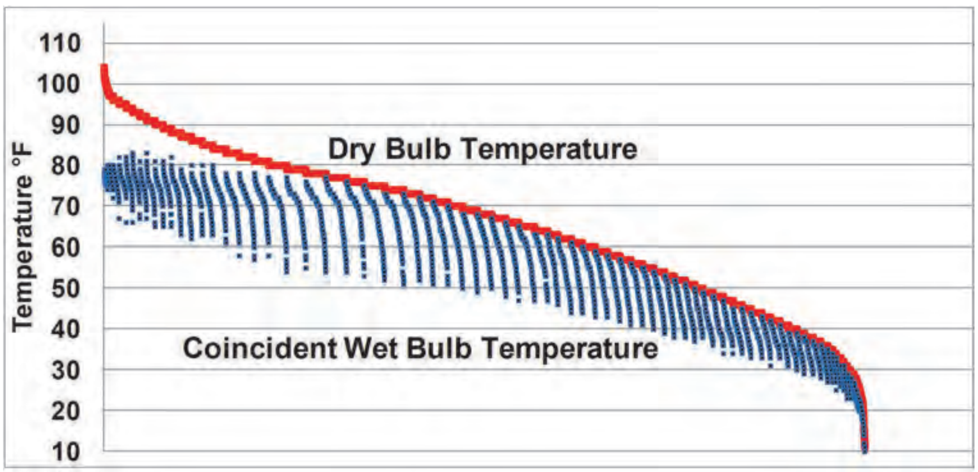

Figure 3-9 Full-Year Hourly Dry-Bulb Temperature from Maximum to Minimum with Coincident Wet-Bulb Temperature for Dallas, Texas

ence between dry-bulb and wet-bulb temperatures is greatest at the high temperatures and becomes quite small as dry-bulb temperature decreases.

The refrigeration system condensing temperature is limited by ambient temperature, based on the ambient dry-bulb or wet-bulb temperature combined with the condenser approach or TD. Air-cooled condensers are commonly selected at 10°F to 15°F TD, whereas evaporative condensers are selected at a higher TD, commonly in the range of 20°F to 25°F above wet-bulb temperature.

Once again looking at Dallas, Texas, Figure 3-10 combines the annual weather with typical condenser TDs to compare condensing temperatures. At high temperatures, the greater difference between dry-bulb and wet-bulb temperatures results in a lower condensing temperature with an evaporative condenser. However, after approximately one-third of the annual hours, the condensing temperatures cross and air-cooled condensing provides a lower condensing temperature. Whereas air-cooled condensing has greater benefit at lower condensing temperatures, the minimum condensing temperature setpoint is an important consideration because lower floating head pressure savings are greater with air-cooled condensing. Figure 3-10 is based on a constant TD for evaporative condensing. In fact, the TD increases as wet-bulb temperature decreases due to the properties of moist air. This can be seen in the application factors in any

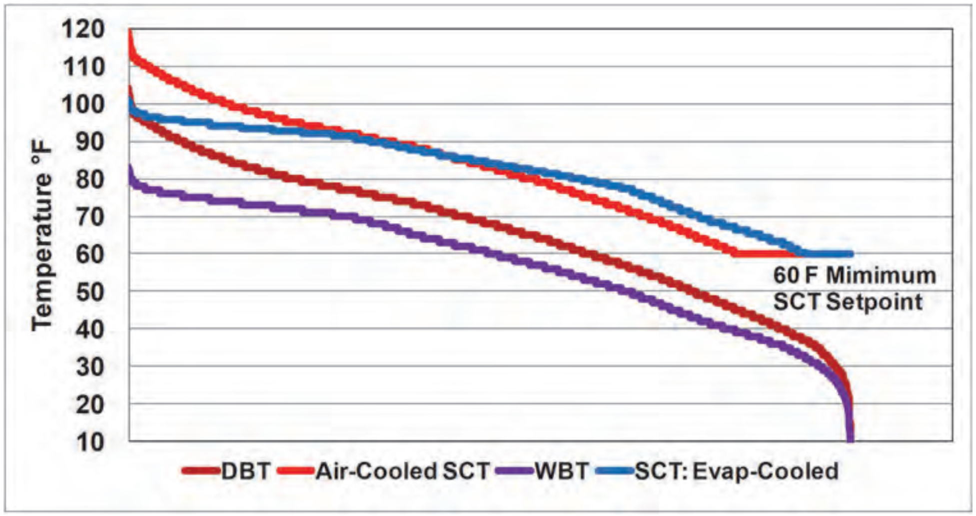

Figure 3-10 Ambient and SCT for Air-Cooled and Evaporative-Cooled Condensers

evaporative condenser catalog. For a given heat of rejection, the TD that the condenser can maintain gets larger at lower wet-bulb temperatures. Thus, in practice, the actual evaporative-cooled condenser saturated condensing temperature (SCT) would be higher and the crossover point in Figure 3-10 would move to the left.

Both types of condensers should employ floating head pressure to a minimum condensing temperature of 60°F or lower, use a variable setpoint control strategy, and control all fans in unison with variable-speed control. Floating head pressure with these setpoints has comparatively greater benefit for air-cooled condensing due to the different responses of air-cooled and evaporative condensers at lower ambient conditions.

Evaporative condensing can reduce first cost. In addition, as a result of the lower design condensing temperatures, evaporative-cooled systems have a lower range of peak condensing temperature to minimum condensing temperature, which reduces the difference between peak capacity and average capacity, generally improving the part-load performance of the entire system. Many utilities are increasingly focused on reducing peak summer demand and consider evaporative cooling as a key means of reducing grid stress.

Choosing a Condenser System

Many factors drive condenser choice, including physical constraints, noise restrictions, and water cost and availability. Rather than specify which type of condenser to select, this document points to how each type of condenser can be most effectively selected and controlled.

The choice of condenser should be approached with an open mind and with the use of appropriate analytical tools, engineering expertise, and operating experience. Hourly full-system simulation with weather and utility rates is very useful in evaluating performance and cost-effectiveness for all condenser options,

With efficient selection and control (e.g., reasonable fan power, variable-speed fan control, and floating head pressure), systems with air-cooled condensing may use less energy throughout the year, but evaporative condensing may reduce overall utility cost due to demand charges and higher energy costs during peak summer periods. The benefit of evaporative condensing is greatest in peak weather, when electric costs are typically higher (particularly for summer-peaking utilities).

It is important to note that condenser applications for grocery stores are different than those for industrial refrigeration; therefore, experience with evaporative condensers in industrial applications does not really apply for grocery stores. On the other hand, techniques can be learned from the industrial sector, particularly regarding piping practice, which is addressed in Chapter 5.

Hybrid Condensers and Precooling

Hybrid condensers incorporate benefits of both air-cooled and evaporative condensers. Hybrid condensers have finned condenser coils and use water for evaporative cooling during high ambient periods to cool the entering air, resulting in lower peak SCT and compressor power than air-cooled condensers. During the balance of the year, the condenser operates as a standard air-cooled condenser, avoiding water use and *potentially* offering lower head pressure as compared with an evaporative condenser.

Hybrid condensers may present an attractive option in certain climates and under certain utility rates with high summer demand and energy rates. To evaluate their cost-effectiveness, the hourly year-round performance of hybrid condensers must be considered. Compare the hybrid condenser dry-mode performance parameters (design approach, specific efficiency, and control methodology) with the recommendations for air-cooled condensers in Chapters 4 and 5. If the hybrid condenser performance equals the air-cooled recommendations, the evaporative-mode benefits are fully realized; otherwise, the peak period advantages must be weighed against the disadvantages during the balance of the year to understand the actual savings.

Another option is an evaporative precooling system, which can be added to standard air-cooled condensers and may be attractive under certain conditions. Considerations include the advantages of evaporative cooling during peak periods where electric costs are high and grid capacity is tight, the advantages of air-cooled condensers during the balance of the year in most areas, and whether water scarcity and cost are of concern. One concern with evaporative precooling systems is the effect of air-side pressure drop on fan power and/or condenser airflow. Control strategies must optimize saturation efficiency along with the other “balancing” objectives described in this chapter and in Chapter 5 to achieve cost-effective results.

Adiabatic Hybrid Condenser

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Table 3-3 Water Costs per 100 ft^3

	Dallas	Chicago	Denver	Miami	Salinas	Portland
Supply Water Cost, $/CCF	2.50	2.40	2.50	1.80	2.00	3.40
Sewer Cost, $/CCF	2.60	2.30	2.70	5.00	1.60	8.70
Sewer Fraction of Water Usage	40%	100%	40%	40%	40%	40%
Effective Cost, $CCF Water Usage	3.54	4.70	3.58	3.80	2.64	6.88

* CCF = hundred cubic feet
Source: Scott (2014)

Water and Water Treatment Costs

Water availability and water costs have become a significant concern in many areas of the country and must be considered in the comparison of air-cooled versus evaporative condensing. Table 3-3 shows water costs per hundred cubic feet (CCF) in several cities. With these water rates, the annual water costs may exceed the peak period electric savings that may result from use of evaporative condensing. Water usage can be estimated using annual simulation of condenser heat rejection and water-use factors provided by evaporative condenser manufacturers.

Water treatment for evaporative condensers is an additional cost, commonly relatively inexpensive, and done properly will reduce water usage. Historically, grocery stores have used low-cost water treatment, too often resulting in high bleed rates (excessive water usage), frequent instances of scaled condensers with high energy use, and shortened condenser life.

DIRECT VERSUS INDIRECT SYSTEM DESIGN AND EFFICIENCY

Direct refrigeration, with refrigerant piped directly to cases and walk-in cooling coils, is standard practice and the most common design for grocery store refrigeration systems. Development of indirect (also called *secondary*) cooling began in the early 1980s, using glycols and other low-temperature heat transfer fluids. Indirect systems can greatly reduce the halocarbon refrigerant charge, with attendant reduction in refrigerant cost and the greenhouse gas effect of refrigerant leakage. In terms of energy efficiency, indirect systems are inherently challenged by several factors, including the following:

- The heat exchange between the primary cooling system and the indirect cooling medium results in a lower compressor suction temperature and thus lower efficiency, although this can be minimized with carbon dioxide (CO_2).
- Indirect systems have parasitic loads not present in direct systems, resulting from pumping energy and often greater cooling load due to heat piping gains.

System applications differ, with indirect systems commonly having only one low-temperature and one medium-temperature fluid temperature, thus requiring all loads to be run at the lowest application temperature.

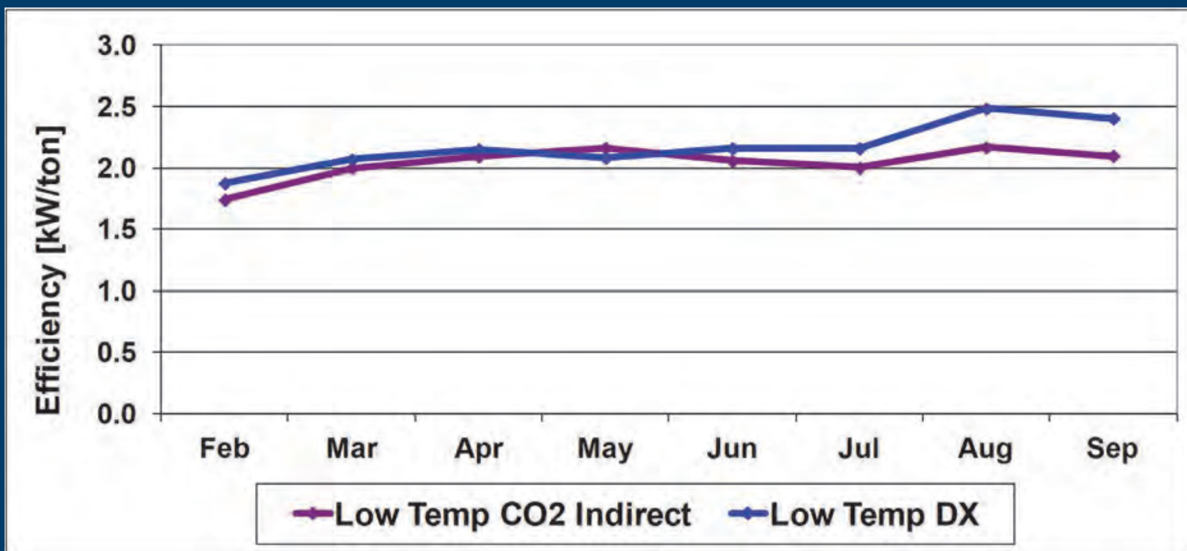
Food-grade propylene glycol has achieved a small market share for use in indirect systems and continues to be used for a relatively small fraction of stores for medium-temperature cases and walk-ins.

Glycol is a single-phase (liquid-only) heat exchange fluid with relatively low sensible-only heat capacity, requiring large flow rates. Indirect glycol systems have multiple inefficiencies, including:

- a lower suction temperature required by the glycol chiller heat exchanger,
- poor heat transfer properties,
- large circulation pumps, and
- large piping with attendant heat gain.

CO_2 Indirect System versus Standard Direct System

A field study compared a low-temperature CO_2 indirect system in a large big-box club store with a nearby reference store using a standard direct-expansion (DX) refrigerant system. The results provided in the following figure show the efficiency of the two systems, expressed in average kilowatt per ton for the month.



Low-Temperature CO_2 Indirect versus Direct Cooling

Normalized for differences in cooling load between the two stores, the results indicated that over the course of a year, the two systems were very similar in energy consumption. It was also observed that the compressor system saturated suction temperature for the two systems was close to the same, concluding that even though an additional heat exchanger is required, the CO_2 system benefits from improved evaporator coil performance and lower saturation temperature loss due to suction pressure drop.

Information used with permission of Walmart Stores, Inc.

Indirect/secondary systems are generally designed with all medium-temperature loads at one suction temperature, as required by the lowest-temperature fixture. As a result, the entire medium-temperature cooling load must be handled at this lower suction temperature, which in some instances can be a very substantial energy penalty. This combination of inefficiencies can result in a 20% energy penalty between an efficient-medium direct refrigerant system and a typical glycol system.

Within the last five years, CO_2 has been used as an indirect refrigerant, typically as a phase-change fluid, supplied as a liquid and returned as a vapor or liquid-vapor mixture. It has excellent heat exchange properties and operates at comparatively high pressures (versus halocarbons). These factors, coupled with the phase-change design, result in low pumping costs and higher suction temperatures for medium-temperature systems as compared with glycol. For low-temperature systems, the suction temperatures at the compressors can be approximately equal to those required for direct halocarbon systems. The inefficiencies that result from a reduced number of suction groups still exist, and care must be taken not to penalize the

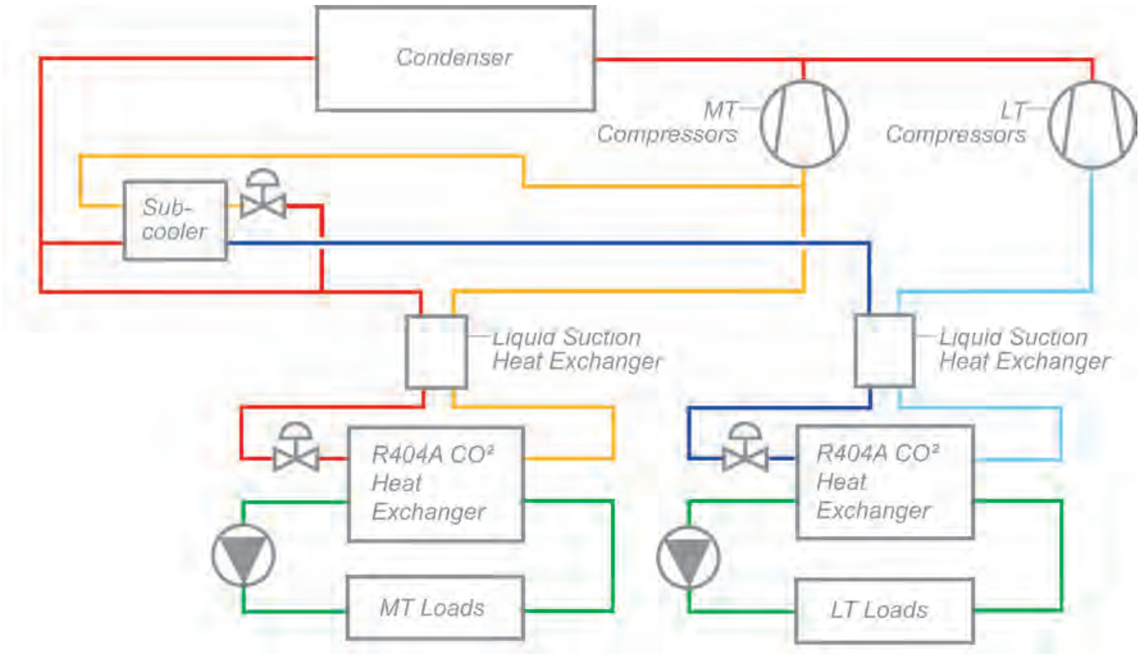


Figure 3-11 Indirect System Configuration

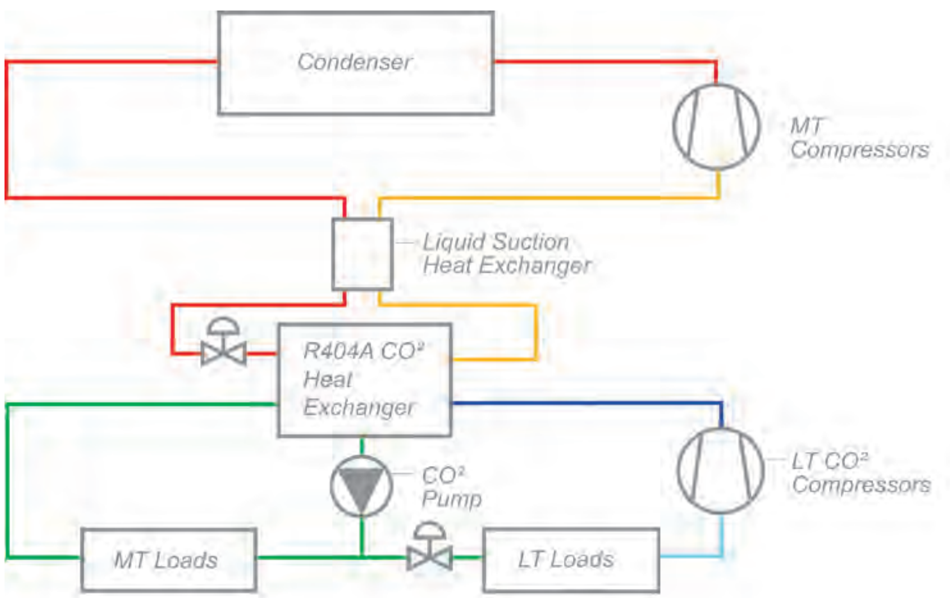


Figure 3-12 Cascade System Configuration

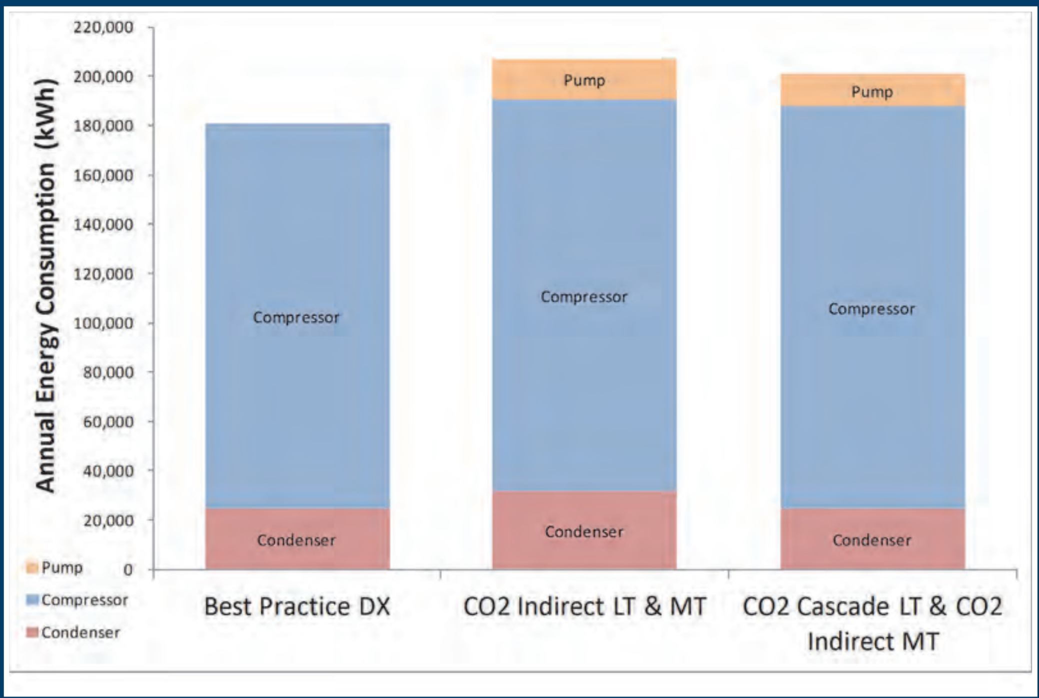
entire system for small loads that require a lower than average suction temperature. (See RF8 in Chapter 5 for more information on creating suction groups.)

Figure 3-11 shows a system configuration with separate low-temperature and medium-temperature indirect systems, which limits the R-404A refrigerant to the compressor system and condenser.

A cascade system is a different indirect method to utilize only CO_2 for the cases and walk-ins, with direct-expansion (DX) CO_2 compressors for the low-temperature cases and indirect pumped phase-change CO_2 cooling for medium-temperature cases and walk-ins. A cascade system is shown in Figure 3-12.

Annual Energy Usage Comparison of Cooling System Designs

An annual simulation study was conducted for a 15,000 ft^2 store designed with a cascade system. The study compared an R-404A/CO_2 cascade design with a CO_2 indirect cooling system and a baseline best-practice R-404A direct-expansion (DX) system. The baseline system used floating head pressure, floating suction pressure, variable-speed condenser control, subcooling, and other state-of-the-art features. As applicable, the cascade and indirect systems incorporated the same features, so all three systems were made as efficient as possible. The results of the study are as shown in the following figure and indicate the baseline system has the lowest energy and also shows the importance of pumping energy, particularly in a small system.



Comparison of DX, CO_2 Indirect, and CO_2 Cascade Annual Energy

DISTRIBUTED VERSUS CENTRALIZED RACK DESIGNS

The primary difference between a distributed rack and a centralized rack is where it is installed. The system components are essentially the same, with some rearrangement to fit into a smaller package in the distributed rack system.

Distributed systems have primarily been used to reduce refrigerant charge since the racks can be distributed throughout the store and placed in closer proximity to the load, thereby reducing distribution piping length and consequently refrigerant charge. Reducing the amount of distribution piping will also reduce the total amount of heat gain into the system through these pipes, which is a good thing from an energy standpoint. Similar energy savings can be realized in a centralized rack design through the use of loop piping (also called *trunk piping*).

Suction grouping is another dynamic that can be affected in the use of distributed systems. To realize the full benefit of charge reduction in a distributed system design, only local loads should be piped to the corresponding system. However, further opportunity for energy savings can be lost if compressor lift is increased to accommodate all local loads with a single distributed system.

ELECTRIC VERSUS HOT-GAS DEFROST

Electric and hot-gas defrost are both used frequently for many different reasons. Both have pros and cons and are sometimes simply a user preference. As shown in Chapters 4 and 5, electric defrost is recommended. While an energy-efficient hot-gas defrost system is possible, when the refrigeration system is evaluated holistically, the disadvantages of the hot-gas defrost often do not justify the energy savings benefits.

Disadvantages of hot-gas systems include higher refrigerant leak potential and indirect energy penalties. The higher refrigerant leak potential is due to the impact of temperature swing and the resulting expansion and contraction of the distribution piping.

Although energy is directly consumed in the use of electric defrost heating elements, there are indirect electricity penalties with hot-gas systems. For example, in order to induce a reverse flow of hot gas through the suction and liquid lines, the head pressure of the system must be increased during defrost cycles so that the liquid header pressure is lower than the hot-gas header pressure. Compressor lift is consequently increased, which results in a compressor energy penalty. Flowing hot gas in reverse through the suction line also heats up this piping, which has to recover after terminating defrost, resulting in temporarily high return gas temperatures. Although hot-gas defrost is possible to use with loop piping, the vast majority of users employ it with single-circuited conventional piping. However, there are energy savings opportunities in reduced return gas temperatures by moving away from conventional piping to loop piping.

SPECIFIC DESIGN PHILOSOPHIES

MASS-FLOW-BASED DESIGN

Traditionally, compressor manufacturers published compressor ratings showing cooling capacity at a given saturated suction temperature (SST) and saturated discharge temperature, presupposing numerous system design parameters. Mass-flow-based design is a more accurate means of understanding and sizing refrigeration systems, resulting in a better grasp of actual cooling performance, a better balance between system components, and a more optimized design that achieves expected performance at a lower cost. Mass-flow-based design requires thinking about cooling in two successive steps. Rather than thinking about the cooling capacity of a compressor, mass-flow-based design involves separately considering the *cooling capacity of the refrigerant* and the *refrigerant pumping capacity of the compressors*.

In the past, the basis for semi-hermetic refrigeration compressor ratings was established with a return gas temperature (the temperature *entering* the compressor) of 65°F for all application conditions. All U.S. manufacturers use AHRI Standard 540 (AHRI 2004), which follows the same basis.

This is very different from current refrigeration applications For example, the catalog cooling capacity rating a low-temperature compressor operating at –25°F SST assumes the refrigerant will enter the compressor at 65°F with a superheat of 90°F, all of which is considered productive refrigeration in the catalog rating. In fact, the temperature leaving the refrigerated case or walk-in is probably closer to –10°F, so the superheat that is realized as productive cooling is only 15°F, with the balance being nonproductive heat gain.

Figure 3-13 illustrates this capacity difference for refrigerant R-404A at 80°F SCT and –25°F SST using a pressure-enthalpy properties chart.

- At 10°F productive superheat the Δh = 50.9 Btu/lb—From point 4 to point 1a, leaving the case or walk-in
- At 65°F return gas temperature Δh = 66.9 Btu/lb—From point 5 to point 7, entering the compressor

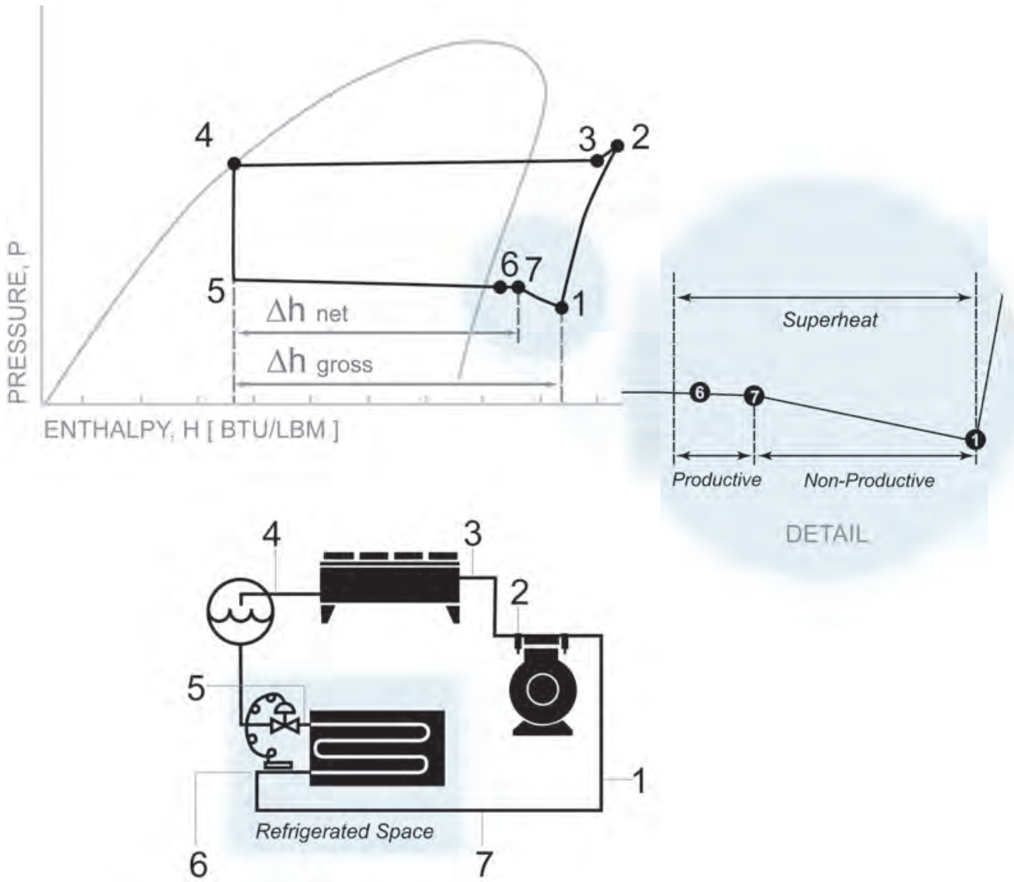


Figure 3-13 Productive and Nonproductive Suction Superheat

For a compressor actually operating with 65°F return gas temperature, there is a 31% error between catalog rating and actual realized cooling. The error increases with higher SCTs and lower SSTs.

Using refrigerant properties with mass-flow-based design allows accurate calculation of the required mass flow to meet design cooling loads. With this method, mass flow is determined by the difference in heat content between the leaving superheated gas temperature and the entering liquid temperature, with both determined at the boundary of the display case or walk-in box. For example, liquid-subcooling and liquid-suction heat exchangers can be explicitly incorporated, with the benefits reflected in lower required mass flow.

The compressor mass-flow pumping capacity can then be determined separately, based on the conditions at the compressor. The actual return gas temperature is normally lower than 65°F, but particularly on low-temperature systems. On most systems there is still a large amount of nonproductive suction superheat between the load and the compressor due to long suction lines.

Compressor selection based on required and actual mass flow allows compressor capacity and desired safety factor to be more exactly specified, known, and designed and also results in greater design accuracy and cost-effectiveness. Compressor manufacturers' software programs provide mass flow and the ability to adjust return gas temperature. However, this adjustment is based on a simple density adjustment and is not part of the testing and rating standards. Since most refrigeration compressors use suction gas to cool the motor and/or have significant internal heating of the return gas before it is compressed, the density adjustment tends to overstate the actual mass flow at lower return gas temperatures. This is an opportunity for future improvement in refrigeration compressor rating standards.

USE OF DATA FROM NEW AND EXISTING STORES

Operating information from existing and newly built stores provides a rich and largely untapped opportunity to improve design of grocery store systems and control systems and achieve lower energy use and more cost-effective system designs. Most refrigeration controls and energy management systems (EMSs) continually record pressures, temperatures, and equipment operating information. This data can be evaluated to compare actual operation to expected operation in terms of capacity, system efficiency, and component performance. With careful engineering processes, actual hourly operation can be correlated with hourly simulation values, with examination of the high and low weather extremes, as well as average operating conditions. This effort often initially yields surprising information about how systems actually operate, leading to successive rounds of improvements and improved understanding regarding balance between system components.

Existing store information is particularly useful when multiple locations can be compared, thus offering a unique opportunity for chain stores. Similar store designs can be compared over varying climates, and varying designs can be evaluated with a degree of statistical validity. Historically, grocery chains have often been fast to try new technologies, since testing a new concept on one or several stores can be a relatively easy decision. However, devoting the time and budget to design test methods, evaluate performance, and draw firm conclusions is far less common.

To a far greater extent than the current grocery store industry practice, existing system performance and in particular new technologies can and should be evaluated to ensure performance meets expectations.

DETERMINING DESIGN EVAPORATOR AND DISCHARGE AIR TEMPERATURE

Determining the required saturated evaporator temperatures (SETs) and associated discharge air temperatures for cases to adequately maintain product temperatures and acceptable shelf life is something that has typically been left up to equipment manufacturers. However, end users with diverse merchandising and operational demands may require holding temperatures to be flexible and to deviate from the manufacturer's provided temperatures. There is also an energy impact to the system if the product is kept colder than necessary.

With the use of technologies such as electronic valves and tighter controls, it is possible to optimize the system's low-side temperature parameters. One way to do this is to use data from existing stores to analyze and monitor actual operational parameters and the resulting performance. Defining the required temperature parameters to satisfy a specific application is valuable knowledge that can be leveraged in commissioning and recommissioning systems moving forward. Reevaluation may be needed periodically as merchandising needs change or as systems and equipment manufacturers evolve. Close collaboration with equipment manufacturers can also prove beneficial from a feedback and equipment development standpoint to improve future iterations of specific case models and their published data.

A product emulation temperature sensor (or similar technology) can also provide valuable feedback on the effect evaporator and air temperatures have on the product. This concept can be taken one step further to control the electronic expansion valve (EEV) or electronic evaporator pressure regulator (EEPR) based on this supposed product temperature. One caution is to understand that certain system issues can easily go unnoticed if evaporator temperatures are allowed to decrease over time to maintain target product temperatures, thereby sacrificing energy savings. Issues such as a dirty or iced coil or a bad evaporator fan can be compensated for by a lower evaporator temperature without setting off any product temperature alarms.

CONTROL SYSTEMS

Central Proprietary versus Distributed Control Architectures

Grocery store control systems evolved uniquely to serve the needs of refrigeration, specifically temperature and defrost control, compressor sequencing, and the control of the various related equipment and loads. These specialized, proprietary control systems are generally designed with one controller per parallel compressor rack. The limitations of single-vendor proprietary, centralized systems are increasingly evident:

- Limiting use of best-in-class controls
- Limiting flexibility in hardware vendors
- Limiting innovation

In contrast, distributed controls have the potential to allow more intelligence at a subsystem level and more flexibility in overall system design. Whether grocery store controls will follow the HVAC example of standards-based, interoperable, multivendor controls (e.g., LonWorks™ [LMI 2015] and BACnet® [ASHRAE 2015]) is yet to be determined. Definitely, improvements in control systems and improved facility integration are essential to achieving and sustaining 50% energy reduction, and chains should be considering more advanced control system designs, including support of standards-based control networks.

CHANGING TECHNOLOGIES AND PARADIGM SHIFTS

IMPLICATIONS FOR FUTURE IMPROVEMENTS

Probably more than ever before, grocery store refrigeration and HVAC systems are changing, driven by

- consideration of natural or lower-GWP refrigerants,
- the need to reduce energy use,
- globalization,
- the issuing of building and appliance efficiency standards for the first time, and
- the desire of large companies to standardize equipment and system design.

There are numerous needs and opportunities for future development by industry and other organizations, including the following:

- Improved refrigeration compressor rating standards and test methods, addressing variable return gas conditions
- Development of standards and protocols to allow interoperability of distributed refrigeration controllers, similar to that enjoyed by HVAC equipment, allowing building-wide systems integration
- Adoption of test and rating standards (with improvements where necessary for today's equipment and conditions) for commercial refrigeration condensers and evaporator coils, followed by certification by manufacturers
- Advancement of modeling tools, toolkits, methods, and education to allow effective modeling of refrigeration systems and to support improved design and efficiency, as well as a means to comply with emerging building standards affecting grocery stores

ELECTRONIC EXPANSION VALVES AND SUPERHEAT OPTMIZATION

The primary purpose of a traditional mechanical thermostatic expansion valve (TXV) is to meter refrigerant into the evaporator at a rate sufficient to satisfy the load on the coil (by maintaining a level of superheat). This function is critical to maintain product temperatures and to keep liquid from returning to the compressor. Separately, the purpose of an evaporator pressure regulator (EPR) is to adjust the evaporating pressure in order to maintain a specified evaporating temperature, also critical in maintaining product temperatures. These two valves have always operated independently of each other.

With the advent of EEVs and smart controllers, it's possible to remove EPRs in some instances and use EEVs to serve both functions. Because controlling superheat and evaporating pressure both ultimately serve the same purpose of maintaining a discharge or return air temperature, we can now look to simply control EEVs to maintain air temperatures directly. Instead of using an EPR to increase evaporator temperatures, the EEV can drive up refrigerant superheat where warmer temperatures are required. The energy benefit of this control philosophy results from an increased refrigerant enthalpy change where superheats are allowed to increase, which ultimately reduces mass flow and, consequently, compressor demand. Conversely, in temperature-critical coils (see RF8 in Chapter 5), EEVs should reduce superheats as much as possible in order to reduce TDs and allow the suction pressure to float as much as possible. Where EEV valve positions indicate a very high superheat due to a reduced load, walk-in box evaporator fans can also be reduced to 80% speed when the load drops below 50%. For display cases where suction temperatures are grouped appropriately and tightly (as recommended in this Guide), this type of control should not result in extremely high superheats. However, there may be instances where a large temperature difference between SST and SET exists for which EEVs are not effective in maintaining discharge air temperatures. In these situations, an EPR may still be required.

One of the main reasons EEVs are becoming more commonly used is their wider range of capacity due to greater stroke with respect to mechanical TXVs. Because of this flexibility, EEVs can maintain their rated capacities under a wider range of differential pressures across the valve. This in turn allows the system head pressure to float to even lower pressures. Additionally, with the added benefit of more precise control, the entire system can operate more efficiently due to more steady operation.

REFRIGERANT HEAT RECOVERY

At one time, recovery and use of heat from refrigeration systems to provide space heating in grocery stores was used extensively and provided all or most of the heat in most stores across the US. However, refrigerant heat recovery has become less common in recent decades, largely to reduce refrigerant change and leakage. To reduce energy usage and operating costs and to meet sustainability objectives, many grocery chains are again considering refrigerant heat recovery.

Heat rejected from refrigeration systems is the sum of the refrigeration cooling load and the electric energy used by the compressors, which all becomes heat. Because display fixtures and walk-ins are continuously removing heat from the store area, there is compelling logic to simply return that same energy to the store. The refrigeration heat of rejection is low-grade heat and there is often a trade-off between obtaining the heating savings and the energy impacts on the refrigeration system, particularly in consideration of the desire to float head pressure and take advantage of low ambient temperatures.

A portion of the heat of rejection is available as superheat, with temperatures ranging from 120°F–150°F down to condensing temperature; however, this is only 15% to 20% of the available heat. To obtain more substantial quantities of heat and justify the system costs, the system must be designed for condensing in addition to desuperheating.

California Title 24 Heat Recovery Requirements

California's 2013 *Building Energy Efficiency Standards for Residential and Nonresidential Buildings* (Title 24) include mandatory requirements for commercial refrigeration and refrigeration heat recovery. Section 120.6(b) 4 requires that grocery stores use at least 25% of the heat from refrigeration for space heating in new stores (CEC 2013).

INTERACTION BETWEEN SYSTEMS

In all buildings, a number of systems are in operation at any given time. There are certain individual systems that can operate isolated from other system operations; however, many systems work together, are interlocked, and serve a purpose that is necessary for other systems to operate properly. Grocery stores exhibit this dynamic even more than the average building type.

The commercial refrigeration systems of a grocery store interact with the building HVAC systems in a number of ways. The simplest interaction is that the open cases provide uncontrolled cooling to the sales space environment, sometimes requiring heating of the space to maintain occupant comfort. Addition of doors to the refrigerated cases minimizes this effect, simplifying temperature control, increasing efficiency, and reducing the total load on the refrigeration system.

A second interaction is that the open cases are subject to condensation and frosting because the dew-point temperature of the occupied space is higher than the temperature of the exposed refrigeration heat transfer surfaces in the case. Because the temperature difference between condensing and evaporation is much higher for the refrigeration system than for the space cooling system, the coefficient of performance (COP) is lower for the refrigeration system. As a result, it is more energy efficient to provide as much dehumidification as possible with the space cooling system to minimize condensation and additional cooling in the refrigeration system. Condensation and frosting in the freezer cases furthermore initiates energy-intensive defrost cycling for the refrigeration system evaporator coils. Maintenance of a low-dew-point temperature in the space by maintaining a low apparatus dew point for the outdoor air ventilation system is the primary strategy for minimizing condensation in refrigerated cases.

Control of infiltration of humid outdoor air is another strategy for minimizing condensing in the refrigerated cases. Standard infiltration controls such as vestibules for entry doors and infiltration control strategies for the loading dock are very important for this occupancy type, especially in humid climate regions. Control of infiltration is made more challenging by the addition of cooking functions and required exhaust hoods to the building. Inadequate makeup air for kitchen hoods can result in negative pressure within the building, increasing infiltration of potentially humid air. To the extent that additional makeup air is required beyond the outdoor air ventilation flow required by ASHRAE Standard 62.1 (ASHRAE 2013c), an additional makeup air system may be required for the kitchen exhaust. Makeup air delivered to the immediate vicinity of the exhaust hood can be conditioned to a much wider range of conditions than the ventilation air delivered to the sales area of the store. Because this air is never in contact with refrigerated cases, it can be delivered at a much higher dew point than is required for the sales ventilation air. It can also be delivered at a lower temperature than may be allowable in the sales area due to the elevated mean radiant temperature in the kitchen area.

KITCHEN EQUIPMENT AND HVAC

For stores with full kitchen operation, a separate dedicated outdoor air system (DOAS) supplying required additional makeup air for the kitchen exhaust hoods may be required. The limitations on supply air conditions for the DOAS serving the sales area impose a significant energy penalty for providing additional air through that system to meet makeup air requirements for kitchen hoods. Should the relief air from the sales air be insufficient to meet the makeup requirements, an additional DOAS should be used. Only in extremely dry climates should the capacity of the sales area DOAS be increased to meet the additional kitchen hood makeup requirement. Air from the kitchen DOAS should be delivered in the vicinity of the exhaust hoods but in a configuration that does not negatively affect the capture of the hoods. Since this air will never be in contact with refrigerated cases, the most energy-efficient strategy for this system is control of the supply air temperature at the widest range consistent with comfort in the kitchen area. Locking out conditioning of the outdoor airstream in the DOAS when the outdoor temperature is between 50°F and 70°F should provide both comfort and energy efficiency. Exhaust air heat recovery can be implemented for this DOAS in cold climates. (See HV13 in Chapter 5.)

If ventilation air from the DOAS is delivered to the space separately from the conditioning air system, the ventilation air ductwork system should be designed to ensure that adequate ventilation is provided over the entire area served by the ventilation system.

Makeup air for the kitchen exhaust should be adequate to prevent the depressurization of the store and the resulting uncontrolled infiltration of potentially humid air into the store. Provision of required ventilation according to ASHRAE Standard 62.1-2013 (ASHRAE 2013c) likely will provide far more makeup air than is required for normal toilet and janitor closet exhaust. Kitchen exhaust in the store may require additional makeup air, but determination of additional makeup air requirements should recognize the contribution of this ventilation air and the balance between required ventilation, required functional exhaust, and kitchen exhaust. Provision of the required additional makeup air and utilization of transfer of ventilation air from the sales area should create a net airflow from the sales area into the kitchen area to minimize the migration of fumes from the kitchen into the sales area.

Cooling of the kitchen area and the areas immediately surrounding it should recognize the radiant heat gain from the hot kitchen equipment to the surrounding surfaces. Additional cooling capacity to offset these gains should be provided to areas within line of sight of the hot kitchen equipment.

REFRIGERATION AND HVAC

Due to large open refrigeration fixture operation within grocery stores, the ambient temperature at which heating is needed in a typical store is often 10°F or more above those of other commercial building types. Also, with open refrigeration equipment operating below the dew-point temperature within a store, much of the mechanical dehumidification conducted within a grocery store is a refrigeration system load rather than a load on the HVAC system. For these reasons, HVAC design in grocery stores has typically considered cooling and dehumidification load reductions (also called *case credits*) to account for refrigeration cooling and has delivered smaller air-conditioning systems as a result.

Ventilation

With occupancy ventilation requirements generally low, ventilation within grocery stores is usually driven primarily by exhaust requirements of kitchen exhaust hoods and a need to maintain positive pressure within the store.

IMPACT OF DOORS ON REFRIGERATED CASES

Case doors decrease heat and air infiltration into refrigerated cases by about 68% as compared with typical open designs. As a result of reduced infiltration, the refrigeration system's load is significantly reduced. When doors are applied to the majority of a grocery store's open cases, there's also an opportunity to significantly increase the suction pressure design parameter for several compressors. Beyond these direct energy benefits to the refrigeration system, several other dynamics within a grocery store are also affected by the addition of doors on open cases.

Thermal Comfort

Open cases deliver significant cooling to the sales floor of a grocery store, both reducing the need for HVAC and increasing the need for heating, depending on the time of year. The difference in local temperatures between aisles with open cases versus aisles with closed classes is pronounced. Closed cases provide significantly better local customer comfort.

Humidification

Open refrigerated cases provide significant dehumidification to a grocery sales area as they operate at temperatures below the in-store dew point. By adding glass doors to open cases, dehumidification requirements are shifted to the HVAC system.

Display Cases with Glass Doors

Open display cases provide uncontrolled cooling and dehumidification to the grocery sales floor. Conversely, display cases with glass doors can reduce refrigeration system load requirements while shifting sales floor temperature control and dehumidification to the HVAC system. The temperature difference between aisles with closed cases and aisles with open cases can be pronounced—clearly, aisles with closed display cases provide significantly better customer comfort.

The Whole Foods store in Oxnard, California, underwent a retrofit of their low-temperature coffin-style display cases in 2014. The store's frozen food section contained 40 ft of open coffin-style low-temperature display cases that were retrofit with glass lids and integrated light-emitting diode (LED) lighting.

The project was approved with a 2.6-year simple payback at $0.12/kWh power rates. The expected benefits include a 70% reduction in energy use, improved product lighting, and an improved shopping experience for customers. In the six months since the glass lids were installed, the store has experienced only positive feedback on the change from customers.

Low-Temperature (Freezer) Display Cases with Glass Lids

Photograph reproduced with permission of Mrs Gooch's Natural Foods Markets, dba Whole Foods Markets

Cases with Open Lids

View from Inside Case

Photographs reproduced with permission of Mrs Gooch's Natural Foods Markets, dba Whole Foods Markets

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Design Strategies and Recommendations by Climate Zone

4

INTRODUCTION

Users should determine the recommendations for their design and construction projects by first locating the correct climate zone. The U.S. Department of Energy (DOE) has identified eight climate zones for the United States, with each defined by county borders, as shown in Figure 4-1. This Guide uses these DOE climate zones in defining energy recommendations that vary by climate. The definitions for the climate zones are provided in Appendix B so that the information can be applied outside the United States.

This chapter contains a unique set of energy-efficient recommendations for each climate zone. The recommendation tables represent *a way* but *not the only way* to reach the 50% energy savings target over ASHRAE/IESNA Standard 90.1–2004 (ASHRAE 2004). Other approaches may save energy, and Chapters 2 and 3 provide information on integrated design, energy targets, benchmarking, and modeling for those who wish to generate other viable energy-efficient design options; confirmation of energy savings for those uniquely designed systems is left to the design team. Users should note that the recommendation tables do not include all of the components listed in Standard 90.1 because the Guide focuses only on the primary energy systems within a building.

When a recommendation is provided, the recommended value differs from the requirements in Standard 90.1–2004. When “No recommendation” is indicated, the user must meet the more stringent of either the applicable edition of Standard 90.1 or the local code requirements.

Each of the climate zone recommendation tables includes a set of common items arranged by building subsystem: envelope, daylighting/lighting, plug loads, kitchen equipment, refrigeration, service water heating (SWH), HVAC, and quality assurance (QA). Recommendations are included for each item, or subsystem, by component within that item or subsystem. For some subsystems, recommendations depend on the construction type. For example, insulation values are given for mass, steel-framed, and wood-framed wall types. For other subsystems, recommendations are given for each subsystem attribute. For example, vertical fenestration recommendations are given for thermal transmittance and solar heat gain coefficient (SHGC).

The fourth column in each table references the locations of how-to tips for implementing the recommended criteria. The tips are found in Chapter 5 under separate sections coded for envelope (EN), daylighting (DL), electric lighting (EL), plug loads (PL), kitchen equipment (KE), refrigeration (RF), service water heating systems and equipment (WH), HVAC systems and equipment (HV), and quality assurance (QA). In addition to representing good practice for design and maintenance suggestions, these how-to tips include cautions for what to avoid. Important QA considerations and recommendations are also given for the building design, construction, and post-occupancy phases. Note that each tip is tied to the applicable climate zone in Chapter 4. The final column of each climate zone recommendation table is provided as a simple checklist to identify the recommendations being used for a specific building design and construction.

CLIMATE ZONE RECOMMENDATIONS

The recommendations presented are minimum, maximum, or specific values (which are both the minimum and maximum values). Appendix A provides U-factors and F-factors for the prescriptive construction R-value options provided in the climate zone recommendation tables for opaque envelope measures.

Minimum values include the following:

- R-values
- Solar Reflectance Index (SRI)
- visible transmittance (VT)
- gas water heater or boiler efficiency
- thermal efficiency (E_t)
- energy factor (EF)
- energy efficiency ratio (EER)
- integrated energy efficiency ratio (IEER)
- integrated part-load value (IPLV)
- coefficient of performance (COP)
- energy recovery effectiveness
- fan or motor efficiency
- duct or pipe insulation thickness

Maximum values include the following:

- fenestration and door U-factors
- fenestration SHGC
- lighting power density (LPD)
- condenser fins per inch (FPI)
- condensing temperature
- fan input power per cubic feet per minute of supply airflow
- external static pressure
- duct friction rate

BONUS SAVINGS

Chapter 5 provides additional recommendations and strategies for energy savings and renewable energy that are over and above the 50% savings recommendations contained in the eight climate zone recommendation tables.

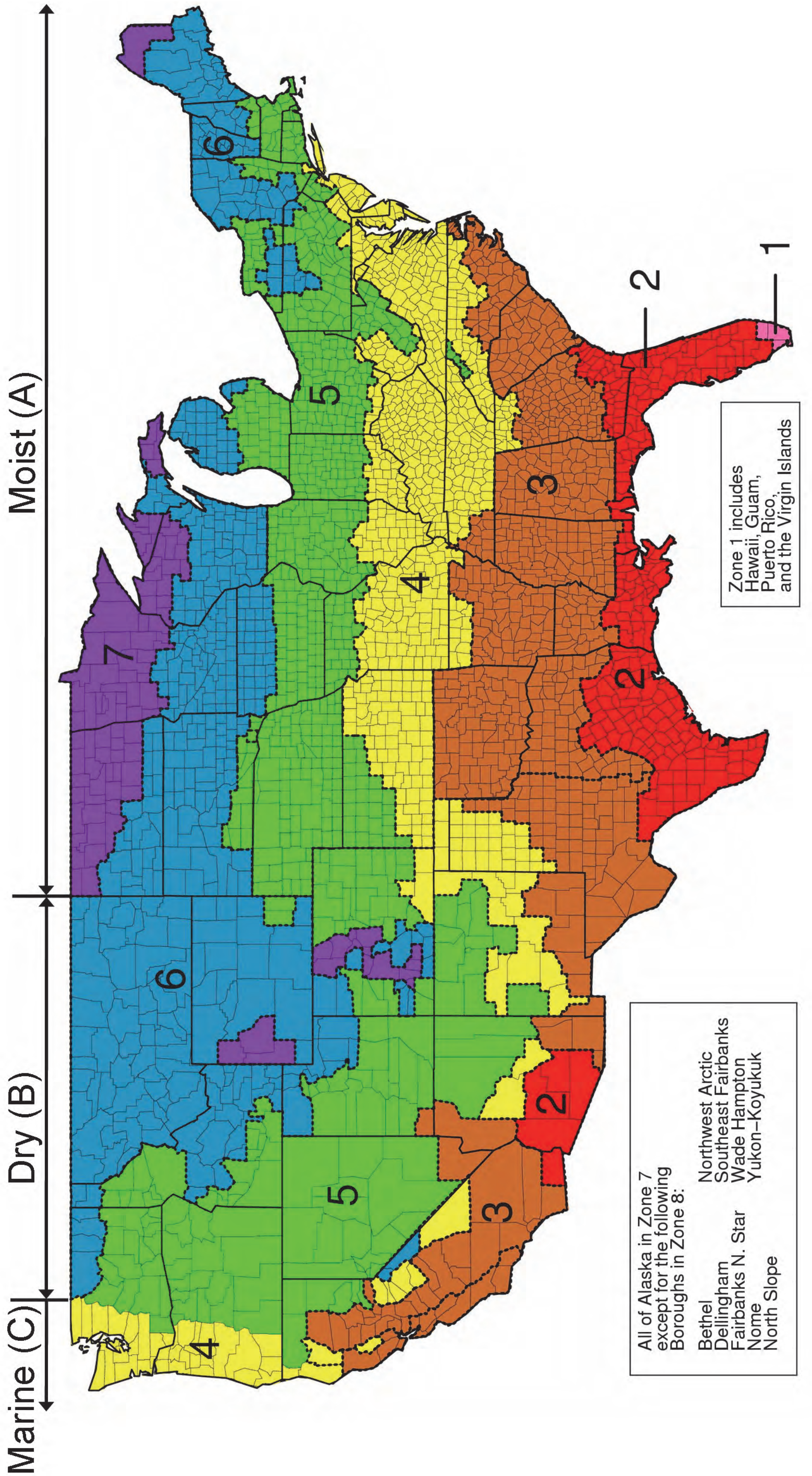
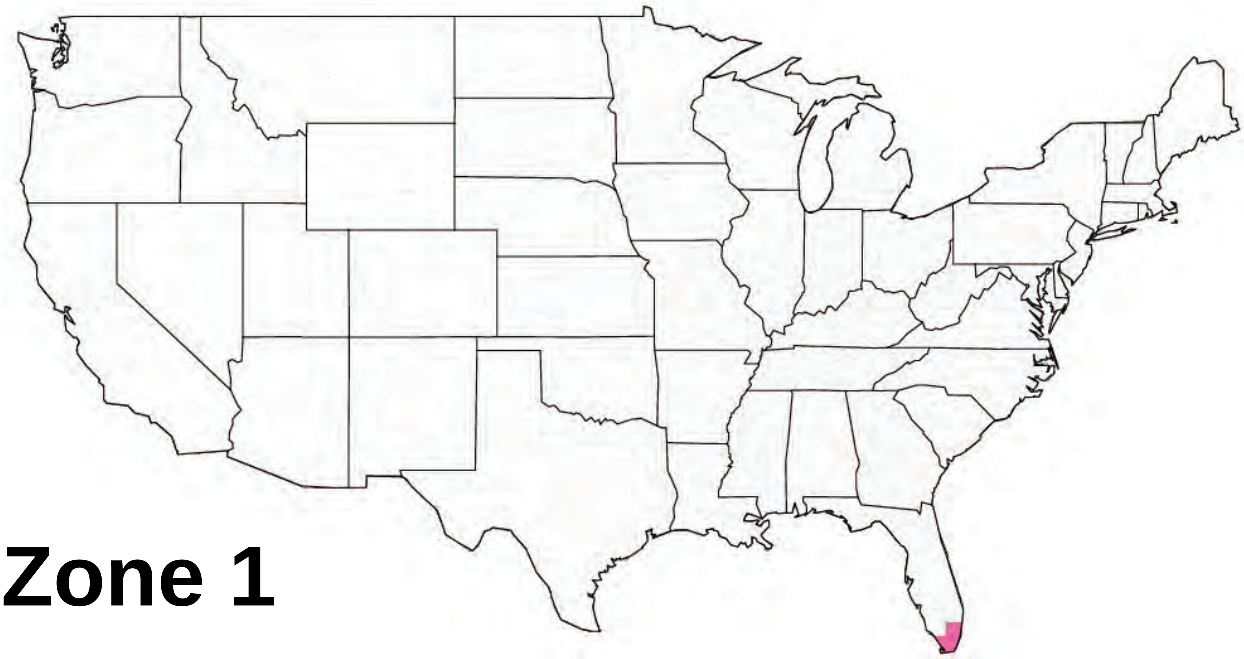


Figure 4-1 U.S. Climate Zone Map (Briggs et al. 2003)



Zone 1

Florida

Broward
Miami-Dade
Monroe

Guam

Hawaii

Puerto Rico

U.S. Virgin Islands

Climate Zone 1 Recommendation Table for Grocery Stores

	Item	Component	Recommendation	How-to Tips	✓
Envelope	Roofs	Insulation entirely above deck	R-20.0 c.i.	EN1, 3, 19–21	
		Metal building	R-19.0 + R-10.0 FC	EN2–3, 19–21	
		SRI	78	EN3	
	Walls	Mass (HC > 7 Btu/ft^2)	No recommendation*	EN4, 19–21	
		Metal building	R-0.0 + R-9.8 c.i.	EN5, 19–21	
		Steel framed	R-13.0	EN6, 19–21	
	Floors	Mass	No recommendation*	EN7, 19–21	
		Steel framed	No recommendation*	EN8, 19–21	
	Slabs	Unheated	No recommendation*	EN19–21	
		Heated	R-10 for 12 in.	EN10–11, 19–21	
		Freezer box floors	See Table 5-2	EN12	
	Doors	Swinging	U-0.70	EN13, 20	
		Nonswinging	U-0.16	EN14, 20	
		Vehicular/dock infiltration—door closed	0.28 cfm/ft^2 of door area	EN15, 20	
		Vehicular/dock infiltration—door open, truck in place	Weather seals for dock, levelers, trailer hinges	EN16, 20	
	Vestibules	All building entrances	Yes	EN17, 20	
	Air Infiltration Control	Continuous air barrier	Entire building envelope	EN18	
	View Fenestration—All Orientations	Thermal transmittance	U-0.57	EN22–25	
		SHGC	0.25	EN23–25	
		VT/SHGC	>1.10	EN29	
Daylighting / Lighting	Daylighting	Skylights or rooftop monitors	Required per Standard 90.1 when sales floor ceiling height > 15 ft and area is ≥ 2500 ft$_2$	DL1–10	
		Daylight area	≥ 50% of the sales floor	DL1–10	
		Skylight to daylight area	Minimum = 3%, maximum = 5%	DL3	
		VT	Skylight VT ≥ 0.40	DL3	
		Skylight SHGC	0.35	EN25–27	
		Skylight thermal transmittance	U-0.75	DL4	
		Effective aperture	Skylight effective aperture ≥ 1%	DL8	
		Glazing material/diffuser	Measured haze value > 90% when tested according to ASTM D1003 (ASTM 2013)	DL3	
		Controls	General lighting auto control in response to daylight	DL6–7, 9–10	
	Interior Lighting	Ambient and accent lighting	Sales floor LPD = 1.15 W/ft^2	EL1, 3–7, 9–11, 17, 25, 30	
			Stock room LPD = 0.6 W/ft^2	EL1, 4, 9–11, 17–19, 29–30	
			Average of all other LPDs = 0.8 W/ft^2	EL1–2, 4, 8–11, 17–19, 25–27, 29–30	
		Light source lamp efficacy (mean lumens per watt)	LED—general = 80, accent = 75, case = 50 LED—exterior = 75 T8 and T5—greater than 2 ft = 92, 2 ft and less = 85	EL12, 14	
		T8 ballasts	Nondimming = NEMA Premium instant start Dimming = NEMA Premium program start	EL12	
		All T5/T5HO ballasts	Electronic program start	EL12	
		All CFL and HID ballasts	Electronic	EL12	
		Lighting controls	Sales floor = time switch—auto to 25% during closed hours, auto to 2% or less when unoccupied	EL16	
			Additional specialty lighting = auto ON only during store open hours	EL16	
			Stock room, restrooms = auto ON/OFF occupancy sensors	EL15	
			All other = manual ON, auto OFF occupancy sensors	EL15	
			After hours = maximum 2% of total building LPD	EL16	
	Exterior Lighting	Façade and landscape lighting	LPD = 0.075 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = auto OFF at business closing	EL33–35	
		Parking lots and drives	LPD = 0.08 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL31, 34–35	
		Walkways, plazas, and special feature areas	LPD = 0.16 W/ft^2 in LZ3 and LZ4, 0.14 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL32, 34–35	
		All other exterior lighting	No recommendation*	EL34–35	
Plug Loads	Plug Loads	Equipment	Specify ENERGY STAR Use laptops, mini desktops, thin clients LED for security lighting and signage	PL1–6, 8–9	
		Controls	Motion-sensing outlet and power strips, network computer control, timer switches	PL1, 7–8	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 1 Recommendation Table for Grocery Stores *(Continued)*

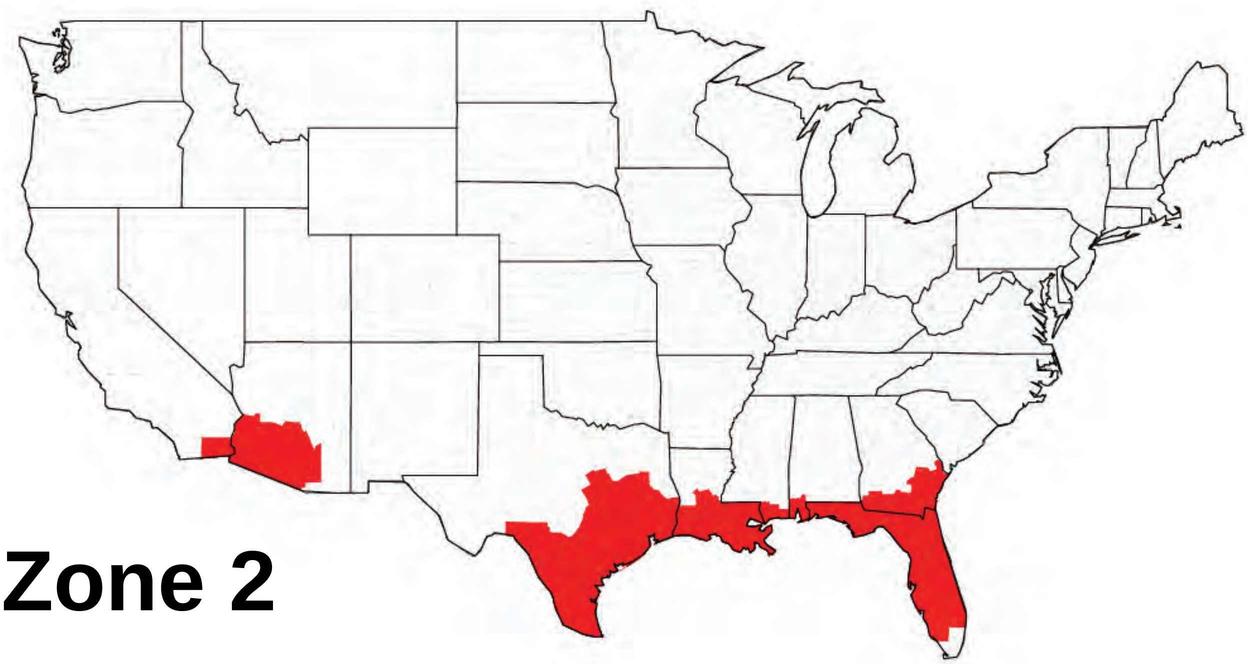
	Item	Component	Recommendation	How-to Tips	✓
Kitchen	Kitchen Equipment	Cooking equipment	ENERGY STAR or utility rebate-qualified equipment	KE1–2, 4, 6	
		Exhaust hoods	Side panels or end walls, larger overhangs, rear seal behind appliances, proximity hoods, DCKV	KE1, 3, 5, 6	
Refrigeration	Air-Cooled Condensers	Condenser sizing	Low temperature = 8°F TD Medium temperature = 12°F TD	RF1–2	
		Fin spacing	Maximum 10 FPI (excluding micro channel)	RF1	
		Specific efficiency	85 Btu/h·W at 10°F TD	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient dry-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Evaporative-Cooled Condensers	Condenser sizing	<68°F wb = 23°F TD 68°F to 75°F wb = 20°F TD >75°F wb = 16°F TD	RF1–2	
		Specific efficiency	180 Btu/h·W at 70°F wb and 100°F SCT	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient wet-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Compressor Systems	Suction groups/levels	Optimize suction groups based on load dynamics	RF8, 13–14	
		Indirect cooling design	CO_2 phase-change heat exchange Low temperature = two stage or cascade	RF9	
		Compressor group staging and capacity control	Continuously variable capacity control on each suction group	RF10	
		Suction group setpoint determination	Suction groups <25°F SCT = floating suction pressure logic	RF10	
		Mechanical subcooling	Low-temperature/medium-temperature SST ≥ 30°F	RF11	
	Display Cases and Walk-In Coolers/Freezers	Case type	Closed = dairy, beverage, deli, packaged salads, horizontal (tub), and low temperature Open = red meat, wet rack produce, specialty	RF15–16	
		Case door heater	Low temperature = 50–120 W/door with PWM controller Medium temperature = zero watt glass doors	RF17	
		Walk-in construction	Walk-in cooler insulation = R-25 Walk-in freezer insulation = R-40 Strip curtains on all doors	RF18, EN12	
		Walk-in doors	Doors < 48 in. = spring assist or cam-lift gravity hinge Doors > 48 in. = spring action door closer All doors = hydraulic door closer	RF19	
		Walk-in door switches and alarms	Freezer = fan and cooling OFF when door is open Cooler = fan and cooling OFF when door is open All = override and alarm integration	RF20	
		Walk-in box fan control	Two-speed fan control Reduce to 80% speed when load is below 50%	RF21	
		Lighting design	Open/closed cases ≤ 10 W/ft Meat cases ≤ 25 W/ft Display doors = motion sensors with 3 min time-out Walk-ins = vaportight LED with motion/vacancy sensor and/or door trigger switch	RF22, EL23, EL25	
		Defrost	Electric defrost = low-temperature cases, freezers, meat coolers; air defrost = all others Time initiation; temperature termination	RF23	
		Temperature and superheat control	Electronic modulating temperature control with floating suction pressure integration	RF24	
		Unit cooler	EC motor; select coils at 8°F TD on freezers Freezer fin spacing = 4 FPI Cooler fin spacing = no recommendation	RF25	
		Liquid-suction heat exchangers	High-efficiency heat exchange at piping exit; sized for additional suction superheat at design of 12°F for freezers and 6°F for coolers	RF26	
	Heat Recovery—HVAC	Heat recovery from refrigeration	Use at least 25% of the refrigeration design heat of rejection for space heating	RF27–31	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 1 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
SHW	Service Water Heating	Gas water heater efficiency	See Table 5-13	WH1–4	
		Electric water heater efficiency	See Table 5-14	WH1–4	
		Electric heat pump water heater	COP 3.0 (interior heat source)	WH1–4	
		Recirculation pump control	Time clock or demand control	WH4	
		Pipe insulation (d < 1.5 in. / $d \geq$ 1.5 in.)	See Table 5-15	WH5	
		Refrigerant heat recovery	Superheat recovery using low-temperature system	WH6, RF27, KE4	
HVAC	Distributed MA SZVAV DX Packaged RTU	Heating efficiency	Indirect gas heat = 80%	HV1–4, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4	
	Distributed MA SZVAV Chilled-Water RTU	Heating efficiency	High-efficiency condensing boiler = 90% Indirect gas heat = 80%	HV1–3, 5, 12	
		Cooling efficiency	Positive-displacement air-cooled chillers: <150 tons = 10.1 EER, 13.7 IPLV ≥150 tons = 10.1 EER, 14.0 IPLV	HV1–3, 5, 12	
		Fan motor efficiency	No recommendation*	HV1–3, 5, 24	
		Pump motor efficiency	No recommendation*	HV1–3, 5	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 5	
	Distributed SZVAV DX Packaged RTU with DOAS	Heating efficiency	Indirect gas heat = 80%	HV1–4, 6, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 6, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4, 6	
	Distributed SZCV Air-Source HP Packaged RTU with DOAS	Heating efficiency	See Table 5-17	HV1–3, 7, 12	
		Cooling efficiency	See Table 5-17	HV1–3, 7, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 7	
	SZVAV WSHP Packaged RTU with DOAS	Heating efficiency	See Table 5-18	HV1–3, 8, 12	
		Cooling efficiency	See Table 5-18	HV1–3, 8, 12	
		Pump motor efficiency	As per Standard 90.1-2013, Table 10.8-2	HV1–3, 8	
		Fluid cooler fans	Variable-speed control	HV1–3, 8, 9, 24	
		High-efficiency boiler	90% E_c	HV1–3, 8	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 8	
	DOAS	Heating efficiency	See Table 5-19	HV10	
		Dehumidification efficiency	See Table 5-20	HV10	
		Moisture removal efficiency	See Table 5-20	HV10	
		Fan and motor	65% mechanical/motor efficiency in absence of whole-unit EER rating Motor efficiency as per Standard 90.1-2013, Table 10.8	HV10, 24	
		Maximum external static pressure	0.7 in. w.c.	HV10	
	Ventilation/Exhaust	Cooling capacity—economizers	No economizer requirement	HV17	
		DCV/performance-based ventilation	Control ventilation air based on pollutant concentrations in space (CO_2, VOCs, etc.)	HV18	
		Exhaust air recovery—in-store cooking with > Standard 62.1 area exhaust requirement	A (humid) zones = 75% total effectiveness B (dry) zones = 75% sensible effectiveness Capture minimum 80% available exhaust air, including general and bathroom, for energy recovery	HV13	
		Exhaust airflow control	Control based on occupancy using time clock or occupancy sensor	HV20	
	Ducts and Dampers	Outdoor air damper	Motorized damper	HV13	
		Duct seal class	Seal Class A	HV21, 23	
		Insulation level	R-6	HV22	
QA	Quality Assurance	Commissioning	Commission throughout design and construction	QA1–13	
		M&V	Employ post-occupancy monitoring	QA14–17	

*Note: Where the table says “No recommendation,” the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.



Zone 2

Alabama

Baldwin
Mobile

Arizona

La Paz
Maricopa
Pima
Pinal
Yuma

California

Imperial

Florida

Alachua
Baker
Bay
Bradford
Brevard
Calhoun
Charlotte
Citrus
Clay
Collier
Columbia
DeSoto
Dixie
Duval
Escambia
Flagler
Franklin
Gadsden
Gilchrist
Glades
Gulf
Hamilton
Hardee
Hendry
Hernando
Highlands
Hillsborough
Holmes
Indian River
Jackson
Jefferson
Lafayette
Lake
Lee
Leon
Levy
Liberty
Madison
Manatee
Marion
Martin
Nassau
Okaloosa
Okeechobee
Orange
Osceola
Palm Beach
Pasco
Pinellas
Polk
Putnam
Santa Rosa
Sarasota
Seminole
St. Johns
St. Lucie
Sumter
Suwannee
Taylor
Union
Volusia
Wakulla
Walton
Washington

Georgia

Appling
Atkinson
Bacon
Baker
Berrien
Brantley
Brooks
Bryan
Camden
Charlton
Chatham
Clinch
Colquitt
Cook
Decatur
Echols
Effingham
Evans
Glynn
Grady
Jeff Davis
Lanier
Liberty
Long
Lowndes
McIntosh
Miller
Mitchell
Pierce
Seminole
Tattnall
Thomas
Toombs
Ware
Wayne

Louisiana

Acadia
Allen
Ascension
Assumption
Avoyelles
Beauregard
Calcasieu
Cameron
East Baton Rouge
East Feliciana
Evangeline
Iberia
Iberville
Jefferson
Jefferson Davis
Lafayette
Lafourche
Livingston
Orleans
Plaquemines
Pointe Coupee
Rapides
St. Bernard
St. Charles
St. Helena
St. James
St. John the Baptist
St. Landry
St. Martin
St. Mary
St. Tammany
Tangipahoa
Terrebonne
Vermilion
Washington
West Baton Rouge
West Feliciana

Mississippi

Hancock
Harrison
Jackson
Pearl River
Stone

Texas

Anderson
Angelina
Aransas
Atascosa
Austin
Bandera
Bastrop
Bee
Bell
Bexar
Bosque
Brazoria
Brazos
Brooks
Burleson
Caldwell
Calhoun
Cameron
Chambers
Cherokee
Colorado
Comal
Coryell
DeWitt
Dimmit
Duval
Edwards
Falls
Fayette
Fort Bend
Freestone
Frio
Galveston
Goliad
Gonzales
Grimes
Guadalupe
Hardin
Harris
Hays
Hidalgo
Hill
Houston
Jackson
Jasper
Jefferson
Jim Hogg
Jim Wells
Karnes
Kenedy
Kinney
Kleberg
La Salle
Lavaca
Lee
Leon
Liberty
Limestone
Live Oak
Madison
Matagorda
Maverick
McLennan
McMullen
Medina
Milam
Montgomery
Newton
Nueces
Orange
Polk
Real
Refugio
Robertson
San Jacinto
San Patricio
Starr
Travis
Trinity
Tyler
Uvalde
Val Verde
Victoria
Walker
Waller
Washington
Webb
Wharton
Willacy
Williamson
Wilson
Zapata
Zavala

Climate Zone 2 Recommendation Table for Grocery Stores

	Item	Component	Recommendation	How-to Tips	✓
Envelope	Roofs	Insulation entirely above deck	R-25.0 c.i.	EN1, 3, 19–21	
		Metal building	R-19.0 + R-10.0 FC	EN2–3, 19–21	
		SRI	78	EN3	
	Walls	Mass (HC > 7 Btu/ft^2)	R-5.7 c.i.	EN4, 19–21	
		Metal building	R-0.0 + R-9.8 c.i.	EN5, 19–21	
		Steel framed	R-13.0 + R-3.8 c.i.	EN6, 19–21	
	Floors	Mass	R-6.3 c.i.	EN7, 19–21	
		Steel framed	R-30.0	EN8, 19–21	
	Slabs	Unheated	No recommendation*	EN19–21	
		Heated	R-10 for 24 in.	EN10–11, 19–21	
		Freezer box floors	See Table 5-2	EN12	
	Doors	Swinging	U-0.70	EN13, 20	
		Nonswinging	U-0.16	En14, 20	
		Vehicular/dock infiltration—door closed	0.28 cfm/ft^2 of door area	EN15, 20	
		Vehicular/dock infiltration—door open, truck in place	Weather seals for dock, levelers, trailer hinges	EN16, 20	
	Vestibules	All building entrances	Yes	EN17, 20	
	Continuous Air Barrier	Continuous air barrier	Entire building envelope	EN18	
	View Fenestration—All Orientations	Thermal transmittance	U-0.57	EN22–25	
		SHGC	0.25	EN23–25	
		VT/SHGC	>1.10	EN29	
Daylighting / Lighting	Daylighting	Skylights or rooftop monitors	Required per Standard 90.1 when sales floor ceiling height > 15 ft and area is ≥ 2500 ft^2	DL1–10	
		Daylight area	≥ 50% of the sales floor	DL1–10	
		Skylight to daylight area	Minimum = 3%, maximum = 5%	DL3	
		VT	Skylight VT ≥ 0.40	DL3	
		Skylight SHGC	0.35	EN25–27	
		Skylight thermal transmittance	U-0.65	DL4	
		Effective aperture	Skylight effective aperture ≥ 1%	DL8	
		Glazing material/diffuser	Measured haze value > 90% when tested according to ASTM D1003 (ASTM 2013)	DL3	
		Controls	General lighting auto control in response to daylight	DL6–7, 9–10	
	Interior Lighting	Ambient and accent lighting	Sales floor LPD = 1.15 W/ft^2	EL1, 3–7, 9–11, 17, 25, 30	
			Stock room LPD = 0.6 W/ft^2	EL1, 4, 9–11, 17–19, 29–30	
			Average of all other LPDs = 0.8 W/ft^2	EL1–2, 4, 8–11, 17–19, 25–27, 29–30	
		Light source lamp efficacy (mean lumens per watt)	LED—general = 80, accent = 75, case = 50 LED—exterior = 75 T8 and T5—greater than 2 ft = 92, 2 ft and less = 85	EL12, 14	
		T8 ballasts	Nondimming = NEMA Premium instant start Dimming = NEMA Premium program start	EL12	
		All T5/T5HO ballasts	Electronic program start	EL12	
		All CFL and HID ballasts	Electronic	EL12	
		Lighting controls	Sales floor = time switch—auto to 25% during closed hours, auto to 2% or less when unoccupied	EL16	
			Additional specialty lighting = auto ON only during store open hours	EL16	
			Stock room, restrooms = auto ON/OFF occupancy sensors	EL15	
			All other = manual ON, auto OFF occupancy sensors	EL15	
			After hours = maximum 2% of total building LPD	EL16	
	Exterior Lighting	Façade and landscape lighting	LPD = 0.075 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = auto OFF at business closing	EL33–35	
		Parking lots and drives	LPD = 0.08 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL31, 34–35	
		Walkways, plazas, and special feature areas	LPD = 0.16 W/ft^2 in LZ3 and LZ4, 0.14 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL32, 34–35	
		All other exterior lighting	No recommendation*	EL34–35	
Plug Loads	Plug Loads	Equipment	Specify ENERGY STAR Use laptops, mini desktops, thin clients LED for security lighting and signage	PL1–6, 8–9	
		Controls	Motion-sensing outlet and power strips, network computer control, timer switches	PL1, 7–8	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 2 Recommendation Table for Grocery Stores *(Continued)*

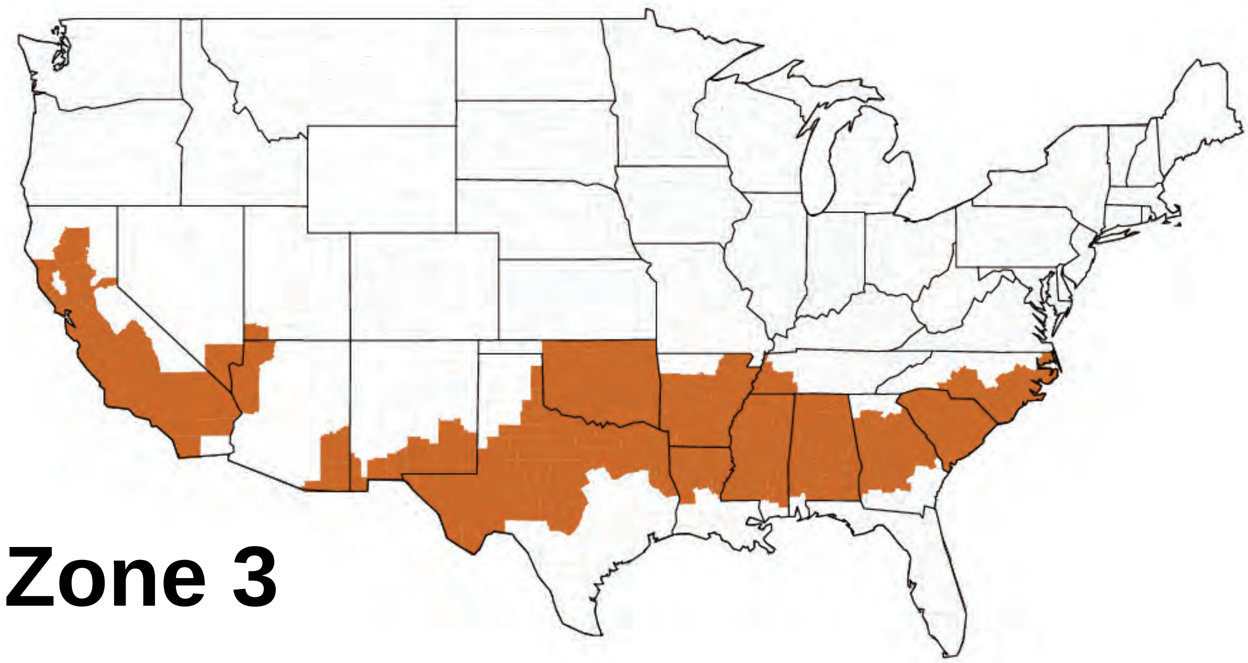
	Item	Component	Recommendation	How-to Tips	✓
Kitchen	Kitchen Equipment	Cooking equipment	ENERGY STAR or utility rebate-qualified equipment	KE1–2, 4, 6	
		Exhaust hoods	Side panels or end walls, larger overhangs, rear seal behind appliances, proximity hoods, DCKV	KE1, 3, 5, 6	
Refrigeration	Air-Cooled Condensers	Condenser sizing	Low temperature = 8°F TD Medium temperature = 12°F TD	RF1–2	
		Fin spacing	Maximum 10 FPI (excluding micro channel)	RF1	
		Specific efficiency	85 Btu/h·W at 10°F TD	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient dry-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Evaporative-Cooled Condensers	Condenser sizing	<68°F wb = 23°F TD 68°F to 75°F wb = 20°F TD >75°F wb = 16°F TD	RF1–2	
		Specific efficiency	180 Btu/h·W at 70°F wb and 100°F SCT	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient wet-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Compressor Systems	Suction groups/levels	Optimize suction groups based on load dynamics	RF8, 13–14	
		Indirect cooling design	CO_2 phase-change heat exchange Low temperature = two stage or cascade	RF9	
		Compressor group staging and capacity control	Continuously variable capacity control on each suction group	RF10	
		Suction group setpoint determination	Suction groups <25°F SCT = floating suction pressure logic	RF10	
		Mechanical subcooling	Low-temperature/medium-temperature SST ≥ 30°F	RF11	
	Display Cases and Walk-In Coolers/Freezers	Case type	Closed = dairy, beverage, deli, packaged salads, horizontal (tub), and low temperature Open = red meat, wet rack produce, specialty	RF15–16	
		Case door heater	Low temperature = 50–120 W/door with PWM controller Medium temperature = zero watt glass doors	RF17	
		Walk-in construction	Walk-in cooler insulation = R-25 Walk-in freezer insulation = R-40 Strip curtains on all doors	RF18, EN12	
		Walk-in doors	Doors < 48 in. = spring assist or cam-lift gravity hinge Doors > 48 in. = spring action door closer All doors = hydraulic door closer	RF19	
		Walk-in door switches and alarms	Freezer = fan and cooling OFF when door is open Cooler = fan and cooling OFF when door is open All = override and alarm integration	RF20	
		Walk-in box fan control	Two-speed fan control Reduce to 80% speed when load is below 50%	RF21	
		Lighting design	Open/closed cases ≤ 10 W/ft Meat cases ≤ 25 W/ft Display doors = motion sensors with 3 min time-out Walk-ins = vaportight LED with motion/vacancy sensor and/or door trigger switch	RF22, EL23, EL25	
		Defrost	Electric defrost = low-temperature cases, freezers, meat coolers; air defrost = all others Time initiation; temperature termination	RF23	
		Temperature and superheat control	Electronic modulating temperature control with floating suction pressure integration	RF24	
		Unit cooler	EC motor; select coils at 8°F TD on freezers Freezer fin spacing = 4 FPI Cooler fin spacing = no recommendation	RF25	
		Liquid-suction heat exchangers	High-efficiency heat exchange at piping exit; sized for additional suction superheat at design of 12°F for freezers and 6°F for coolers	RF26	
	Heat Recovery—HVAC	Heat recovery from refrigeration	Use at least 25% of the refrigeration design heat of rejection for space heating	RF27–31	
SHW	Service Water Heating	Gas water heater efficiency	See Table 5-13	WH1–4	
		Electric water heater efficiency	See Table 5-14	WH1–4	
		Electric heat pump water heater	COP 3.0 (interior heat source)	WH1–4	
		Recirculation pump control	Time clock or demand control	WH4	
		Pipe insulation (d < 1.5 in. / d ≥ 1.5 in.)	See Table 5-15	WH5	
		Refrigerant heat recovery	Superheat recovery using low-temperature system	WH6, RF27, KE4	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 2 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
HVAC	Distributed MA SZVAV DX Packaged RTU	Heating efficiency	Indirect gas heat = 80%	HV1–4, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4	
	Distributed MA SZVAV Chilled-Water RTU	Heating efficiency	High-efficiency condensing boiler = 90% Indirect gas heat = 80%	HV1–3, 5, 12	
		Cooling efficiency	Positive-displacement air-cooled chillers: <150 tons = 10.1 EER, 13.7 IPLV ≥150 tons = 10.1 EER, 14.0 IPLV	HV1–3, 5, 12	
		Fan motor efficiency	No recommendation*	HV1–3, 5, 24	
		Pump motor efficiency	No recommendation*	HV1–3, 5	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 5	
	Distributed SZVAV DX Packaged RTU with DOAS	Heating efficiency	Indirect gas heat = 80%	HV1–4, 6, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 6, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4, 6	
	Distributed SZCV Air-Source HP Packaged RTU with DOAS	Heating efficiency	See Table 5-17	HV1–3, 7, 12	
		Cooling efficiency	See Table 5-17	HV1–3, 7, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 7	
	SZVAV WSHP Packaged RTU with DOAS	Heating efficiency	See Table 5-18	HV1–3, 8, 12	
		Cooling efficiency	See Table 5-18	HV1–3, 8, 12	
		Pump motor efficiency	As per Standard 90.1-2013, Table 10.8-2	HV1–3, 8	
		Fluid cooler fans	Variable-speed control	HV1–3, 8, 9, 24	
		High-efficiency boiler	90% E_c	HV1–3, 8	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 8	
	DOAS	Heating efficiency	See Table 5-19	HV10	
		Dehumidification efficiency	See Table 5-20	HV10	
		Moisture removal efficiency	See Table 5-20	HV10	
		Fan and motor	65% mechanical/motor efficiency in absence of whole-unit EER rating Motor efficiency as per Standard 90.1-2013, Table 10.8	HV10, 24	
		Maximum external static pressure	0.7 in. w.c.	HV10	
	Ventilation/Exhaust	Cooling capacity—economizers	≥ 54,000 Btu/h	HV17	
		DCV/performance-based ventilation	Control ventilation air based on pollutant concentrations in space (CO_2, VOCs, etc.)	HV18	
		Exhaust air recovery—in-store cooking with > Standard 62.1 area exhaust requirement	A (humid) zones = 75% total effectiveness B (dry) zones = 75% sensible effectiveness Capture minimum 80% available exhaust air, including general and bathroom, for energy recovery	HV13	
		Exhaust airflow control	Control based on occupancy using time clock or occupancy sensor	HV20	
	Ducts and Dampers	Outdoor air damper	Motorized damper	HV13	
		Duct seal class	Seal Class A	HV21, 23	
		Insulation level	R-6	HV22	
QA	Quality Assurance	Commissioning	Commission throughout design and construction	QA1–13	
		M&V	Employ post-occupancy monitoring	QA14–17	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.



Zone 3

Alabama

All counties except:
Baldwin
Mobile

Arizona

Cochise
Graham
Greenlee
Mohave
Santa Cruz

Arkansas

All counties except:
Baxter
Benton
Boone
Carroll
Fulton
Izard
Madison
Marion
Newton
Searcy
Stone
Washington

California

All counties except:
Alpine
Amador
Calaveras
Del Norte
El Dorado
Humboldt
Imperial
Inyo
Lake
Lassen
Mariposa
Modoc
Mono
Nevada
Plumas
Sierra
Siskiyou
Trinity
Tuolumne

Georgia

All counties except:
Appling
Atkinson
Bacon
Baker
Banks
Berrien
Brantley
Brooks
Bryan
Catoosa
Camden
Charlton
Chatham
Chattooga
Clinch
Colquitt
Cook
Dade
Dawson
Decatur
Echols
Effingham
Evans
Fannin
Floyd
Franklin
Gilmer
Glynn
Gordon
Grady
Habersham
Hall
Jeff Davis
Lanier
Liberty
Long
Lowndes
Lumpkin
McIntosh
Miller
Mitchell
Murray
Pickens
Pierce
Rabun
Seminole
Stephens
Tattnall
Thomas
Toombs
Towns
Union
Walker
Ware
Wayne
White
Whitfield

Louisiana

Bienville
Bossier
Caddo
Caldwell
Catahoula
Claiborne
Concordia
De Soto
East Carroll
Franklin
Grant
Jackson
La Salle
Lincoln
Madison
Morehouse
Natchitoches
Ouachita
Red River
Richland
Sabine
Tensas
Union
Vernon
Webster
West Carroll
Winn

Mississippi

All counties except:
Hancock
Harrison
Jackson
Pearl River
Stone

New Mexico

Chaves
Dona Ana
Eddy
Hidalgo
Lea
Luna
Otero

Nevada

Clark

Texas

Andrews
Archer
Baylor
Blanco
Borden
Bowie
Brewster
Brown
Burnet
Callahan
Camp
Cass
Childress
Clay
Coke
Coleman
Collingsworth
Collin
Comanche
Concho
Cottle
Cooke
Crane
Crockett
Crosby
Culberson
Dallas
Dawson
Delta
Denton
Dickens
Eastland
Ector
El Paso
Ellis
Erath
Fannin
Fisher
Foard
Franklin
Gaines
Garza
Gillespie
Glasscock
Grayson
Gregg
Hall
Hamilton
Hardeman
Harrison
Haskell
Hemphill
Henderson
Hood
Hopkins
Howard
Hudspeth
Hunt
Irion
Jack
Jeff Davis
Johnson
Jones
Kaufman
Kendall
Kent
Kerr
Kimble
King
Knox
Lamar
Lampasas
Llano
Loving
Lubbock
Lynn
Marion
Martin
Mason
McCulloch
Menard
Midland
Mills
Mitchell
Montague
Morris
Motley
Nacogdoches
Navarro
Nolan
Palo Pinto
Panola
Parker
Pecos
Presidio
Rains
Reagan
Reeves
Red River
Rockwall
Runnels
Rusk
Sabine
San Augustine
San Saba
Schleicher
Scurry
Shackelford
Shelby
Smith
Somervell
Stephens
Sterling
Stonewall
Sutton
Tarrant
Taylor
Terrell
Terry
Throckmorton
Titus
Tom Green
Upshur
Upton
Van Zandt
Ward
Wheeler
Wichita
Wilbarger
Winkler
Wise
Wood
Young

Utah

Washington

North Carolina

Anson
Beaufort
Bladen
Brunswick
Cabarrus
Camden
Carteret
Chowan
Columbus
Craven
Cumberland
Currituck
Dare
Davidson
Duplin
Edgecombe
Gaston
Greene
Hoke
Hyde
Johnston
Jones
Lenoir
Martin
Mecklenburg
Montgomery
Moore
New Hanover
Onslow
Pamlico
Pasquotank
Pender
Perquimans
Pitt
Randolph
Richmond
Robeson
Rowan
Sampson
Scotland
Stanly
Tyrrell
Union
Washington
Wayne
Wilson

Oklahoma

All counties except:
Beaver
Cimarron
Texas

South Carolina

All counties

Tennessee

Chester
Crockett
Dyer
Fayette
Hardeman
Hardin
Haywood
Henderson
Lake
Lauderdale
Madison
McNairy
Shelby
Tipton

Climate Zone 3 Recommendation Table for Grocery Stores

	Item	Component	Recommendation	How-to Tips	✓
Envelope	Roofs	Insulation entirely above deck	R-25.0 c.i.	EN1, 3, 19–21	
		Metal building	R-19.0 + R-10.0 FC	EN2–3, 19–21	
		SRI	78	EN3	
	Walls	Mass (HC > 7 Btu/ft^2)	R- 7.6 c.i.	EN4, 19–21	
		Metal building	R-0.0 + R-9.8 c.i.	EN5, 19–21	
		Steel framed	R-13.0 + R-5.0 c.i.	EN6, 19–21	
	Floors	Mass	R-10.0 c.i.	EN7, 19–21	
		Steel framed	R-30.0	EN8, 19–21	
	Slabs	Unheated	No recommendation*	EN19–21	
		Heated	R-15 for 24 in.	EN10-11, 19–21	
		Freezer box floors	See Table 5-2	EN12	
	Doors	Swinging	U-0.70	EN13, 20	
		Nonswinging	U-0.16	En14, 20	
		Vehicular/dock infiltration—door closed	0.28 cfm/ft^2 of door area	EN15, 20	
		Vehicular/dock infiltration—door open, truck in place	Weather seals for dock, levelers, trailer hinges	EN16, 20	
	Vestibules	All building entrances	Yes	EN17, 20	
	Continuous Air Barrier	Continuous air barrier	Entire building envelope	EN18	
	View Fenestration—All Orientations	Thermal transmittance	U-0.50	EN22–25, 28	
		SHGC	0.25	EN23–25	
		VT/SHGC	>1.10	EN29	
Daylighting / Lighting	Daylighting	Skylights or rooftop monitors	Required per Standard 90.1 when sales floor ceiling height > 15 ft and area is ≥ 2500 ft^2	DL1–10	
		Daylight area	≥ 50% of the sales floor	DL1–10	
		Skylight to daylight area	Minimum = 3%, maximum = 5%	DL3	
		VT	Skylight VT ≥ 0.40	DL3	
		Skylight SHGC	0.35	EN25–27	
		Skylight thermal transmittance	U-0.55	DL4	
		Effective aperture	Skylight effective aperture ≥ 1%	DL8	
		Glazing material/diffuser	Measured haze value > 90% when tested according to ASTM D1003 (ASTM 2013)	DL3	
		Controls	General lighting auto control in response to daylight	DL6–7, 9–10	
	Interior Lighting	Ambient and accent lighting	Sales floor LPD = 1.15 W/ft^2	EL1, 3–7, 9–11, 17, 25, 30	
			Stock room LPD = 0.6 W/ft^2	EL1, 4, 9–11, 17–19, 29–30	
			Average of all other LPDs = 0.8 W/ft^2	EL1–2, 4, 8–11, 17–19, 25–27, 29–30	
		Light source lamp efficacy (mean lumens per watt)	LED—general = 80, accent = 75, case = 50 LED—exterior = 75 T8 and T5—greater than 2 ft = 92, 2 ft and less = 85	EL12, 14	
		T8 ballasts	Nondimming = NEMA Premium instant start Dimming = NEMA Premium program start	EL12	
		All T5/T5HO ballasts	Electronic program start	EL12	
		All CFL and HID ballasts	Electronic	EL12	
		Lighting controls	Sales floor = time switch—auto to 25% during closed hours, auto to 2% or less when unoccupied	EL16	
			Additional specialty lighting = auto ON only during store open hours	EL16	
			Stock room, restrooms = auto ON/OFF occupancy sensors	EL15	
			All other = manual ON, auto OFF occupancy sensors	EL15	
			After hours = maximum 2% of total building LPD	EL16	
	Exterior Lighting	Façade and landscape lighting	LPD = 0.075 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = auto OFF at business closing	EL33–35	
		Parking lots and drives	LPD = 0.08 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL31, 34–35	
		Walkways, plazas, and special feature areas	LPD = 0.16 W/ft^2 in LZ3 and LZ4, 0.14 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL32, 34–35	
		All other exterior lighting	No recommendation*	EL34–35	
Plug Loads	Plug Loads	Equipment	Specify ENERGY STAR Use laptops, mini desktops, thin clients LED for security lighting and signage	PL1–6, 8–9	
		Controls	Motion-sensing outlet and power strips, network computer control, timer switches	PL1, 7–8	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 3 Recommendation Table for Grocery Stores *(Continued)*

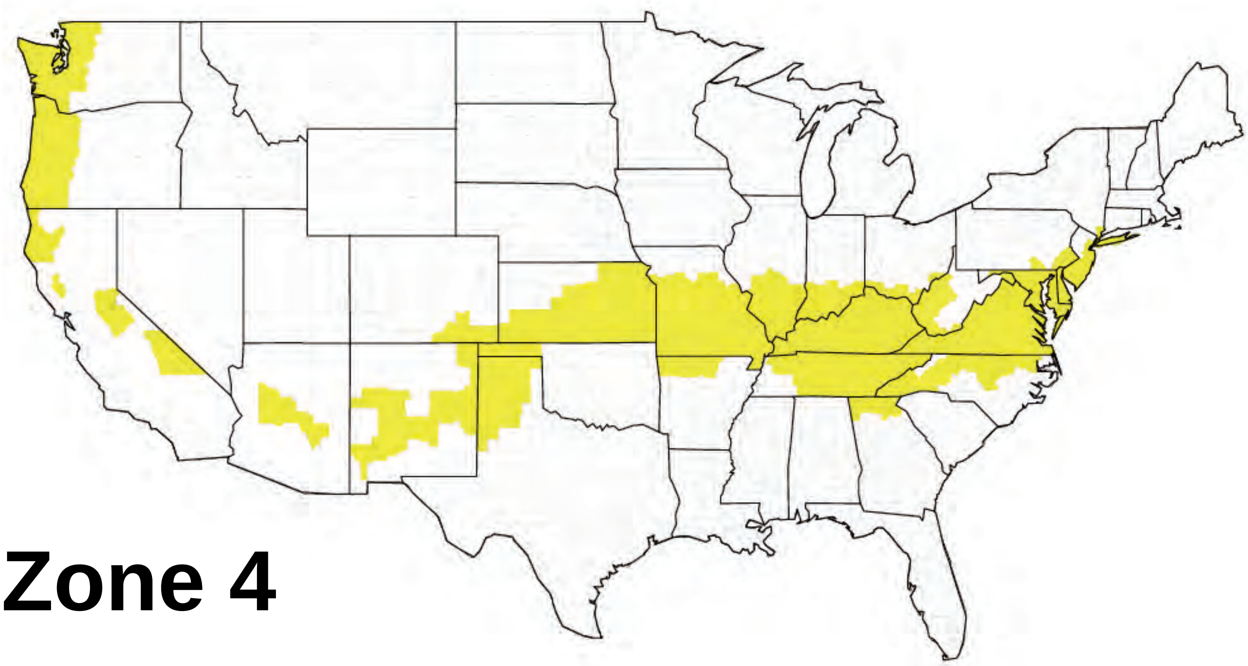
	Item	Component	Recommendation	How-to Tips	✓
Kitchen	Kitchen Equipment	Cooking equipment	ENERGY STAR or utility rebate-qualified equipment	KE1–2, 4, 6	
		Exhaust hoods	Side panels or end walls, larger overhangs, rear seal behind appliances, proximity hoods, DCKV	KE1, 3, 5, 6	
Refrigeration	Air-Cooled Condensers	Condenser sizing	Low temperature = 10°F TD Medium temperature = 15°F TD	RF1–2	
		Fin spacing	Maximum 10 FPI (excluding micro channel)	RF1	
		Specific efficiency	85 Btu/h·W at 10°F TD	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient dry-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Evaporative-Cooled Condensers	Condenser sizing	<68°F wb = 23°F TD 68°F to 75°F wb = 20°F TD >75°F wb = 16°F TD	RF1–2	
		Specific efficiency	180 Btu/h·W at 70°F wb and 100°F SCT	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient wet-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Compressor Systems	Suction groups/levels	Optimize suction groups based on load dynamics	RF8, 13–14	
		Indirect cooling design	CO_2 phase-change heat exchange Low temperature = two stage or cascade	RF9	
		Compressor group staging and capacity control	Continuously variable capacity control on each suction group	RF10	
		Suction group setpoint determination	Suction groups <25°F SCT = floating suction pressure logic	RF10	
		Mechanical subcooling	Low-temperature/medium-temperature SST ≥ 30°F	RF11	
	Display Cases and Walk-In Coolers/Freezers	Case type	Closed = dairy, beverage, deli, packaged salads, horizontal (tub), and low temperature Open = red meat, wet rack produce, specialty	RF15–16	
		Case door heater	Low temperature = 50–120 W/door with PWM controller Medium temperature = zero watt glass doors	RF17	
		Walk-in construction	Walk-in cooler insulation = R-25 Walk-in freezer insulation = R-40 Strip curtains on all doors	RF18, EN12	
		Walk-in doors	Doors < 48 in. = spring assist or cam-lift gravity hinge Doors > 48 in. = spring action door closer All doors = hydraulic door closer	RF19	
		Walk-in door switches and alarms	Freezer = fan and cooling OFF when door is open Cooler = fan and cooling OFF when door is open All = override and alarm integration	RF20	
		Walk-in box fan control	Two-speed fan control Reduce to 80% speed when load is below 50%	RF21	
		Lighting design	Open/closed cases ≤ 10 W/ft Meat cases ≤ 25 W/ft Display doors = motion sensors with 3 min time-out Walk-ins = vaportight LED with motion/vacancy sensor and/or door trigger switch	RF22, EL23, EL25	
		Defrost	Electric defrost = low-temperature cases, freezers, meat coolers; air defrost = all others Time initiation; temperature termination	RF23	
		Temperature and superheat control	Electronic modulating temperature control with floating suction pressure integration	RF24	
		Unit cooler	EC motor; select coils at 8°F TD on freezers Freezer fin spacing = 4 FPI Cooler fin spacing = no recommendation	RF25	
		Liquid-suction heat exchangers	High-efficiency heat exchange at piping exit; sized for additional suction superheat at design of 12°F for freezers and 6°F for coolers	RF26	
	Heat Recovery–HVAC	Heat recovery from refrigeration	Use at least 25% of the refrigeration design heat of rejection for space heating	RF27–31	
SHW	SWH	Gas water heater efficiency	See Table 5-13	WH1–4	
		Electric water heater efficiency	See Table 5-14	WH1–4	
		Electric heat pump water heater	COP 3.0 (interior heat source)	WH1–4	
		Recirculation pump control	Time clock or demand control	WH4	
		Pipe insulation (d < 1.5 in. / d ≥ 1.5 in.)	See Table 5-15	WH5	
		Refrigerant heat recovery	Superheat recovery using low-temperature system	WH6, RF27, KE4	

*Note: Where the table says “No recommendation,” the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 3 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
HVAC	Distributed MA SZVAV DX Packaged RTU	Heating efficiency	Indirect gas heat = 80%	HV1–4, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4	
	Distributed MA SZVAV Chilled-Water RTU	Heating efficiency	High-efficiency condensing boiler = 90% Indirect gas heat = 80%	HV1–3, 5, 12	
		Cooling efficiency	Positive-displacement air-cooled chillers: <150 tons = 10.1 EER, 13.7 IPLV ≥150 tons = 10.1 EER, 14.0 IPLV	HV1–3, 5, 12	
		Fan motor efficiency	No recommendation*	HV1–3, 5, 24	
		Pump motor efficiency	No recommendation*	HV1–3, 5	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 5	
	Distributed SZVAV DX Packaged RTU with DOAS	Heating efficiency	Indirect gas heat = 80%	HV1–4, 6, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 6, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4, 6	
	Distributed SZCV Air-Source HP Packaged RTU with DOAS	Heating efficiency	See Table 5-17	HV1–3, 7, 12	
		Cooling efficiency	See Table 5-17	HV1–3, 7, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 7	
	SZVAV WSHP Packaged RTU with DOAS	Heating efficiency	See Table 5-18	HV1–3, 8, 12	
		Cooling efficiency	See Table 5-18	HV1–3, 8, 12	
		Pump motor efficiency	As per Standard 90.1-2013, Table 10.8-2	HV1–3, 8	
		Fluid cooler fans	Variable-speed control	HV1–3, 8, 9, 24	
		High-efficiency boiler	90% E_c	HV1–3, 8	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 8	
	DOAS	Heating efficiency	See Table 5-19	HV10	
		Dehumidification efficiency	See Table 5-20	HV10	
		Moisture removal efficiency	See Table 5-20	HV10	
		Fan and motor	65% mechanical/motor efficiency in absence of whole-unit EER rating Motor efficiency as per Standard 90.1-2013, Table 10.8	HV10, 24	
		Maximum external static pressure	0.7 in. w.c.	HV10	
	Ventilation/Exhaust	Cooling capacity—economizers	≥ 54,000 Btu/h	HV17	
		DCV/performance-based ventilation	Control ventilation air based on pollutant concentrations in space (CO_2, VOCs, etc.)	HV18	
		Exhaust air recovery—in-store cooking with > Standard 62.1 area exhaust requirement	A (humid) zones = 75% total effectiveness B (dry) zones = 75% sensible effectiveness C (marine) zones = no recommendation Capture minimum 80% available exhaust air, including general and bathroom, for energy recovery	HV13	
		Exhaust airflow control	Control based on occupancy using time clock or occupancy sensor	HV20	
	Ducts and Dampers	Outdoor air damper	Motorized damper	HV13	
		Duct seal class	Seal Class A	HV21, 23	
		Insulation level	R-6	HV22	
QA	Quality Assurance	Commissioning	Commission throughout design and construction	QA1–13	
		M&V	Employ post-occupancy monitoring	QA14–17	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.



Zone 4

Arizona
Gila
Yavapai

Arkansas
Baxter
Benton
Boone
Carroll
Fulton
Izard
Madison
Marion
Newton
Searcy
Stone
Washington

California
Amador
Calaveras
Del Norte
El Dorado
Humboldt
Inyo
Lake
Mariposa
Trinity
Tuolumne

Colorado
Baca
Las Animas
Otero

Delaware
All counties

District of Columbia

Georgia
Banks
Catoosa
Chattooga
Dade
Dawson
Fannin
Floyd
Franklin
Gilmer
Gordon
Habersham
Hall
Lumpkin
Murray
Pickens
Rabun
Stephens
Towns
Union
Walker
White
Whitfield

Illinois
Alexander
Bond
Brown
Christian
Clay
Clinton
Crawford
Edwards
Effingham
Fayette
Franklin
Gallatin
Hamilton
Hardin
Jackson
Jasper
Jefferson
Johnson
Lawrence
Macoupin
Madison
Marion
Massac
Monroe
Montgomery
Perry
Pope
Pulaski
Randolph
Richland
Saline
Shelby
St. Clair
Union
Wabash
Washington
Wayne
White
Williamson

Indiana
Clark
Crawford
Daviess
Dearborn
Dubois
Floyd
Gibson
Greene
Harrison
Jackson
Jefferson
Jennings
Knox
Lawrence
Martin
Monroe
Ohio
Orange
Perry
Pike
Posey
Ripley
Scott
Spencer
Sullivan
Switzerland
Vanderburgh
Warrick
Washington

Kansas
All counties except:
Cheyenne
Cloud
Decatur
Ellis
Gove
Graham
Greeley
Hamilton
Jewell
Lane
Logan
Mitchell
Ness
Norton
Osborne
Phillips
Rawlins
Republic
Rooks
Scott
Sheridan
Sherman
Smith
Thomas
Trego
Wallace
Wichita

Kentucky
All counties

Maryland
All counties except:
Garrett

Missouri
All counties except:
Adair
Andrew
Atchison
Buchanan
Caldwell
Chariton
Clark
Clinton
Daviess
DeKalb
Gentry
Grundy
Harrison
Holt
Knox
Lewis
Linn
Livingston
Macon
Marion
Mercer
Nodaway
Pike
Putnam
Ralls
Schuyler
Scotland
Shelby
Sullivan
Worth

New Jersey
All counties except:
Bergen
Hunterdon
Mercer
Morris
Passaic
Somerset
Sussex
Warren

New Mexico
Bernalillo
Cibola
Curry
DeBaca
Grant
Guadalupe
Lincoln
Quay
Roosevelt
Sierra
Socorro
Union
Valencia

New York
Bronx
Kings
Nassau
New York
Queens
Richmond
Suffolk
Westchester

North Carolina
Alamance
Alexander
Bertie
Buncombe
Burke
Caldwell
Caswell
Catawba
Chatham
Cherokee
Clay
Cleveland
Davie
Durham
Forsyth
Franklin
Gates
Graham
Granville
Guilford
Halifax
Harnett
Haywood
Henderson
Hertford
Iredell
Jackson
Lee
Lincoln
Macon
Madison
McDowell
Nash
Northampton
Orange
Person
Polk
Rockingham
Rutherford
Stokes
Surry
Swain
Transylvania
Vance
Wake
Warren
Wilkes
Yadkin

Ohio
Adams
Brown
Clermont
Gallia
Hamilton
Lawrence
Pike
Scioto
Washington

Oklahoma
Beaver
Cimarron
Texas

Oregon
Benton
Clackamas
Clatsop
Columbia
Coos
Curry
Douglas
Jackson
Josephine
Lane
Lincoln
Linn
Marion
Multnomah
Polk
Tillamook
Washington
Yamhill

Pennsylvania
Bucks
Chester
Delaware
Montgomery
Philadelphia
York

Tennessee
All counties except:
Chester
Crockett
Dyer
Fayette
Hardeman
Hardin
Haywood
Henderson
Lake
Lauderdale
Madison
McNairy
Shelby
Tipton

Texas
Armstrong
Bailey
Briscoe
Carson
Castro
Cochran
Dallam
Deaf Smith
Donley
Floyd
Gray
Hale
Hansford
Hartley
Hockley
Hutchinson
Lamb
Lipscomb
Moore
Ochiltree
Oldham
Parmer
Potter
Randall
Roberts
Sherman
Swisher
Yoakum

Virginia
All counties

Washington
Clallam
Clark
Cowlitz
Grays Harbor
Island
Jefferson
King
Kitsap
Lewis
Mason
Pacific
Pierce
San Juan
Skagit
Snohomish
Thurston
Wahkiakum
Whatcom

West Virginia
Berkeley
Boone
Braxton
Cabell
Calhoun
Clay
Gilmer
Jackson
Jefferson
Kanawha
Lincoln
Logan
Mason
McDowell
Mercer
Mingo
Monroe
Morgan
Pleasants
Putnam
Ritchie
Roane
Tyler
Wayne
Wirt
Wood
Wyoming

Climate Zone 4 Recommendation Table for Grocery Stores

	Item	Component	Recommendation	How-to Tips	✓
Envelope	Roofs	Insulation entirely above deck	R-30.0 c.i.	EN1, 19–21	
		Metal building	R-19.0 + R-11.0 Ls	EN2, 19–21	
		SRI	No recommendation*	None	
	Walls	Mass (HC > 7 Btu/ft^2)	R-9.5 c.i.	EN4, 19–21	
		Metal building	R-0.0 + R-15.8 c.i.	EN5, 19–21	
		Steel framed	R-13.0 + R-7.5 c.i.	EN6, 19–21	
	Floors	Mass	R-14.6 c.i.	EN7, 19–21	
		Steel framed	R-30.0	EN8, 19–21	
	Slabs	Unheated	R-15 for 24 in.	EN9, 11, 19–21	
		Heated	R-20 for 24 in.	EN10–11, 19–21	
		Freezer box floors	See Table 5-2	EN12	
	Doors	Swinging	U-0.50	EN13, 20	
		Nonswinging	U-0.16	En14, 20	
		Vehicular/dock infiltration—door closed	0.28 cfm/ft^2 of door area	EN15, 20	
		Vehicular/dock infiltration—door open, truck in place	Weather seals for dock, levelers, trailer hinges	EN16, 20	
	Vestibules	All building entrances	Yes	EN17, 20	
	Continuous Air Barrier	Continuous air barrier	Entire building envelope	EN18	
	View Fenestration—All Orientations	Thermal transmittance	U-0.42	EN22–23, 26–28	
		SHGC	0.40	EN23, 26–28	
		VT/SHGC	>1.10	EN29	
Daylighting / Lighting	Daylighting	Skylights or rooftop monitors	Required per Standard 90.1 when sales floor ceiling height > 15 ft and area is ≥ 2500 ft^2	DL1–10	
		Daylight area	≥ 50% of the sales floor	DL1–10	
		Skylight to daylight area	Minimum = 3%, maximum = 5%	DL3	
		VT	Skylight VT ≥ 0.40	DL3	
		Skylight SHGC	0.35	EN25–27	
		Skylight thermal transmittance	U-0.50	DL4	
		Effective aperture	Skylight effective aperture ≥ 1%	DL8	
		Glazing material/diffuser	Measured haze value > 90% when tested according to ASTM D1003 (ASTM 2013)	DL3	
		Controls	General lighting auto control in response to daylight	DL6–7, 9–10	
	Interior Lighting	Ambient and accent lighting	Sales floor LPD = 1.15 W/ft^2	EL1, 3–7, 9–11, 17, 25, 30	
			Stock room LPD = 0.6 W/ft^2	EL1, 4, 9–11, 17–19, 29–30	
			Average of all other LPDs = 0.8 W/ft^2	EL1–2, 4, 8–11, 17–19, 25–27, 29–30	
		Light source lamp efficacy (mean lumens per watt)	LED—general = 80, accent = 75, case = 50 LED—exterior = 75 T8 and T5—greater than 2 ft = 92, 2 ft and less = 85	EL12, 14	
		T8 ballasts	Nondimming = NEMA Premium instant start Dimming = NEMA Premium program start	EL12	
		All T5/T5HO ballasts	Electronic program start	EL12	
		All CFL and HID ballasts	Electronic	EL12	
		Lighting controls	Sales floor = time switch—auto to 25% during closed hours, auto to 2% or less when unoccupied	EL16	
			Additional specialty lighting = auto ON only during store open hours	EL16	
			Stock room, restrooms = auto ON/OFF occupancy sensors	EL15	
			All other = manual ON, auto OFF occupancy sensors	EL15	
			After hours = maximum 2% of total building LPD	EL16	
	Exterior Lighting	Façade and landscape lighting	LPD = 0.075 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = auto OFF at business closing	EL33–35	
		Parking lots and drives	LPD = 0.08 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL31, 34–35	
		Walkways, plazas, and special feature areas	LPD = 0.16 W/ft^2 in LZ3 and LZ4, 0.14 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL32, 34–35	
		All other exterior lighting	No recommendation*	EL34–35	
Plug Loads	Plug Loads	Equipment	Specify ENERGY STAR Use laptops, mini desktops, thin clients LED for security lighting and signage	PL1–6, 8–9	
		Controls	Motion-sensing outlet and power strips, network computer control, timer switches	PL1, 7–8	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 4 Recommendation Table for Grocery Stores *(Continued)*

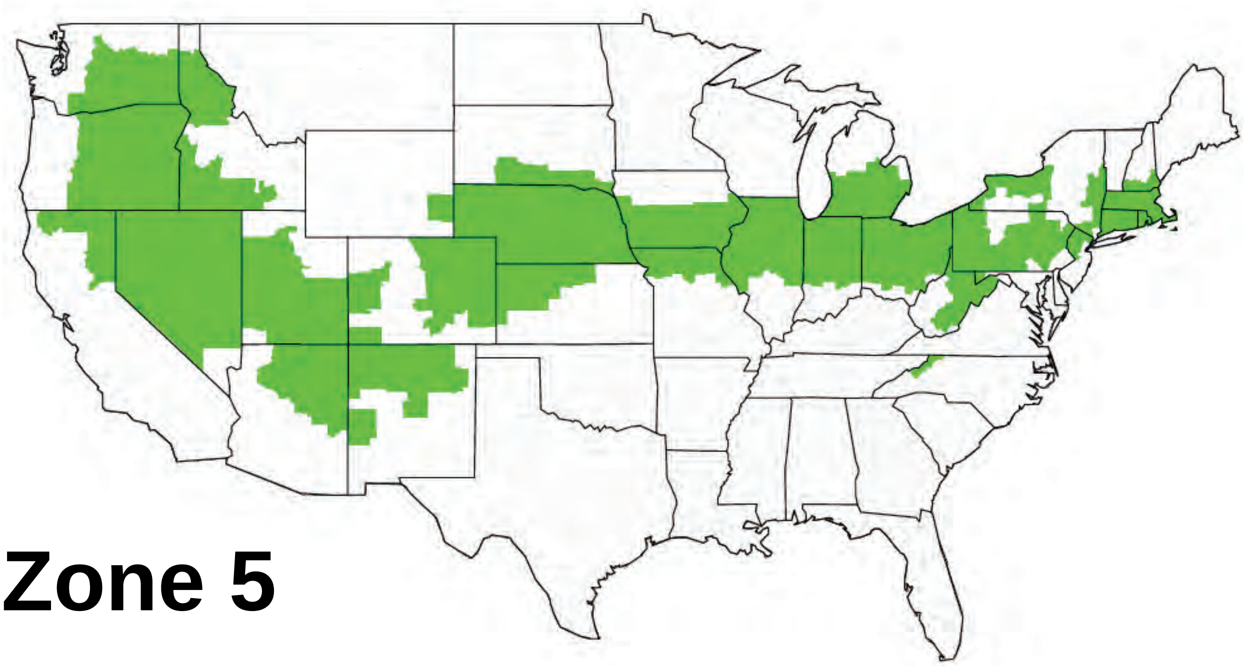
	Item	Component	Recommendation	How-to Tips	✓
Kitchen	Kitchen Equipment	Cooking equipment	ENERGY STAR or utility rebate-qualified equipment	KE1–2, 4, 6	
		Exhaust hoods	Side panels or end walls, larger overhangs, rear seal behind appliances, proximity hoods, DCKV	KE1, 3, 5, 6	
Refrigeration	Air-Cooled Condensers	Condenser sizing	Low temperature = 10°F TD Medium temperature = 15°F TD	RF1–2	
		Fin spacing	Maximum 10 FPI (excluding micro channel)	RF1	
		Specific efficiency	85 Btu/h·W at 10°F TD	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient dry-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Evaporative-Cooled Condensers	Condenser sizing	<68°F wb = 23°F TD 68°F to 75°F wb = 20°F TD >75°F wb = 16°F TD	RF1–2	
		Specific efficiency	180 Btu/h·W at 70°F wb and 100°F SCT	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient wet-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Compressor Systems	Suction groups/levels	Optimize suction groups based on load dynamics	RF8, 13–14	
		Indirect cooling design	CO_2 phase-change heat exchange Low temperature = two stage or cascade	RF9	
		Compressor group staging and capacity control	Continuously variable capacity control on each suction group	RF10	
		Suction group setpoint determination	Suction groups <25°F SCT = floating suction pressure logic	RF10	
		Mechanical subcooling	Low-temperature/medium-temperature SST ≥ 30°F	RF11	
	Display Cases and Walk-In Coolers/Freezers	Case type	Closed = dairy, beverage, deli, packaged salads, horizontal (tub), and low temperature Open = red meat, wet rack produce, specialty	RF15–16	
		Case door heater	Low temperature = 50–120 W/door with PWM controller Medium temperature = zero watt glass doors	RF17	
		Walk-in construction	Walk-in cooler insulation = R-25 Walk-in freezer insulation = R-40 Strip curtains on all doors	RF18, EN12	
		Walk-in doors	Doors < 48 in. = spring assist or cam-lift gravity hinge Doors > 48 in. = spring action door closer All doors = hydraulic door closer	RF19	
		Walk-in door switches and alarms	Freezer = fan and cooling OFF when door is open Cooler = fan and cooling OFF when door is open All = override and alarm integration	RF20	
		Walk-in box fan control	Two-speed fan control Reduce to 80% speed when load is below 50%	RF21	
		Lighting design	Open/closed cases ≤ 10 W/ft Meat cases ≤ 25 W/ft Display doors = motion sensors with 3 min time-out Walk-ins = vaportight LED with motion/vacancy sensor and/or door trigger switch	RF22, EL23, EL25	
		Defrost	Electric defrost = low-temperature cases, freezers, meat coolers; air defrost = all others Time initiation; temperature termination	RF23	
		Temperature and superheat control	Electronic modulating temperature control with floating suction pressure integration	RF24	
		Unit cooler	EC motor; select coils at 8°F TD on freezers Freezer fin spacing = 4 FPI Cooler fin spacing = no recommendation	RF25	
		Liquid-suction heat exchangers	High-efficiency heat exchange at piping exit; sized for additional suction superheat at design of 12°F for freezers and 6°F for coolers	RF26	
	Heat Recovery—HVAC	Heat recovery from refrigeration	Use at least 25% of the refrigeration design heat of rejection for space heating	RF27–31	
SHW	SWH	Gas water heater efficiency	See Table 5-13	WH1–4	
		Electric water heater efficiency	See Table 5-14	WH1–4	
		Electric heat pump water heater	COP 3.0 (interior heat source)	WH1–4	
		Recirculation pump control	Time clock or demand control	WH4	
		Pipe insulation (*d* < 1.5 in. / *d* ≥ 1.5 in.)	See Table 5-15	WH5	
		Refrigerant heat recovery	Superheat recovery using low-temperature system	WH6, RF27, KE4	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 4 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
HVAC	Distributed MA SZVAV DX Packaged RTU	Heating efficiency	Indirect gas heat = 80%	HV1–4, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4	
	Distributed MA SZVAV Chilled-Water RTU	Heating efficiency	High-efficiency condensing boiler = 90% Indirect gas heat = 80%	HV1–3, 5, 12	
		Cooling efficiency	Positive-displacement air-cooled chillers: <150 tons = 10.1 EER, 13.7 IPLV ≥150 tons = 10.1 EER, 14.0 IPLV	HV1–3, 5, 12	
		Fan motor efficiency	No recommendation*	HV1–3, 5, 24	
		Pump motor efficiency	No recommendation*	HV1–3, 5	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 5	
	Distributed SZVAV DX Packaged RTU with DOAS	Heating efficiency	Indirect gas heat = 80%	HV1–4, 6, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 6, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4, 6	
	Distributed SZCV Air-Source HP Packaged RTU with DOAS	Heating efficiency	See Table 5-17	HV1–3, 7, 12	
		Cooling efficiency	See Table 5-17	HV1–3, 7, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 7	
	SZVAV WSHP Packaged RTU with DOAS	Heating efficiency	See Table 5-18	HV1–3, 8, 12	
		Cooling efficiency	See Table 5-18	HV1–3, 8, 12	
		Pump motor efficiency	As per Standard 90.1-2013, Table 10.8-2	HV1–3, 8	
		Fluid cooler fans	Variable-speed control	HV1–3, 8, 9, 24	
		High-efficiency boiler	90% E_c	HV1–3, 8	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 8	
	DOAS	Heating efficiency	See Table 5-19	HV10	
		Dehumidification efficiency	See Table 5-20	HV10	
		Moisture removal efficiency	See Table 5-20	HV10	
		Fan and motor	65% mechanical/motor efficiency in absence of whole-unit EER rating Motor efficiency as per Standard 90.1-2013, Table 10.8	HV10, 24	
		Maximum external static pressure	0.7 in. w.c.	HV10	
	Ventilation/Exhaust	Cooling capacity—economizers	≥ 54,000 Btu/h	HV17	
		DCV/performance-based ventilation	Control ventilation air based on pollutant concentrations in space (CO_2, VOCs, etc.)	HV18	
		Exhaust air recovery—in-store cooking with > Standard 62.1 area exhaust requirement	A (humid) zones = 75% total effectiveness B (dry) zones = 75% sensible effectiveness C (marine) zones = 75% sensible effectiveness Capture minimum 80% available exhaust air, including general and bathroom, for energy recovery	HV13	
		Exhaust airflow control	Control based on occupancy using time clock or occupancy sensor	HV20	
	Ducts and Dampers	Outdoor air damper	Motorized damper	HV13	
		Duct seal class	Seal Class A	HV21, 23	
		Insulation level	R-6	HV22	
QA	Quality Assurance	Commissioning	Commission throughout design and construction	QA1–13	
		M&V	Employ post-occupancy monitoring	QA14–17	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.



Zone 5

Arizona
- Apache
- Coconino
- Navajo

California
- Lassen
- Modoc
- Nevada
- Plumas
- Sierra
- Siskiyou

Colorado
- Adams
- Arapahoe
- Bent
- Boulder
- Cheyenne
- Crowley
- Delta
- Denver
- Douglas
- Elbert
- El Paso
- Fremont
- Garfield
- Gilpin
- Huerfano
- Jefferson
- Kiowa
- Kit Carson
- La Plata
- Larimer
- Lincoln
- Logan
- Mesa
- Montezuma
- Montrose
- Morgan
- Phillips
- Prowers
- Pueblo
- Sedgwick
- Teller
- Washington
- Weld
- Yuma

Connecticut
- All counties

Idaho
- Ada
- Benewah
- Canyon
- Cassia
- Clearwater
- Elmore
- Gem
- Gooding
- Idaho
- Jerome
- Kootenai
- Latah
- Lewis
- Lincoln
- Minidoka
- Nez Perce
- Owyhee
- Payette
- Power
- Shoshone
- Twin Falls
- Washington

Illinois
All counties except:
- Alexander
- Bond
- Christian
- Clay
- Clinton
- Crawford
- Edwards
- Effingham
- Fayette
- Franklin
- Gallatin
- Hamilton
- Hardin
- Jackson
- Jasper
- Jefferson
- Johnson
- Lawrence
- Macoupin
- Madison
- Marion
- Massac
- Monroe
- Montgomery
- Perry
- Pope
- Pulaski
- Randolph
- Richland
- Saline
- Shelby
- St. Clair
- Union
- Wabash
- Washington
- Wayne
- White
- Williamson
- Brown

Indiana
All counties except:
- Clark
- Crawford
- Daviess
- Dearborn
- Dubois
- Floyd
- Gibson
- Greene
- Harrison
- Jackson
- Jefferson
- Jennings
- Knox
- Lawrence
- Martin
- Monroe
- Ohio
- Orange
- Perry
- Pike
- Posey
- Ripley
- Scott
- Spencer
- Sullivan
- Switzerland
- Vanderburgh
- Warrick
- Washington

Iowa
All counties except:
- Allamakee
- Black Hawk
- Bremer
- Buchanan
- Buena Vista
- Butler
- Calhoun
- Cerro Gordo
- Cherokee
- Chickasaw
- Clay
- Clayton
- Delaware
- Dickinson
- Emmet
- Fayette
- Floyd
- Franklin
- Grundy
- Hamilton
- Hancock
- Hardin
- Howard
- Humboldt
- Ida
- Kossuth
- Lyon
- Mitchell
- O'Brien
- Osceola
- Palo Alto
- Plymouth
- Pocahontas
- Sac
- Sioux
- Webster
- Winnebago
- Winneshiek
- Worth
- Wright

Kansas
- Cheyenne
- Cloud
- Decatur
- Ellis
- Gove
- Graham
- Greeley
- Hamilton
- Jewell
- Lane
- Logan
- Mitchell
- Ness
- Norton
- Osborne
- Phillips
- Rawlins
- Republic
- Rooks
- Scott
- Sheridan
- Sherman
- Smith
- Thomas
- Trego
- Wallace
- Wichita

Maryland
- Garrett

Massachusetts
- All counties

Michigan
- Allegan
- Barry
- Bay
- Berrien
- Branch
- Calhoun
- Cass
- Clinton
- Eaton
- Genesee
- Gratiot
- Hillsdale
- Ingham
- Ionia
- Jackson
- Kalamazoo
- Kent
- Lapeer
- Lenawee
- Livingston
- Macomb
- Midland
- Monroe
- Montcalm
- Muskegon
- Oakland
- Ottawa
- Saginaw
- Shiawassee
- St. Clair
- St. Joseph
- Tuscola
- Van Buren
- Washtenaw
- Wayne

Missouri
- Adair
- Andrew
- Atchison
- Buchanan
- Caldwell
- Chariton
- Clark
- Clinton
- Daviess
- DeKalb
- Gentry
- Grundy
- Harrison
- Holt
- Knox
- Lewis
- Linn
- Livingston
- Macon
- Marion
- Mercer
- Nodaway
- Pike
- Putnam
- Ralls
- Schuyler
- Scotland
- Shelby
- Sullivan
- Worth

Nebraska
- All counties

Nevada
All counties except:
- Clark

New Hampshire
- Cheshire
- Hillsborough
- Rockingham
- Strafford

New Jersey
- Bergen
- Hunterdon
- Mercer
- Morris
- Passaic
- Somerset
- Sussex
- Warren

New Mexico
- Catron
- Colfax
- Harding
- Los Alamos
- McKinley
- Mora
- Rio Arriba
- Sandoval
- San Juan
- San Miguel
- Santa Fe
- Taos
- Torrance

New York
- Albany
- Cayuga
- Chautauqua
- Chemung
- Columbia
- Cortland
- Dutchess
- Erie
- Genesee
- Greene
- Livingston
- Monroe
- Niagara
- Onondaga
- Ontario
- Orange
- Orleans
- Oswego
- Putnam
- Rensselaer
- Rockland
- Saratoga
- Schenectady
- Seneca
- Tioga
- Washington
- Wayne
- Yates

North Carolina
- Alleghany
- Ashe
- Avery
- Mitchell
- Watauga
- Yancey

Ohio
All counties except:
- Adams
- Brown
- Clermont
- Gallia
- Hamilton
- Lawrence
- Pike
- Scioto
- Washington

Oregon
- Baker
- Crook
- Deschutes
- Gilliam
- Grant
- Harney
- Hood River
- Jefferson
- Klamath
- Lake
- Malheur
- Morrow
- Sherman
- Umatilla
- Union
- Wallowa
- Wasco
- Wheeler

Pennsylvania
All counties except:
- Bucks
- Cameron
- Chester
- Clearfield
- Delaware
- Elk
- McKean
- Montgomery
- Philadelphia
- Potter
- Susquehanna
- Tioga
- Wayne
- York

Rhode Island
- All counties

South Dakota
- Bennett
- Bon Homme
- Charles Mix
- Clay
- Douglas
- Gregory
- Hutchinson
- Jackson
- Mellette
- Todd
- Tripp
- Union
- Yankton

Utah
All counties except:
- Box Elder
- Cache
- Carbon
- Daggett
- Duchesne
- Morgan
- Rich
- Summit
- Uintah
- Wasatch
- Washington

Washington
- Adams
- Asotin
- Benton
- Chelan
- Columbia
- Douglas
- Franklin
- Garfield
- Grant
- Kittitas
- Klickitat
- Lincoln
- Skamania
- Spokane
- Walla Walla
- Whitman
- Yakima

Wyoming
- Goshen
- Platte

West Virginia
- Barbour
- Brooke
- Doddridge
- Fayette
- Grant
- Greenbrier
- Hampshire
- Hancock
- Hardy
- Harrison
- Lewis
- Marion
- Marshall
- Mineral
- Monongalia
- Nicholas
- Ohio
- Pendleton
- Pocahontas
- Preston
- Raleigh
- Randolph
- Summers
- Taylor
- Tucker
- Upshur
- Webster
- Wetzel

Climate Zone 5 Recommendation Table for Grocery Stores

	Item	Component	Recommendation	How-to Tips	✓
Envelope	Roofs	Insulation entirely above deck	R-30.0 c.i.	EN1, 19–21	
		Metal building	R-19.0 + R-11.0 Ls	EN2, 19–21	
		SRI	No recommendation*	None	
	Walls	Mass (HC > 7 Btu/ft^2)	R-11.4 c.i.	EN4, 19–21	
		Metal building	R-0.0 + R-19.0 c.i.	EN5, 19–21	
		Steel framed	R-13.0 + R-10.0 c.i.	EN6, 19–21	
	Floors	Mass	R-14.6 c.i.	EN7, 19–21	
		Steel framed	R-30.0	EN8, 19–21	
	Slabs	Unheated	R-15 for 24 in.	EN9, 11, 19–21	
		Heated	R-20 for 48 in.	EN10–11, 19–21	
		Freezer box floors	See Table 5-2	EN12	
	Doors	Swinging	U-0.50	EN13, 20	
		Nonswinging	U-0.16	En14, 20	
		Vehicular/dock infiltration—door closed	0.28 cfm/ft^2 of door area	EN15, 20	
		Vehicular/dock infiltration—door open, truck in place	Weather seals for dock, levelers, trailer hinges	EN16, 20	
	Vestibules	All building entrances	Yes	EN17, 20	
	Continuous Air Barrier	Continuous air barrier	Entire building envelope	EN18	
	View Fenestration—All Orientations	Thermal transmittance	U-0.42	EN22–23, 26–28	
		SHGC	0.40	EN23, 26–28	
		VT/SHGC	>1.10	EN29	
Daylighting / Lighting	Daylighting	Skylights or rooftop monitors	Required per Standard 90.1 when sales floor ceiling height > 15 ft and area is ≥ 2500 ft^2	DL1–10	
		Daylight area	≥ 50% of the sales floor	DL1–10	
		Skylight to daylight area	Minimum = 3%, maximum = 5%	DL3	
		VT	Skylight VT ≥ 0.40	DL3	
		Skylight SHGC	0.35	EN25–27	
		Skylight thermal transmittance	U-0.50	DL4	
		Effective aperture	Skylight effective aperture ≥ 1%	DL8	
		Glazing material/diffuser	Measured haze value > 90% when tested according to ASTM D1003 (ASTM 2013)	DL3	
		Controls	General lighting auto control in response to daylight	DL6–7, 9–10	
	Interior Lighting	Ambient and accent lighting	Sales floor LPD = 1.15 W/ft^2	EL1, 3–7, 9–11, 17, 25, 30	
			Stock room LPD = 0.6 W/ft^2	EL1, 4, 9–11, 17–19, 29–30	
			Average of all other LPDs = 0.8 W/ft^2	EL1–2, 4, 8–11, 17–19, 25–27, 29–30	
		Light source lamp efficacy (mean lumens per watt)	LED—general = 80, accent = 75, case = 50 LED—exterior = 75 T8 and T5—greater than 2 ft = 92, 2 ft and less = 85	EL12, 14	
		T8 ballasts	Nondimming = NEMA Premium instant start Dimming = NEMA Premium program start	EL12	
		All T5/T5HO ballasts	Electronic program start	EL12	
		All CFL and HID ballasts	Electronic	EL12	
		Lighting controls	Sales floor = time switch—auto to 25% during closed hours, auto to 2% or less when unoccupied	EL16	
			Additional specialty lighting = auto ON only during store open hours	EL16	
			Stock room, restrooms = auto ON/OFF occupancy sensors	EL15	
			All other = manual ON, auto OFF occupancy sensors	EL15	
			After hours = maximum 2% of total building LPD	EL16	
	Exterior Lighting	Façade and landscape lighting	LPD = 0.075 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = auto OFF at business closing	EL33–35	
		Parking lots and drives	LPD = 0.08 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL31, 34–35	
		Walkways, plazas, and special feature areas	LPD = 0.16 W/ft^2 in LZ3 and LZ4, 0.14 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL32, 34–35	
		All other exterior lighting	No recommendation*	EL34–35	
Plug Loads	Plug Loads	Equipment	Specify ENERGY STAR Use laptops, mini desktops, thin clients LED for security lighting and signage	PL1–6, 8–9	
		Controls	Motion-sensing outlet and power strips, network computer control, timer switches	PL1, 7–8	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 5 Recommendation Table for Grocery Stores *(Continued)*

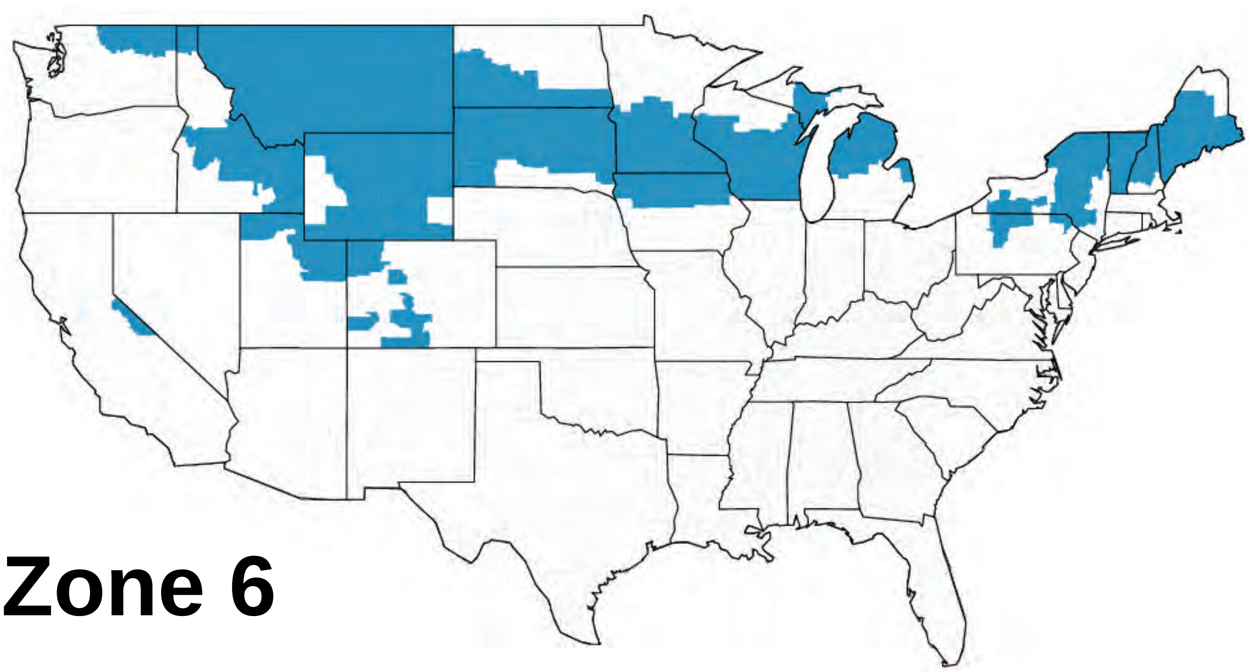
	Item	Component	Recommendation	How-to Tips	✓
Kitchen	Kitchen Equipment	Cooking equipment	ENERGY STAR or utility rebate-qualified equipment	KE1–2, 4, 6	
		Exhaust hoods	Side panels or end walls, larger overhangs, rear seal behind appliances, proximity hoods, DCKV	KE1, 3, 5, 6	
Refrigeration	Air-Cooled Condensers	Condenser sizing	Low temperature = 10°F TD Medium temperature = 15°F TD	RF1–2	
		Fin spacing	Maximum 10 FPI (excluding micro channel)	RF1	
		Specific efficiency	85 Btu/h·W at 10°F TD	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient dry-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Evaporative-Cooled Condensers	Condenser sizing	<68°F wb = 23°F TD 68°F to 75°F wb = 20°F TD >75°F wb = 16°F TD	RF1–2	
		Specific efficiency	180 Btu/h·W at 70°F wb and 100°F SCT	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient wet-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Compressor Systems	Suction groups/levels	Optimize suction groups based on load dynamics	RF8, 13–14	
		Indirect cooling design	CO_2 phase-change heat exchange Low temperature = two stage or cascade	RF9	
		Compressor group staging and capacity control	Continuously variable capacity control on each suction group	RF10	
		Suction group setpoint determination	Suction groups <25°F SCT = floating suction pressure logic	RF10	
		Mechanical subcooling	Low-temperature/medium-temperature SST ≥ 30°F	RF11	
	Display Cases and Walk-In Coolers/Freezers	Case type	Closed = dairy, beverage, deli, packaged salads, horizontal (tub), and low temperature Open = red meat, wet rack produce, specialty	RF15–16	
		Case door heater	Low temperature = 50–120 W/door with PWM controller Medium temperature = zero watt glass doors	RF17	
		Walk-in construction	Walk-in cooler insulation = R-25 Walk-in freezer insulation = R-40 Strip curtains on all doors	RF18, EN12	
		Walk-in doors	Doors < 48 in. = spring assist or cam-lift gravity hinge Doors > 48 in. = spring action door closer All doors = hydraulic door closer	RF19	
		Walk-in door switches and alarms	Freezer = fan and cooling OFF when door is open Cooler = fan and cooling OFF when door is open All = override and alarm integration	RF20	
		Walk-in box fan control	Two-speed fan control Reduce to 80% speed when load is below 50%	RF21	
		Lighting design	Open/closed cases ≤ 10 W/ft Meat cases ≤ 25 W/ft Display doors = motion sensors with 3 min time-out Walk-ins = vaportight LED with motion/vacancy sensor and/or door trigger switch	RF22, EL23, EL25	
		Defrost	Electric defrost = low-temperature cases, freezers, meat coolers; air defrost = all others Time initiation; temperature termination	RF23	
		Temperature and superheat control	Electronic modulating temperature control with floating suction pressure integration	RF24	
		Unit cooler	EC motor; select coils at 8°F TD on freezers Freezer fin spacing = 4 FPI Cooler fin spacing = no recommendation	RF25	
		Liquid-suction heat exchangers	High-efficiency heat exchange at piping exit; sized for additional suction superheat at design of 12°F for freezers and 6°F for coolers	RF26	
	Heat Recovery—HVAC	Heat recovery from refrigeration	Use at least 25% of the refrigeration design heat of rejection for space heating	RF27–31	
SHW	SWH	Gas water heater efficiency	See Table 5-13	WH1–4	
		Electric water heater efficiency	See Table 5-14	WH1–4	
		Electric heat pump water heater	COP 3.0 (interior heat source)	WH1–4	
		Recirculation pump control	Time clock or demand control	WH4	
		Pipe insulation (*d* < 1.5 in. / *d* ≥ 1.5 in.)	See Table 5-15	WH5	
		Refrigerant heat recovery	Superheat recovery using low-temperature system	WH6, RF27, KE4	

*Note: Where the table says “No recommendation,” the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 5 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
HVAC	Distributed MA SZVAV DX Packaged RTU	Heating efficiency	Indirect gas heat = 80%	HV1–4, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4	
	Distributed MA SZVAV Chilled-Water RTU	Heating efficiency	High-efficiency condensing boiler = 90% Indirect gas heat = 80%	HV1–3, 5, 12	
		Cooling efficiency	Positive-displacement air-cooled chillers: <150 tons = 10.1 EER, 13.7 IPLV ≥150 tons = 10.1 EER, 14.0 IPLV	HV1–3, 5, 12	
		Fan motor efficiency	No recommendation*	HV1–3, 5, 24	
		Pump motor efficiency	No recommendation*	HV1–3, 5	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 5	
	Distributed SZVAV DX Packaged RTU with DOAS	Heating efficiency	Indirect gas heat = 80%	HV1–4, 6, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 6, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4, 6	
	Distributed SZCV, Air-Source HP Packaged RTU with DOAS	Heating efficiency	See Table 5-17	HV1–3, 7, 12	
		Cooling efficiency	See Table 5-17	HV1–3, 7, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 7	
	SZVAV WSHP Packaged RTU with DOAS	Heating efficiency	See Table 5-18	HV1–3, 8, 12	
		Cooling efficiency	See Table 5-18	HV1–3, 8, 12	
		Pump motor efficiency	As per Standard 90.1-2013, Table 10.8-2	HV1–3, 8	
		Fluid cooler fans	Variable-speed control	HV1–3, 8, 9, 24	
		High-efficiency boiler	90% E_c	HV1–3, 8	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 8	
	DOAS	Heating efficiency	See Table 5-19	HV10	
		Dehumidification efficiency	See Table 5-20	HV10	
		Moisture removal efficiency	See Table 5-20	HV10	
		Fan and motor	65% mechanical/motor efficiency in absence of whole-unit EER rating Motor efficiency as per Standard 90.1-2013, Table 10.8	HV10, 24	
		Maximum external static pressure	0.7 in. w.c.	HV10	
	Ventilation/Exhaust	Cooling capacity—economizers	≥ 54,000 Btu/h	HV17	
		DCV/performance-based ventilation	Control ventilation air based on pollutant concentrations in space (CO_2, VOCs, etc.)	HV18	
		Exhaust air recovery—in-store cooking with > Standard 62.1 area exhaust requirement	A (humid) zones = 75% total effectiveness B (dry) zones = 75% sensible effectiveness Capture minimum 80% available exhaust air, including general and bathroom, for energy recovery	HV13	
		Exhaust airflow control	Control based on occupancy using time clock or occupancy sensor	HV20	
	Ducts and Dampers	Outdoor air damper	Motorized damper	HV13	
		Duct seal class	Seal Class A	HV21, 23	
		Insulation level	R-6	HV22	
QA	Quality Assurance	Commissioning	Commission throughout design and construction	QA1–13	
		M&V	Employ post-occupancy monitoring	QA14–17	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.



Zone 6

California

Alpine
Mono

Colorado

Alamosa
Archuleta
Chaffee
Conejos
Costilla
Custer
Dolores
Eagle
Moffat
Ouray
Rio Blanco
Saguache
San Miguel

Idaho

Adams
Bannock
Bear Lake
Bingham
Blaine
Boise
Bonner
Bonneville
Boundary
Butte
Camas
Caribou
Clark
Custer
Franklin
Fremont
Jefferson
Lemhi
Madison
Oneida
Teton
Valley

Iowa

Allamakee
Black Hawk
Bremer
Buchanan
Buena Vista
Butler
Calhoun
Cerro Gordo
Cherokee
Chickasaw
Clay
Clayton
Delaware
Dickinson
Emmet
Fayette
Floyd
Franklin
Grundy
Hamilton
Hancock
Hardin
Howard
Humboldt
Ida
Kossuth
Lyon
Mitchell
O'Brien
Osceola
Palo Alto
Plymouth
Pocahontas
Sac
Sioux
Webster
Winnebago
Winneshiek
Worth
Wright

Maine

All counties except:
Aroostook

Michigan

Alcona
Alger
Alpena
Antrim
Arenac
Benzie
Charlevoix
Cheboygan
Clare
Crawford
Delta
Dickinson
Emmet
Gladwin
Grand Traverse
Huron
Iosco
Isabella
Kalkaska
Lake
Leelanau
Manistee
Marquette
Mason
Mecosta
Menominee
Missaukee
Montmorency
Newaygo
Oceana
Ogemaw
Osceola
Oscoda
Otsego
Presque Isle
Roscommon
Sanilac
Wexford

Minnesota

Anoka
Benton
Big Stone
Blue Earth
Brown
Carver
Chippewa
Chisago
Cottonwood
Dakota
Dodge
Douglas
Faribault
Fillmore
Freeborn
Goodhue
Hennepin
Houston
Isanti
Jackson
Kandiyohi
Lac qui Parle
Le Sueur
Lincoln
Lyon
Martin
McLeod
Meeker
Morrison
Mower
Murray
Nicollet
Nobles
Olmsted
Pipestone
Pope
Ramsey
Redwood
Renville
Rice
Rock
Scott
Sherburne
Sibley
Stearns
Steele
Stevens
Swift
Todd
Traverse
Wabasha
Waseca
Washington
Watonwan
Winona
Wright
Yellow Medicine

Montana

All counties

New Hampshire

Belknap
Carroll
Coos
Grafton
Merrimack
Sullivan

New York

Allegany
Broome
Cattaraugus
Chenango
Clinton
Delaware
Essex
Franklin
Fulton
Hamilton
Herkimer
Jefferson
Lewis
Madison
Montgomery
Oneida
Otsego
Schoharie
Schuyler
Steuben
St. Lawrence
Sullivan
Tompkins
Ulster
Warren
Wyoming

North Dakota

Adams
Billings
Bowman
Burleigh
Dickey
Dunn
Emmons
Golden Valley
Grant
Hettinger
LaMoure
Logan
McIntosh
McKenzie
Mercer
Morton
Oliver
Ransom
Richland
Sargent
Sioux
Slope
Stark

Pennsylvania

Cameron
Clearfield
Elk
McKean
Potter
Susquehanna
Tioga
Wayne

South Dakota

All counties except:
Bennett
Bon Homme
Charles Mix
Clay
Douglas
Gregory
Hutchinson
Jackson
Mellette
Todd
Tripp
Union
Yankton

Utah

Box Elder
Cache
Carbon
Daggett
Duchesne
Morgan
Rich
Summit
Uintah
Wasatch

Vermont

All counties

Washington

Ferry
Okanogan
Pend Oreille
Stevens

Wisconsin

All counties except:
Ashland
Bayfield
Burnett
Douglas
Florence
Forest
Iron
Langlade
Lincoln
Oneida
Price
Sawyer
Taylor
Vilas
Washburn

Wyoming

All counties except:
Goshen
Platte
Lincoln
Sublette
Teton

Climate Zone 6 Recommendation Table for Grocery Stores

	Item	Component	Recommendation	How-to Tips	✓
Envelope	Roofs	Insulation entirely above deck	R-30.0 c.i.	EN1, 19–21	
		Metal building	R-25.0 + R-11.0 Ls	EN2, 19–21	
		SRI	No recommendation*	None	
	Walls	Mass (HC > 7 Btu/ft^2)	R-13.3 c.i.	EN4, 19–21	
		Metal building	R-0.0 + R-19.0 c.i.	EN5, 19–21	
		Steel framed	R-13.0 + R-12.5 c.i.	EN6, 19–21	
	Floors	Mass	R-16.7 c.i.	EN7, 19–21	
		Steel framed	R-38.0	EN8, 19–21	
	Slabs	Unheated	R-20 for 24 in.	EN9, 11, 19–21	
		Heated	R-20 for 48 in.	EN10–11, 19–21	
		Freezer box floors	See Table 5-2	EN12	
	Doors	Swinging	U-0.50	EN13, 20	
		Nonswinging	U-0.16	En14, 20	
		Vehicular/dock infiltration—door closed	0.28 cfm/ft^2 of door area	EN15, 20	
		Vehicular/dock infiltration—door open, truck in place	Weather seals for dock, levelers, trailer hinges	EN16, 20	
	Vestibules	All building entrances	Yes	EN17, 20	
	Continuous Air Barrier	Continuous air barrier	Entire building envelope	EN18	
	View Fenestration—All Orientations	Thermal transmittance	U-0.42	EN22–23, 26–28	
		SHGC	0.40	EN23, 26–28	
		VT/SHGC	>1.10	EN29	
Daylighting / Lighting	Daylighting	Skylights or rooftop monitors	Not required, follow recommendations when used	DL1–10	
		Daylight area	≥ 50% of the sales floor	DL1–10	
		Skylight to daylight area	Minimum = 3%, maximum = 5%	DL3	
		VT	Skylight VT ≥ 0.40	DL3	
		Skylight SHGC	0.35	EN25–27	
		Skylight thermal transmittance	U-0.20	DL4	
		Effective aperture	Skylight effective aperture ≥ 1%	DL8	
		Glazing material/diffuser	Measured haze value > 90% when tested according to ASTM D1003 (ASTM 2013)	DL3	
		Controls	General lighting auto control in response to daylight	DL6–7, 9–10	
	Interior Lighting	Ambient and accent lighting	Sales floor LPD = 1.15 W/ft^2	EL1, 3–7, 9–11, 17, 25, 30	
			Stock room LPD = 0.6 W/ft^2	EL1, 4, 9–11, 17–19, 29–30	
			Average of all other LPDs = 0.8 W/ft^2	EL1–2, 4, 8–11, 17–19, 25–27, 29–30	
		Light source lamp efficacy (mean lumens per watt)	LED—general = 80, accent = 75, case = 50 LED—exterior = 75 T8 and T5—greater than 2 ft = 92, 2 ft and less = 85	EL12, 14	
		T8 ballasts	Nondimming = NEMA Premium instant start Dimming = NEMA Premium program start	EL12	
		All T5/T5HO ballasts	Electronic program start	EL12	
		All CFL and HID ballasts	Electronic	EL12	
		Lighting controls	Sales floor = time switch—auto to 25% during closed hours, auto to 2% or less when unoccupied	EL16	
			Additional specialty lighting = auto ON only during store open hours	EL16	
			Stock room, restrooms = auto ON/OFF occupancy sensors	EL15	
			All other = manual ON, auto OFF occupancy sensors	EL15	
			After hours = maximum 2% of total building LPD	EL16	
	Exterior Lighting	Façade and landscape lighting	LPD = 0.075 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = auto OFF at business closing	EL33–35	
		Parking lots and drives	LPD = 0.08 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL31, 34–35	
		Walkways, plazas, and special feature areas	LPD = 0.16 W/ft^2 in LZ3 and LZ4, 0.14 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL32, 34–35	
		All other exterior lighting	No recommendation*	EL34–35	
Plug Loads	Plug Loads	Equipment	Specify ENERGY STAR Use laptops, mini desktops, thin clients LED for security lighting and signage	PL1–6, 8–9	
		Controls	Motion-sensing outlet and power strips, network computer control, timer switches	PL1, 7–8	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 6 Recommendation Table for Grocery Stores *(Continued)*

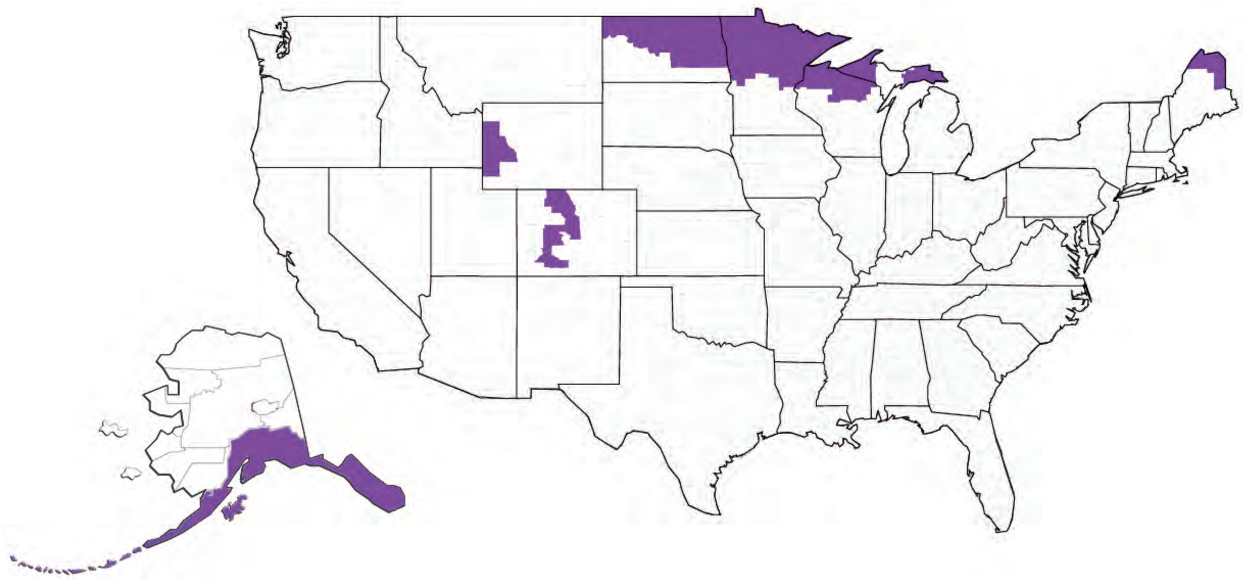
	Item	Component	Recommendation	How-to Tips	✓
Kitchen	Kitchen Equipment	Cooking equipment	ENERGY STAR or utility rebate-qualified equipment	KE1–2, 4, 6	
		Exhaust hoods	Side panels or end walls, larger overhangs, rear seal behind appliances, proximity hoods, DCKV	KE1, 3, 5, 6	
Refrigeration	Air-Cooled Condensers	Condenser sizing	Low temperature = 12°F TD Medium temperature = 18°F TD	RF1–2	
		Fin spacing	Maximum 10 FPI (excluding micro channel)	RF1	
		Specific efficiency	85 Btu/h·W at 10°F TD	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient dry-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Evaporative-Cooled Condensers	Condenser sizing	<68°F wb = 23°F TD 68°F to 75°F wb = 20°F TD >75°F wb = 16°F TD	RF1–2	
		Specific efficiency	180 Btu/h·W at 70°F wb and 100°F SCT	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient wet-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Compressor Systems	Suction groups/levels	Optimize suction groups based on load dynamics	RF8, 13–14	
		Indirect cooling design	CO_2 phase-change heat exchange Low temperature = two stage or cascade	RF9	
		Compressor group staging and capacity control	Continuously variable capacity control on each suction group	RF10	
		Suction group setpoint determination	Suction groups <25°F SCT = floating suction pressure logic	RF10	
		Mechanical subcooling	Low-temperature/medium-temperature SST ≥ 30°F	RF11	
	Display Cases and Walk-In Coolers/Freezers	Case type	Closed = dairy, beverage, deli, packaged salads, horizontal (tub), and low temperature Open = red meat, wet rack produce, specialty	RF15–16	
		Case door heater	Low temperature = 50–120 W/door with PWM controller Medium temperature = zero watt glass doors	RF17	
		Walk-in construction	Walk-in cooler insulation = R-25 Walk-in freezer insulation = R-40 Strip curtains on all doors	RF18, EN12	
		Walk-in doors	Doors < 48 in. = spring assist or cam-lift gravity hinge Doors > 48 in. = spring action door closer All doors = hydraulic door closer	RF19	
		Walk-in door switches and alarms	Freezer = fan and cooling OFF when door is open Cooler = fan and cooling OFF when door is open All = override and alarm integration	RF20	
		Walk-in box fan control	Two-speed fan control Reduce to 80% speed when load is below 50%	RF21	
		Lighting design	Open/closed cases ≤ 10 W/ft Meat cases ≤ 25 W/ft Display doors = motion sensors with 3 min time-out Walk-ins = vaportight LED with motion/vacancy sensor and/or door trigger switch	RF22, EL23, EL25	
		Defrost	Electric defrost = low-temperature cases, freezers, meat coolers; air defrost = all others Time initiation; temperature termination	RF23	
		Temperature and superheat control	Electronic modulating temperature control with floating suction pressure integration	RF24	
		Unit cooler	EC motor; select coils at 8°F TD on freezers Freezer fin spacing = 4 FPI Cooler fin spacing = no recommendation	RF25	
		Liquid-suction heat exchangers	High-efficiency heat exchange at piping exit; sized for additional suction superheat at design of 12°F for freezers and 6°F for coolers	RF26	
	Heat Recovery—HVAC	Heat recovery from refrigeration	Use at least 25% of the refrigeration design heat of rejection for space heating	RF27–31	
SHW	SWH	Gas water heater efficiency	See Table 5-13	WH1–4	
		Electric water heater efficiency	See Table 5-14	WH1–4	
		Electric heat pump water heater	COP 3.0 (interior heat source)	WH1–4	
		Recirculation pump control	Time clock or demand control	WH4	
		Pipe insulation (d < 1.5 in. / d ≥ 1.5 in.)	See Table 5-15	WH5	
		Refrigerant heat recovery	Superheat recovery using low-temperature system	WH6, RF27, KE4	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 6 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
HVAC	Distributed MA SZVAV DX Packaged RTU	Heating efficiency	Indirect gas heat = 80%	HV1–4, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4	
	Distributed MA SZVAV Chilled-Water RTU	Heating efficiency	High-efficiency condensing boiler = 90% Indirect gas heat = 80%	HV1–3, 5, 12	
		Cooling efficiency	Positive-displacement air-cooled chillers: <150 tons = 10.1 EER, 13.7 IPLV ≥150 tons = 10.1 EER, 14.0 IPLV	HV1–3, 5, 12	
		Fan motor efficiency	No recommendation*	HV1–3, 5, 24	
		Pump motor efficiency	No recommendation*	HV1–3, 5	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 5	
	Distributed SZVAV DX Packaged RTU with DOAS	Heating efficiency	Indirect gas heat = 80%	HV1–4, 6, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 6, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4, 6	
	Distributed SZCV, Air-Source HP Packaged RTU with DOAS	Heating efficiency	See Table 5-17	HV1–3, 7, 12	
		Cooling efficiency	See Table 5-17	HV1–3, 7, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 7	
	SZVAV WSHP Packaged RTU with DOAS	Heating efficiency	See Table 5-18	HV1–3, 8, 12	
		Cooling efficiency	See Table 5-18	HV1–3, 8, 12	
		Pump motor efficiency	As per Standard 90.1-2013, Table 10.8-2	HV1–3, 8	
		Fluid cooler fans	Variable-speed control	HV1–3, 8, 9, 24	
		High-efficiency boiler	90% E_c	HV1–3, 8	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 8	
	DOAS	Heating efficiency	See Table 5-19	HV10	
		Dehumidification efficiency	See Table 5-20	HV10	
		Moisture removal efficiency	See Table 5-20	HV10	
		Fan and motor	65% mechanical/motor efficiency in absence of whole-unit EER rating Motor efficiency as per Standard 90.1-2013, Table 10.8	HV10, 24	
		Maximum external static pressure	0.7 in. w.c.	HV10	
	Ventilation/Exhaust	Cooling capacity—economizers	≥ 54,000 Btu/h	HV17	
		DCV/performance-based ventilation	Control ventilation air based on pollutant concentrations in space (CO_2, VOCs, etc.)	HV18	
		Exhaust air recovery—in-store cooking with > Standard 62.1 area exhaust requirement	A (humid) zones = 75% total effectiveness B (dry) zones = 75% sensible effectiveness Capture minimum 80% available exhaust air, including general and bathroom, for energy recovery	HV13	
		Exhaust airflow control	Control based on occupancy using time clock or occupancy sensor	HV20	
	Ducts and Dampers	Outdoor air damper	Motorized damper	HV13	
		Duct seal class	Seal Class A	HV21, 23	
		Insulation level	R-6	HV22	
QA	Quality Assurance	Commissioning	Commission throughout design and construction	QA1–13	
		M&V	Employ post-occupancy monitoring	QA14–17	

*Note: Where the table says “No recommendation,” the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.



Zone 7

Alaska

Aleutians East
Aleutians West (CA)
Anchorage
Angoon (CA)
Bristol Bay
Denali
Haines
Juneau
Kenai Peninsula
Ketchikan (CA)
Ketchikan Gateway
Kodiak Island
Lake and Peninsula
Matanuska-Susitna
Prince of Wales-Outer
Sitka
Skagway-Hoonah-
Valdez-Cordova (CA)
Wrangell-Petersburg (CA)
Yakutat

Colorado

Clear Creek
Grand
Gunnison
Hinsdale
Jackson
Lake
Mineral
Park
Pitkin
Rio Grande
Routt
San Juan
Summit

Maine

Aroostook

Michigan

Baraga
Chippewa
Gogebic
Houghton
Iron
Keweenaw
Luce
Mackinac
Ontonagon
Schoolcraft

Minnesota

Aitkin
Becker
Beltrami
Carlton
Cass
Clay
Clearwater
Cook
Crow Wing
Grant
Hubbard
Itasca
Kanabec
Kittson
Koochiching
Lake
Lake of the Woods
Mahnomen
Marshall
Mille Lacs
Norman
Otter Tail
Pennington
Pine
Polk
Red Lake
Roseau
St. Louis
Wadena
Wilkin

North Dakota

Barnes
Benson
Bottineau
Burke
Cass
Cavalier
Divide
Eddy
Foster
Grand Forks
Griggs
Kidder
McHenry
McLean
Mountrail
Nelson
Pembina
Pierce
Ramsey
Renville
Rolette
Sheridan
Steele
Stutsman
Towner
Traill
Walsh
Ward
Wells
Williams

Wisconsin

Ashland
Bayfield
Burnett
Douglas
Florence
Forest
Iron
Langlade
Lincoln
Oneida
Price
Sawyer
Taylor
Vilas
Washburn

Wyoming

Lincoln
Sublette
Teton

Climate Zone 7 Recommendation Table for Grocery Stores

	Item	Component	Recommendation	How-to Tips	✓
Envelope	Roofs	Insulation entirely above deck	R-35.0 c.i.	EN1, 19–21	
		Metal building	R-30.0 + R-11.0 Ls	EN2, 19–21	
		SRI	No recommendation*	None	
	Walls	Mass (HC > 7 Btu/ft^2)	R- 15.2 c.i.	EN4, 19–21	
		Metal building	R-0.0 + R-22.1 c.i.	EN5, 19–21	
		Steel framed	R-13.0 + R-12.5 c.i.	EN6, 19–21	
	Floors	Mass	R-20.9 c.i.	EN7, 19–21	
		Steel framed	R-38.0	EN8, 19–21	
	Slabs	Unheated	R-20 for 24 in.	EN9, 11, 19–21	
		Heated	R-25 for 48 in.	EN10–11, 19–21	
		Freezer box floors	See Table 5-2	EN12	
	Doors	Swinging	U-0.50	EN13, 20	
		Nonswinging	U-0.16	En14, 20	
		Vehicular/dock infiltration—door closed	0.28 cfm/ft^2 of door area	EN15, 20	
		Vehicular/dock infiltration—door open, truck in place	Weather seals for dock, levelers, trailer hinges	EN16, 20	
	Vestibules	At building entrance	Yes	EN17, 20	
	Continuous Air Barrier	Continuous air barrier	Entire building envelope	EN18	
	View Fenestration—All Orientations	Thermal transmittance	U-0.38	EN22–23, 26–28	
		SHGC	0.45	EN23, 26–28	
		VT/SHGC	>1.10	EN29	
Daylighting / Lighting	Daylighting	Skylights or rooftop monitors	Not required, follow recommendations when used	DL1–10	
		Daylight area	≥ 50% of the sales floor	DL1–10	
		Skylight to daylight area	Minimum = 3%, maximum = 5%	DL3	
		VT	Skylight VT ≥ 0.40	DL3	
		Skylight SHGC	0.35	EN25–27	
		Skylight thermal transmittance	U-0.20	DL4	
		Effective aperture	Skylight effective aperture ≥ 1%	DL8	
		Glazing material/diffuser	Measured haze value > 90% when tested according to ASTM D1003 (ASTM 2013)	DL3	
		Controls	General lighting auto control in response to daylight	DL6–7, 9–10	
	Interior Lighting	Ambient and accent lighting	Sales floor LPD = 1.15 W/ft^2	EL1, 3–7, 9–11, 17, 25, 30	
			Stock room LPD = 0.6 W/ft^2	EL1, 4, 9–11, 17–19, 29–30	
			Average of all other LPDs = 0.8 W/ft^2	EL1–2, 4, 8–11, 17–19, 25–27, 29–30	
		Light source lamp efficacy (mean lumens per watt)	LED—general = 80, accent = 75, case = 50 LED—exterior = 75 T8 and T5—greater than 2 ft = 92, 2 ft and less = 85	EL12, 14	
		T8 ballasts	Nondimming = NEMA Premium instant start Dimming = NEMA Premium program start	EL12	
		All T5/T5HO ballasts	Electronic program start	EL12	
		All CFL and HID ballasts	Electronic	EL12	
		Lighting controls	Sales floor = time switch—auto to 25% during closed hours, auto to 2% or less when unoccupied	EL16	
			Additional specialty lighting = auto ON only during store open hours	EL16	
			Stock room, restrooms = auto ON/OFF occupancy sensors	EL15	
			All other = manual ON, auto OFF occupancy sensors	EL15	
			After hours = maximum 2% of total building LPD	EL16	
	Exterior Lighting	Façade and landscape lighting	LPD = 0.075 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = auto OFF at business closing	EL33–35	
		Parking lots and drives	LPD = 0.08 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL31, 34–35	
		Walkways, plazas, and special feature areas	LPD = 0.16 W/ft^2 in LZ3 and LZ4, 0.14 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL32, 34–35	
		All other exterior lighting	No recommendation*	EL34–35	
Plug Loads	Plug Loads	Equipment	Specify ENERGY STAR Use laptops, mini desktops, thin clients LED for security lighting and signage	PL1–6, 8–9	
		Controls	Motion-sensing outlet and power strips, network computer control, timer switches	PL1, 7–8	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 7 Recommendation Table for Grocery Stores *(Continued)*

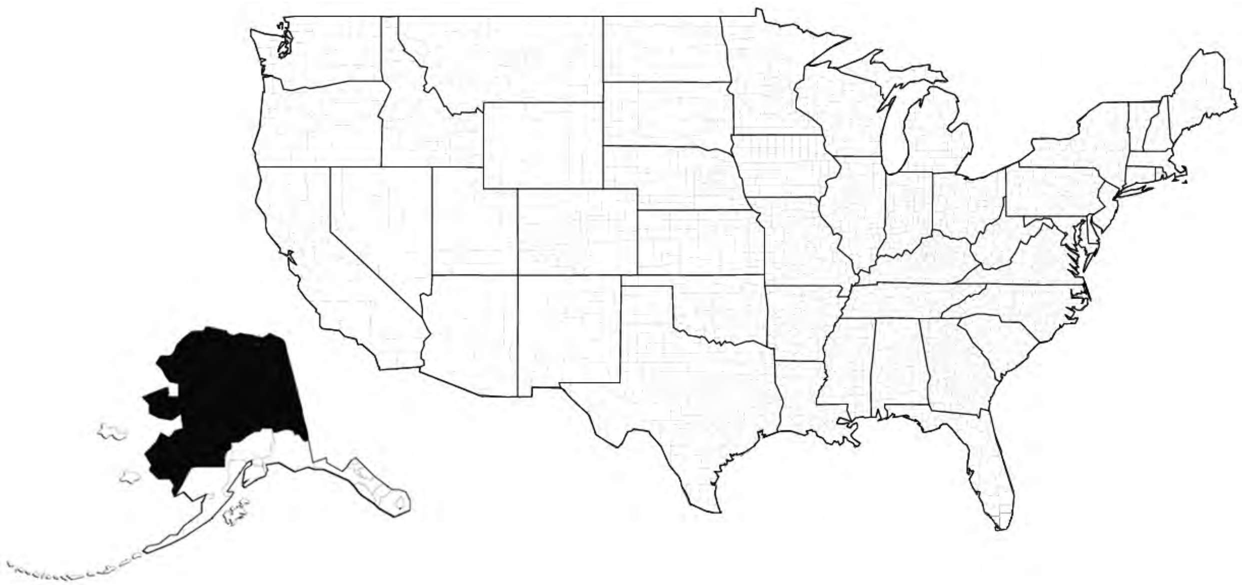
	Item	Component	Recommendation	How-to Tips	✓
Kitchen	Kitchen Equipment	Cooking equipment	ENERGY STAR or utility rebate-qualified equipment	KE1–2, 4, 6	
		Exhaust hoods	Side panels or end walls, larger overhangs, rear seal behind appliances, proximity hoods, DCKV	KE1, 3, 5, 6	
Refrigeration	Air-Cooled Condensers	Condenser sizing	Low temperature = 12°F TD Medium temperature = 18°F TD	RF1–2	
		Fin spacing	Maximum 10 FPI (excluding micro channel)	RF1	
		Specific efficiency	85 Btu/h·W at 10°F TD	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient dry-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Evaporative-Cooled Condensers	Condenser sizing	<68°F wb = 23°F TD 68°F to 75°F wb = 20°F TD >75°F wb = 16°F TD	RF1–2	
		Specific efficiency	180 Btu/h·W at 70°F wb and 100°F SCT	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient wet-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Compressor Systems	Suction groups/levels	Optimize suction groups based on load dynamics	RF8, 13–14	
		Indirect cooling design	CO_2 phase-change heat exchange Low temperature = two stage or cascade	RF9	
		Compressor group staging and capacity control	Continuously variable capacity control on each suction group	RF10	
		Suction group setpoint determination	Suction groups <25°F SCT = floating suction pressure logic	RF10	
		Mechanical subcooling	Low-temperature/medium-temperature SST ≥ 30°F	RF11	
	Display Cases and Walk-In Coolers/Freezers	Case type	Closed = dairy, beverage, deli, packaged salads, horizontal (tub), and low temperature Open = red meat, wet rack produce, specialty	RF15–16	
		Case door heater	Low temperature = 50–120 W/door with PWM controller Medium temperature = zero watt glass doors	RF17	
		Walk-in construction	Walk-in cooler insulation = R-25 Walk-in freezer insulation = R-40 Strip curtains on all doors	RF18, EN12	
		Walk-in doors	Doors < 48 in. = spring assist or cam-lift gravity hinge Doors > 48 in. = spring action door closer All doors = hydraulic door closer	RF19	
		Walk-in door switches and alarms	Freezer = fan and cooling OFF when door is open Cooler = fan and cooling OFF when door is open All = override and alarm integration	RF20	
		Walk-in box fan control	Two-speed fan control Reduce to 80% speed when load is below 50%	RF21	
		Lighting design	Open/closed cases ≤ 10 W/ft Meat cases ≤ 25 W/ft Display doors = motion sensors with 3 min time-out Walk-ins = vaportight LED with motion/vacancy sensor and/or door trigger switch	RF22, EL23, EL25	
		Defrost	Electric defrost = low-temperature cases, freezers, meat coolers; air defrost = all others Time initiation; temperature termination	RF23	
		Temperature and superheat control	Electronic modulating temperature control with floating suction pressure integration	RF24	
		Unit cooler	EC motor; select coils at 8°F TD on freezers Freezer fin spacing = 4 FPI Cooler fin spacing = no recommendation	RF25	
		Liquid-suction heat exchangers	High-efficiency heat exchange at piping exit; sized for additional suction superheat at design of 12°F for freezers and 6°F for coolers	RF26	
	Heat Recovery—HVAC	Heat recovery from refrigeration	Use at least 25% of the refrigeration design heat of rejection for space heating	RF27–31	
SHW	SWH	Gas water heater efficiency	See Table 5-13	WH1–4	
		Electric water heater efficiency	See Table 5-14	WH1–4	
		Electric heat pump water heater	COP 3.0 (interior heat source)	WH1–4	
		Recirculation pump control	Time clock or demand control	WH4	
		Pipe insulation (d < 1.5 in. / d ≥ 1.5 in.)	See Table 5-15	WH5	
		Refrigerant heat recovery	Superheat recovery using low-temperature system	WH6, RF27, KE4	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 7 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
HVAC	Distributed MA SZVAV DX Packaged RTU	Heating efficiency	Indirect gas heat = 80%	HV1–4, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4	
	Distributed MA SZVAV Chilled-Water RTU	Heating efficiency	High-efficiency condensing boiler = 90% Indirect gas heat = 80%	HV1–3, 5, 12	
		Cooling efficiency	Positive-displacement air-cooled chillers: <150 tons = 10.1 EER, 13.7 IPLV ≥150 tons = 10.1 EER, 14.0 IPLV	HV1–3, 5, 12	
		Fan motor efficiency	No recommendation*	HV1–3, 5, 24	
		Pump motor efficiency	No recommendation*	HV1–3, 5	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 5	
	Distributed SZVAV DX Packaged RTU with DOAS	Heating efficiency	Indirect gas heat = 80%	HV1–4, 6, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 6, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4, 6	
	Distributed SZCV Air-Source HP Packaged RTU with DOAS	Heating efficiency	See Table 5-17	HV1–3, 7, 12	
		Cooling efficiency	See Table 5-17	HV1–3, 7, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 7	
	SZVAV WSHP Packaged RTU with DOAS	Heating efficiency	See Table 5-18	HV1–3, 8, 12	
		Cooling efficiency	See Table 5-18	HV1–3, 8, 12	
		Pump motor efficiency	As per Standard 90.1-2013, Table 10.8-2	HV1–3, 8	
		Fluid cooler fans	Variable-speed control	HV1–3, 8, 9, 24	
		High-efficiency boiler	90% E_c	HV1–3, 8	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 8	
	DOAS	Heating efficiency	See Table 5-19	HV10	
		Dehumidification efficiency	See Table 5-20	HV10	
		Moisture removal efficiency	See Table 5-20	HV10	
		Fan and motor	65% mechanical/motor efficiency in absence of whole-unit EER rating Motor efficiency as per Standard 90.1-2013, Table 10.8	HV10, 24	
		Maximum external static pressure	0.7 in. w.c.	HV10	
	Ventilation/Exhaust	Cooling capacity—economizers	≥ 54,000 Btu/h	HV17	
		DCV/performance-based ventilation	Control ventilation air based on pollutant concentrations in space (CO_2, VOCs, etc.)	HV18	
		Exhaust air recovery—in-store cooking with > Standard 62.1 area exhaust requirement	A (humid) zones = 75% total effectiveness B (dry) zones = 75% sensible effectiveness Capture minimum 80% available exhaust air, including general and bathroom, for energy recovery	HV13	
		Exhaust airflow control	Control based on occupancy using time clock or occupancy sensor	HV20	
	Ducts and Dampers	Outdoor air damper	Motorized damper	HV13	
		Duct seal class	Seal Class A	HV21, 23	
		Insulation level	R-6	HV22	
QA	Quality Assurance	Commissioning	Commission throughout design and construction	QA1–13	
		M&V	Employ post-occupancy monitoring	QA14–17	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.



Zone 8

Alaska

Bethel (CA)
Dillingham (CA)
Fairbanks North Star
Nome (CA)
North Slope
Northwest Arctic
Southeast Fairbanks (CA)
Wade Hampton (CA)
Yukon-Koyukuk (CA)

Climate Zone 8 Recommendation Table for Grocery Stores

	Item	Component	Recommendation	How-to Tips	✓
Envelope	Roofs	Insulation entirely above deck	R-35.0 c.i.	EN1, 19–21	
		Metal building	R-25.0 + R-11.0 + R-11.0 Ls	EN2, 19–21	
		SRI	No recommendation*	None	
	Walls	Mass (HC > 7 Btu/ft^2)	R-19.0 c.i.	EN4, 19–21	
		Metal building	R-0.0 + R-25.0 c.i.	EN5, 19–21	
		Steel framed	R-13.0 + R-18.8 c.i.	EN6, 19–21	
	Floors	Mass	R-23.0 c.i.	EN7, 19–21	
		Steel framed	R-38.0	EN8, 19–21	
	Slabs	Unheated	R-20 for 48 in.	EN9, 11, 19–21	
		Heated	R-25 for 48 in.	EN10–11, 19–21	
		Freezer box floors	See Table 5-2	EN12	
	Doors	Swinging	U-0.50	EN13, 20	
		Nonswinging	U-0.16	En14, 20	
		Vehicular/dock infiltration—door closed	0.28 cfm/ft^2 of door area	EN15, 20	
		Vehicular/dock infiltration—door open, truck in place	Weather seals for dock, levelers, trailer hinges	EN16, 20	
	Vestibules	All building entrances	Yes	EN17, 20	
	Continuous Air Barrier	Continuous air barrier	Entire building envelope	EN18	
	View Fenestration—All Orientations	Thermal transmittance	U-0.38	EN22–23, 26–28	
		SHGC	0.45	EN23, 26–28	
		VT/SHGC	>1.10	EN29	
Daylighting / Lighting	Daylighting	Skylights or rooftop monitors	Not required, follow recommendations when used	DL1–10	
		Daylight area	≥ 50% of the sales floor	DL1–10	
		Skylight to daylight area	Minimum = 3%, maximum = 5%	DL3	
		VT	Skylight VT ≥ 0.40	DL3	
		Skylight SHGC	0.35	EN25–27	
		Skylight thermal transmittance	U-0.20	DL4	
		Effective aperture	Skylight effective aperture ≥ 1%	DL8	
		Glazing material/diffuser	Measured haze value > 90% when tested according to ASTM D1003 (ASTM 2013)	DL3	
		Controls	General lighting auto control in response to daylight	DL6–7, 9–10	
	Interior Lighting	Ambient and accent lighting	Sales floor LPD = 1.15 W/ft^2	EL1, 3–7, 9–11, 17, 25, 30	
			Stock room LPD = 0.6 W/ft^2	EL1, 4, 9–11, 17–19, 29–30	
			Average of all other LPDs = 0.8 W/ft^2	EL1–2, 4, 8–11, 17–19, 25–27, 29–30	
		Light source lamp efficacy (mean lumens per watt)	LED—general = 80, accent = 75, case = 50 LED—exterior = 75 T8 and T5—greater than 2 ft = 92, 2 ft and less = 85	EL12, 14	
		T8 ballasts	Nondimming = NEMA Premium instant start Dimming = NEMA Premium program start	EL12	
		All T5/T5HO ballasts	Electronic program start	EL12	
		All CFL and HID ballasts	Electronic	EL12	
		Lighting controls	Sales floor = time switch—auto to 25% during closed hours, auto to 2% or less when unoccupied	EL16	
			Additional specialty lighting = auto ON only during store open hours	EL16	
			Stock room, restrooms = auto ON/OFF occupancy sensors	EL15	
			All other = manual ON, auto OFF occupancy sensors	EL15	
			After hours = maximum 2% of total building LPD	EL16	
	Exterior Lighting	Façade and landscape lighting	LPD = 0.075 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = auto OFF at business closing	EL33–35	
		Parking lots and drives	LPD = 0.08 W/ft^2 in LZ3 and LZ4, 0.05 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL31, 34–35	
		Walkways, plazas, and special feature areas	LPD = 0.16 W/ft^2 in LZ3 and LZ4, 0.14 W/ft^2 in LZ2 Controls = occupancy sensor to at most 50% power	EL32, 34–35	
		All other exterior lighting	No recommendation*	EL34–35	
Plug Loads	Plug Loads	Equipment	Specify ENERGY STAR Use laptops, mini desktops, thin clients LED for security lighting and signage	PL1–6, 8–9	
		Controls	Motion-sensing outlet and power strips, network computer control, timer switches	PL1, 7–8	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 8 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
Kitchen	Kitchen Equipment	Cooking equipment	ENERGY STAR or utility rebate-qualified equipment	KE1–2, 4, 6	
		Exhaust hoods	Side panels or end walls, larger overhangs, rear seal behind appliances, proximity hoods, DCKV	KE1, 3, 5, 6	
Refrigeration	Air-Cooled Condensers	Condenser sizing	Low temperature = 12°F TD Medium temperature = 18°F TD	RF1–2	
		Fin spacing	Maximum 10 FPI (excluding micro channel)	RF1	
		Specific efficiency	85 Btu/h·W at 10°F TD	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient dry-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Evaporative-Cooled Condensers	Condenser sizing	<68°F wb = 23°F TD 68°F to 75°F wb = 20°F TD >75°F wb = 16°F TD	RF1–2	
		Specific efficiency	180 Btu/h·W at 70°F wb and 100°F SCT	RF3	
		Minimum condensing temperature	≤60°F SCT	RF4–5	
		Fan control	Variable-speed control with all fans in unison	RF4, 6	
		Control setpoint method	Variable setpoint using ambient wet-bulb temperature plus design TD; additional adjustment for optimum speed at average load	RF4, 7	
	Compressor Systems	Suction groups/levels	Optimize suction groups based on load dynamics	RF8, 13–14	
		Indirect cooling design	CO_2 phase-change heat exchange Low temperature = two stage or cascade	RF9	
		Compressor group staging and capacity control	Continuously variable capacity control on each suction group	RF10	
		Suction group setpoint determination	Suction groups <25°F SCT = floating suction pressure logic	RF10	
		Mechanical subcooling	Low-temperature/medium-temperature SST ≥ 30°F	RF11	
	Display Cases and Walk-In Coolers/Freezers	Case type	Closed = dairy, beverage, deli, packaged salads, horizontal (tub), and low temperature Open = red meat, wet rack produce, specialty	RF15–16	
		Case door heater	Low temperature = 50–120 W/door with PWM controller Medium temperature = zero watt glass doors	RF17	
		Walk-in construction	Walk-in cooler insulation = R-25 Walk-in freezer insulation = R-40 Strip curtains on all doors	RF18, EN12	
		Walk-in doors	Doors < 48 in. = spring assist or cam-lift gravity hinge Doors > 48 in. = spring action door closer All doors = hydraulic door closer	RF19	
		Walk-in door switches and alarms	Freezer = fan and cooling OFF when door is open Cooler = fan and cooling OFF when door is open All = override and alarm integration	RF20	
		Walk-in box fan control	Two-speed fan control Reduce to 80% speed when load is below 50%	RF21	
		Lighting design	Open/closed cases ≤ 10 W/ft Meat cases ≤ 25 W/ft Display doors = motion sensors with 3 min time-out Walk-ins = vaportight LED with motion/vacancy sensor and/or door trigger switch	RF22, EL23, EL25	
		Defrost	Electric defrost = low-temperature cases, freezers, meat coolers; air defrost = all others Time initiation; temperature termination	RF23	
		Temperature and superheat control	Electronic modulating temperature control with floating suction pressure integration	RF24	
		Unit cooler	EC motor; select coils at 8°F TD on freezers Freezer fin spacing = 4 FPI Cooler fin spacing = no recommendation	RF25	
		Liquid-suction heat exchangers	High-efficiency heat exchange at piping exit; sized for additional suction superheat at design of 12°F for freezers and 6°F for coolers	RF26	
	Heat Recovery—HVAC	Heat recovery from refrigeration	Use at least 25% of the refrigeration design heat of rejection for space heating	RF27–31	
SHW	SWH	Gas water heater efficiency	See Table 5-13	WH1–4	
		Electric water heater efficiency	See Table 5-14	WH1–4	
		Electric heat pump water heater	COP 3.0 (interior heat source)	WH1–4	
		Recirculation pump control	Time clock or demand control	WH4	
		Pipe insulation (*d* < 1.5 in. / *d* ≥ 1.5 in.)	See Table 5-15	WH5	
		Refrigerant heat recovery	Superheat recovery using low-temperature system	WH6, RF27, KE4	

*Note: Where the table says "No recommendation," the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

Climate Zone 8 Recommendation Table for Grocery Stores *(Continued)*

	Item	Component	Recommendation	How-to Tips	✓
HVAC	Distributed MA SZVAV DX Packaged RTU	Heating efficiency	Indirect gas heat = 80%	HV1–4, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4	
	Distributed MA SZVAV Chilled-Water RTU	Heating efficiency	High-efficiency condensing boiler = 90% Indirect gas heat = 80%	HV1–3, 5, 12	
		Cooling efficiency	Positive-displacement air-cooled chillers: <150 tons = 10.1 EER, 13.7 IPLV ≥150 tons = 10.1 EER, 14.0 IPLV	HV1–3, 5, 12	
		Fan motor efficiency	No recommendation*	HV1–3, 5, 24	
		Pump motor efficiency	No recommendation*	HV1–3, 5	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 5	
	Distributed SZVAV DX Packaged RTU with DOAS	Heating efficiency	Indirect gas heat = 80%	HV1–4, 6, 12	
		Cooling efficiency	See Table 5-16	HV1–4, 6, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–4, 6	
	Distributed SZCV Air-Source HP Packaged RTU with DOAS	Heating efficiency	See Table 5-17	HV1–3, 7, 12	
		Cooling efficiency	See Table 5-17	HV1–3, 7, 12	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 7	
	SZVAV WSHP Packaged RTU with DOAS	Heating efficiency	See Table 5-18	HV1–3, 8, 12	
		Cooling efficiency	See Table 5-18	HV1–3, 8, 12	
		Pump motor efficiency	As per Standard 90.1-2013, Table 10.8-2	HV1–3, 8	
		Fluid cooler fans	Variable-speed control	HV1–3, 8, 9, 24	
		High-efficiency boiler	90% E_c	HV1–3, 8	
		Maximum external static pressure	0.7 in. w.c.	HV1–3, 8	
	DOAS	Heating efficiency	See Table 5-19	HV10	
		Dehumidification efficiency	See Table 5-20	HV10	
		Moisture removal efficiency	See Table 5-20	HV10	
		Fan and motor	65% mechanical/motor efficiency in absence of whole-unit EER rating Motor efficiency as per Standard 90.1-2013, Table 10.8	HV10, 24	
		Maximum external static pressure	0.7 in. w.c.	HV10	
	Ventilation/Exhaust	Cooling capacity—economizers	≥ 54,000 Btu/h	HV17	
		DCV/performance-based ventilation	Control ventilation air based on pollutant concentrations in space (CO_2, VOCs, etc.)	HV18	
		Exhaust air recovery—in-store cooking with > Standard 62.1 area exhaust requirement	A (humid) zones = 75% total effectiveness B (dry) zones = 75% sensible effectiveness Capture minimum 80% available exhaust air, including general and bathroom, for energy recovery	HV13	
		Exhaust airflow control	Control based on occupancy using time clock or occupancy sensor	HV20	
	Ducts and Dampers	Outdoor air damper	Motorized damper	HV13	
		Duct seal class	Seal Class A	HV21, 23	
		Insulation level	R-6	HV22	
QA	Quality Assurance	Commissioning	Commission throughout design and construction	QA1–13	
		M&V	Employ post-occupancy monitoring	QA14–17	

*Note: Where the table says “No recommendation,” the user must meet the more stringent of either the most recent edition of ASHRAE/IES Standard 90.1 or the local code requirements.

REFERENCES

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How to Implement Recommendations

5

Recommendations for energy-saving measures for each climate zone are contained in the individual tables in Chapter 4. The following how-to tips are intended to provide guidance on good practices for implementing the recommendations as well as cautions to avoid known problems and obstacles to energy-efficient construction.

ENVELOPE

OPAQUE ENVELOPE COMPONENTS

Good Design Practice

EN1 ***Roofs—Insulation Entirely above Deck*** **(Climate Zones: all)**

The insulation entirely above deck should be continuous insulation (c.i.) rigid boards. Continuous insulation is important because no framing members are present that would introduce thermal bridges or short circuits to bypass the insulation. When two layers of c.i. are used in this construction, the board edges should be staggered to reduce the potential for convection losses or thermal bridging. If an inverted or protected membrane roof system is used, at least one layer of insulation is placed above the membrane and a maximum of one layer is placed beneath the membrane.

EN2 ***Roofs—Metal Buildings*** **(Climate Zones: all)**

Metal buildings pose particular challenges in the pursuit of designing and constructing advanced buildings. The metal skin and purlin/girt connection, even with compressed fiberglass between them, is highly conductive, which limits the effectiveness of the insulation. A purlin is a horizontal structural member that supports the roof covering. In metal building construction, this is typically a z-shaped cold-formed steel member; but a steel bar or open web joists can be used for longer spans.

The thermal performance of metal building roofs with fiberglass batts is improved by treating the thermal bridging associated with fasteners. Use of foam blocks is a proven technique to reduce the thermal bridging. Thermal blocks, with minimum dimensions of 1 × 3 in., should be R-5 rigid insulation installed parallel to the purlins (see Figure 5-1).

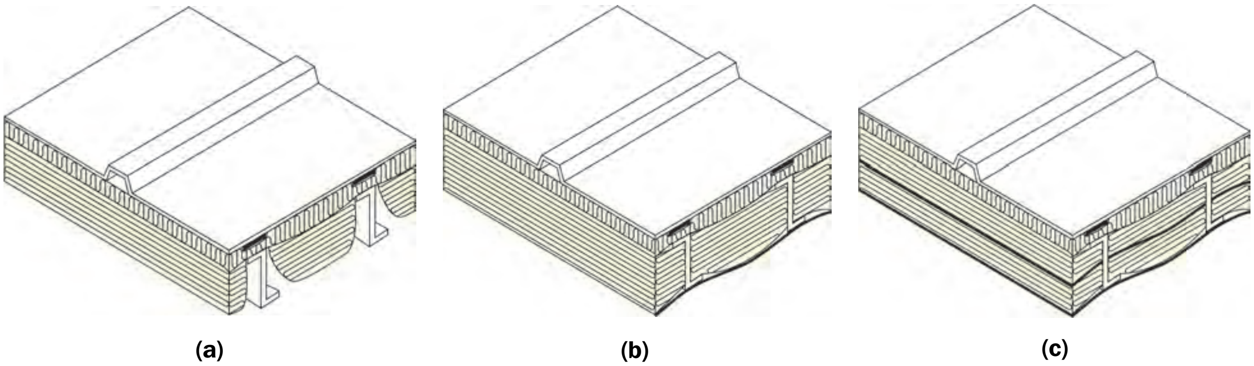


Figure 5-1 (EN2) Prefabricated Metal Roofs Showing Thermal Blocking of Purlins—(a) Filled Cavity; (b) Liner System, Two Layers; and (c) Liner System, Three Layers

Thermal blocks can be used successfully with standing seam roofs that use concealed clips of varying heights to accommodate the block. However, a thermal block cannot be used with a through-fastened roof that is screwed directly to the purlins because it diminishes the structural load carrying capacity by "softening" the connection and restraint provided to the purlin by the roof.

In climate zones 1 through 3, the recommended construction is a filled cavity that has the first layer of insulation parallel to and between the purlins and the second layer of insulation perpendicular to and over the top of the purlins (see Figure 5-1a).

In climate zones 4 through 7, the recommended construction is a liner system that has the first layer of insulation parallel to and between the purlins and the second layer of R-11 insulation perpendicular to and over the top of the purlins (see Figure 5-1b).

In climate zone 8, the recommended construction is a liner system with the first layer of insulation, R-25, and second layer of insulation, R-11, parallel to and between the purlins and the third layer of insulation, R-11, perpendicular to and over the top of the purlins (see Figure 5-1c).

Rigid c.i. can be added to provide additional insulation if required to meet the U-factors listed in Appendix A. In any case, rigid c.i. or other high-performance insulation systems may be used provided the total roof assembly has a U-factor that is less than or equal to the appropriate climate zone construction listed in Appendix A.

EN3 *Cool Roofs* (Climate Zones: ❶ ❷ ❸)

A cool roof with a Solar Reflectance Index (SRI) of 78 or higher is recommended. A high reflectance keeps much of the sun's energy from being absorbed, while a high thermal emissivity surface radiates away any solar energy that is absorbed, allowing the roof to cool more rapidly. Cool roofs are typically white and have a smooth surface. Commercial roof products that qualify as cool roofs fall into three categories: single ply, liquid applied, and metal panels. Examples are presented in Table 5-1.

The solar reflectance and thermal emissivity property values represent initial conditions as determined by a laboratory accredited by the Cool Roof Rating Council (CRRC). An SRI can be determined by the following equation:

$$\mathrm{SRI} = 123.97 - 141.35(\chi) + 9.655(\chi^2)$$

where

$$\chi = \frac{20.797 \times \alpha - 0.603 \times \varepsilon}{9.5205 \times \varepsilon + 12.0}$$

α = solar absorptance = 1 – solar reflectance

ε = thermal emissivity

These equations were derived from ASTM E1980 (ASTM 2011) assuming a medium wind speed. Note that cool roofs are not a substitute for the appropriate amount of insulation.

Table 5-1 Examples of Cool Roofs

Category	Product	Reflectance	Emissivity	SRI
Single ply	White polyvinyl chloride (PVC)	0.86	0.86	107
	White chlorinated polyethylene (CPE)	0.86	0.88	108
	White chlorosulfonated polyethylene (CPSE)	0.85	0.87	106
	White thermoplastic polyolefin (TSO)	0.77	0.87	95
Liquid applied	White elastomeric, polyurethane, acrylic coating	0.71	0.86	86
	White paint (on metal or concrete)	0.71	0.85	86
Metal panels	Factory-coated white finish	0.90	0.87	113

EN4 *Walls—Mass* (Climate Zones: all)

Mass walls are defined as those with a heat capacity (HC) exceeding 7 Btu/ft^2·°F. Insulation may be placed on the inside or the outside or sandwiched in the middle (tilt-up) of the mass. When insulation is placed on the exterior of the mass, rigid c.i. is recommended. When insulation is placed on the interior of the mass, a furring or framing system may be used, provided the total wall assembly has a U-factor that is less than or equal to the appropriate climate zone construction listed in Appendix A.

The greatest advantages of a mass wall can be obtained when insulation is placed on its exterior. In this case, the mass absorbs heat from the interior spaces that is later released in the evenings when the buildings are not occupied. The thermal mass of a building (typically contained in the building envelope) absorbs heat during the day and reduces the magnitude of indoor air temperature swings, reduces peak cooling loads, and transfers some of the absorbed heat into the night hours. The cooling load can then be covered by passive cooling techniques (natural ventilation) when the outdoor conditions are more favorable. An unoccupied building can also be precooled during the night by natural or mechanical ventilation to reduce the cooling energy use. This same effect reduces heating load as well. It is important to integrate the operation of the thermostat control strategies to maximize the thermal benefits (see HV16 and HV26).

Thermal mass also has a positive effect on thermal comfort. High-mass buildings attenuate interior air and wall temperature variations and sustain a stable overall thermal environment. This increases thermal comfort, particularly during mild seasons (spring and fall), during large air temperature changes (high solar gain), and in areas with large day/night temperature swings.

A designer should keep in mind that the occupant will be the final determinant on the extent of the usability of any building system, including thermal mass. Changing the use of internal spaces and surfaces can drastically reduce the effectiveness of thermal storage. The final use of the space must be considered when making the heating and cooling load calculations and incorporating possible energy savings from thermal mass effects (see HV1).

EN5 *Walls—Metal Building* (Climate Zones: all)

In all climate zones, rigid c.i. on the exterior of the girts is the recommendation. Alternative constructions are allowed provided the total wall assembly has a U-factor that is less than or equal to the appropriate climate zone construction listed in Appendix A.

If a single layer of faced fiberglass batt insulation is proposed, the insulation is installed continuously perpendicular to the exterior of the girts and is compressed as the metal panel is attached to the girts. If a layer of faced fiberglass batt insulation and a layer of rigid board insulation are proposed, the layer of faced fiberglass is installed continuously perpendicular to the exterior of the girts and is compressed as the rigid board insulation is installed continuously and perpendicular then attached to the girts from the exterior (on top of the fiberglass). The metal panels are then attached over the rigid board insulation using screws that penetrate through the insulation assembly into the girts.

EN6 *Walls—Steel Framed* (Climate Zones: all)

Cold-formed steel framing members are thermal bridges to the cavity insulation. Adding exterior foam sheathing as c.i. is the preferred method to upgrade the wall thermal performance because it increases the overall wall thermal performance and tends to minimize the impact of the thermal bridging.

Alternative combinations of cavity insulation and sheathing in thicker steel-framed walls can be used, provided that the proposed total wall assembly has a U-factor that is less than or equal to the U-factor for the appropriate climate zone construction listed in Appendix A. Batt insulation installed in cold-formed steel-framed wall assemblies is to be ordered as “full width batts,” and installation is normally by friction fit. Batt insulation should fill the entire cavity and not be cut short.

EN7 *Floors—Mass* (Climate Zones: all)

Insulation should be continuous and either integral to or above the slab. This can be achieved by placing high-density extruded polystyrene above the slab with either plywood or a thin layer of concrete on top. Placing insulation below the deck is not recommended due to losses through any concrete support columns or through the slab perimeter.

Exception: Buildings or zones within buildings that have durable floors for heavy machinery or equipment could place insulation below the deck.

EN8 *Floors—Metal Joist or Steel Framed* (Climate Zones: all)

Insulation should be installed parallel to the framing members and in intimate contact with the flooring system supported by the framing member in order to avoid the potential thermal short-circuiting associated with open or exposed air spaces. Nonrigid insulation should be supported from below, no less frequently than 24 in. on center.

EN9 *Slab-on-Grade Floors, Unheated* (Climate Zones: 4 5 6 7 8)

Rigid c.i. should be used around the perimeter of the slab and should reach the depth listed in the recommendation or to the bottom of the footing, whichever is less.

EN10 *Slab-on-Grade Floors, Heated* (Climate Zones: all)

Rigid c.i. should be used around the perimeter of the slab and should reach to the depth listed or to the frost line, whichever is deeper. Additionally, in climate zone 8, c.i. should be placed below the slab as well.

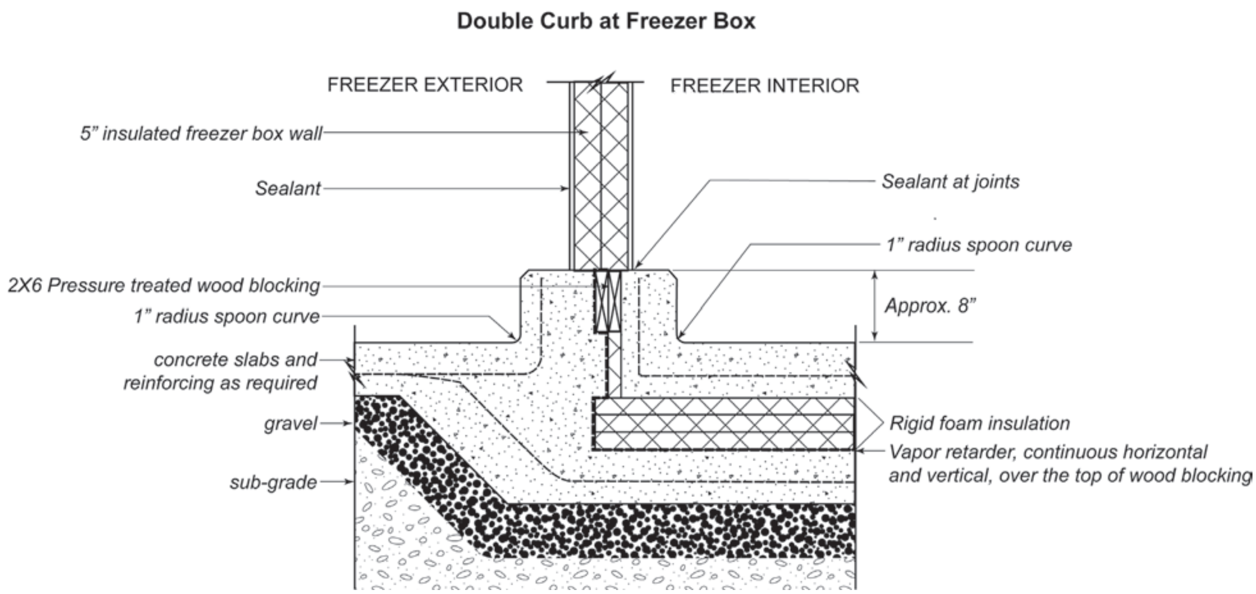
EN11 *Slab Edge Insulation* (Climate Zones: all)

Use of slab edge insulation improves thermal performance, but problems can occur in regions that have termites.

EN12 *Freezer Box Floors* (Climate Zones: all)

Site-built freezer boxes need to be constructed so as to prevent moisture problems due to thermal bridging at the intersection of the slab and wall. Figure 5-2 presents construction details to avoid the moisture problem: Figure 5-2a illustrates how to avoid moisture problems with a double curb at the freezer box while Figure 5-2b provides a comparable solution without the curb.

The insulation thickness below the slab of the freezer box is determined by the temperature within the freezer box. These values are shown in Table 5-2. *ASHRAE Handbook—Refrigeration* (ASHRAE 2010) further recommends that the floor insulation have a heating system installed to avoid underfloor ice formation, which can lead to heaving of the floor.



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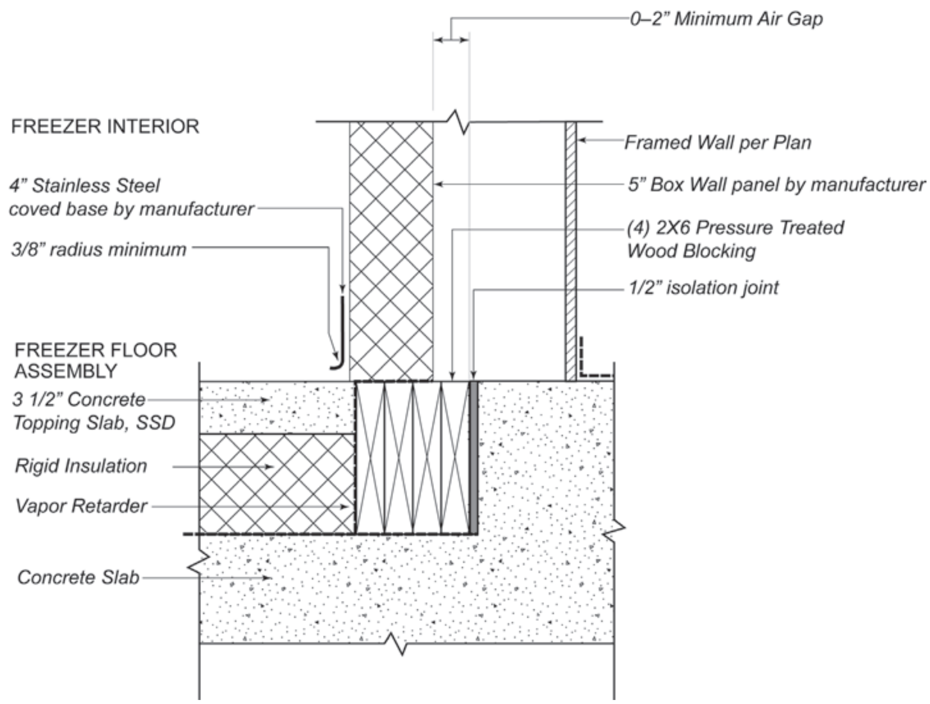


(b)

Figure 5-2 (EN12) a) Freezer Slab Edge Details with Double Curb and b) Freezer Slab Edge Details without Curb

Table 5-2 Freezer Box Floor Insulation Levels

Temperature, °F	R-Value
35°	23
25°	25
10°	29
0°	31
–10°	33
–20°	36
–30°	38
–40°	40
–50°	43

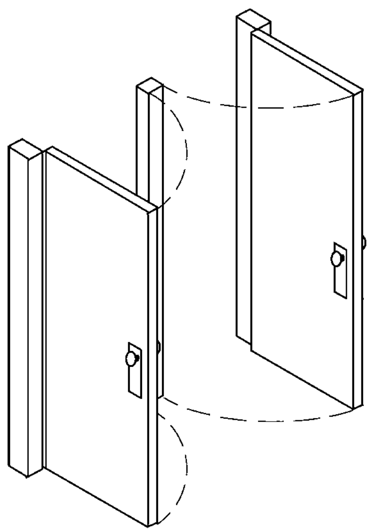


Figure 5-3 (EN13) Swinging Doors—Opaque Doors with Hinges on One Side, Closing to a Center Post

EN13 Doors—Opaque, Swinging (Climate Zones: all)

A U-factor of 0.7 corresponds to a double-panel metal door. A U-factor of 0.5 corresponds to an insulated double-panel metal door. If at all possible, single swinging doors should be used. Double swinging doors are difficult to seal at the center of the doors unless there is a center post (see Figure 5-3). Double swinging doors without a center post should be minimized and limited to areas where width is important. Vestibules can be added to further improve the energy efficiency.

EN14 Doors—Opaque, Nonswinging (Climate Zones: all)

Nonswinging opaque doors are recommended to have a minimum R-14 rigid insulation by determination in accordance with DASMA TDS #163 (DASMA 2008) or meet the recommended U-factor by determination in accordance with either NFRC 100 (NFRC 2014) or DASMA 105 (DASMA 2004). Doors that have solar exposure should be painted with a reflective paint (or high emissivity) and/or should be shaded. Metal doors often have poor emissivity and therefore collect heat that is transmitted through even the best insulated door, causing cooling loads and thermal comfort issues in the space.

EN15 Doors—Vehicular/Dock Opaque—Infiltration with Door Closed (Climate Zones: all)

Air leakage associated with vehicular/dock doors in the closed position is reduced with weather seals on doors that minimize air infiltration to less than 0.28 cfm/ft^2 of door area and that should be tested to a pressure of 1.57 lb/ft^2 in accordance with either ASTM E283 (ASTM 2012) or DASMA 105 (DASMA 2004). Dock seals should conform closely to the sides, top, and bottom of the door to minimize area of opening for infiltration. Insulated sectional or sliding doors can provide a tighter seal to minimize infiltration.

EN16 Doors—Vehicular/Dock Opaque—Infiltration with Door Open, Truck in Place (Climate Zones: all)

Infiltration through loading dock doors when they are open for truck loading or unloading can result in significant energy consumption. ASHRAE/IES Standard 90.1 (ASHRAE 2013a) requires that loading dock doors be equipped with weather seals to restrict infiltration when the doors are open and trailers are in place. Dock seals or shelters should conform closely to the sides and top of the trailer to minimize area of opening for infiltration. Two additional apertures, the dock leveler operating clearance and the hinge gap of the trailer doors, are not covered by this standard. Dock levelers should be furnished with brush-type seals to reduce the effective leakage crack width of the operating clearance from approximately 1.125 to less than 0.25 in. Inflatable or foam-type hinge seals should be utilized to minimize infiltration through this gap.

EN17 Vestibules (Climate Zones: all)

Vestibules are recommended for building entrances routinely used by occupants, not for dedicated emergency exits, maintenance doors, loading docks, or any other specialty entrances. Vestibules are to be designed so that in passing through the vestibule it is not necessary for the interior and exterior doors to open at the same time. Sensors that automatically control the opening of the doors should be calibrated such that the interior and exterior doors do not open simultaneously but only in response to the flow of the pedestrian traffic.

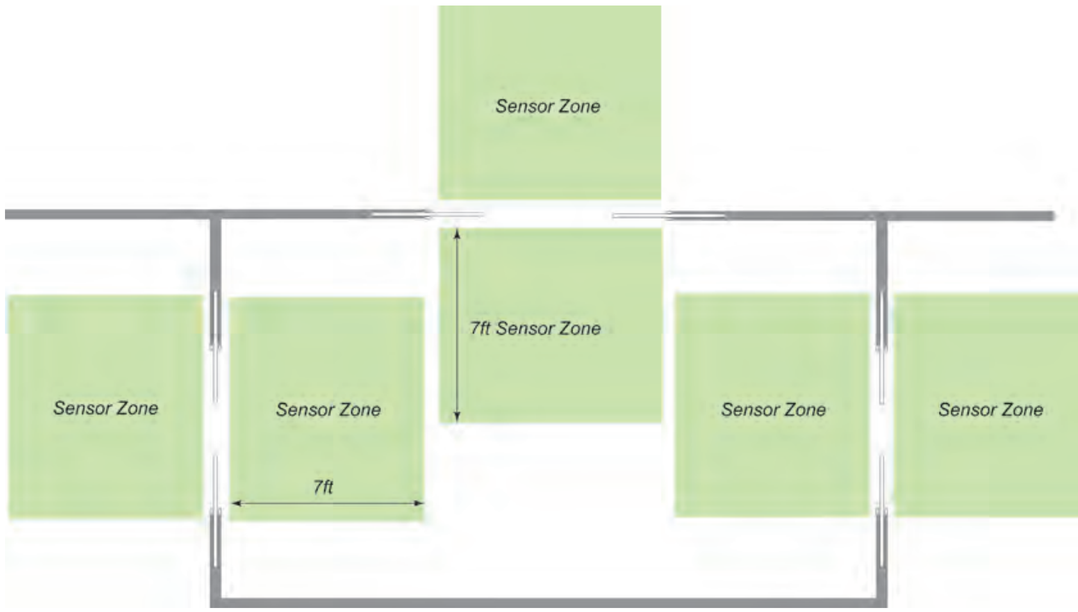
Interior and exterior doors should have a minimum distance between them such that sensor zones do not overlap, as shown in Figure 5-4. Interior and exterior doors should have a minimum distance between them of not less than 7 ft when the doors opening into and out of the vestibule are equipped with self-closing devices.

Vestibules are to be designed as areas only for traversing the exterior and the interior of the building. The vestibule should be a semi-heated space and not mechanically heated to above 45°F. The interior and exterior envelope that isolates the semi-heated space should comply with the requirements for a conditioned space.

Pedestrian traffic flows have a direct influence on the airflow patterns that impact the building energy use. Walkways that have 90-degree turns are the preferred geometry (Figure 5-4a). Straight-through entrance and 90-degree exit walkways are the next best geometry (Figure 5-4b). Offset walkways (Figure 5-4c) are better than straight-through walkways (Figure 5-4d), which are the least energy efficient.

EN18 Air Infiltration Control (Climate Zones: all)

The building envelope should be designed and constructed with a continuous air barrier system to control air leakage into or out of the conditioned space and should extend over all surfaces of the building envelope (at the lowest floor, exterior walls, and ceiling or roof). An air barrier system should also be provided for interior separations between conditioned space and space designed to maintain temperature or humidity levels that differ from those in the conditioned space by more than 50% of the difference between the conditioned space and design ambient conditions. If possible, a blower door should be used to depressurize the building to



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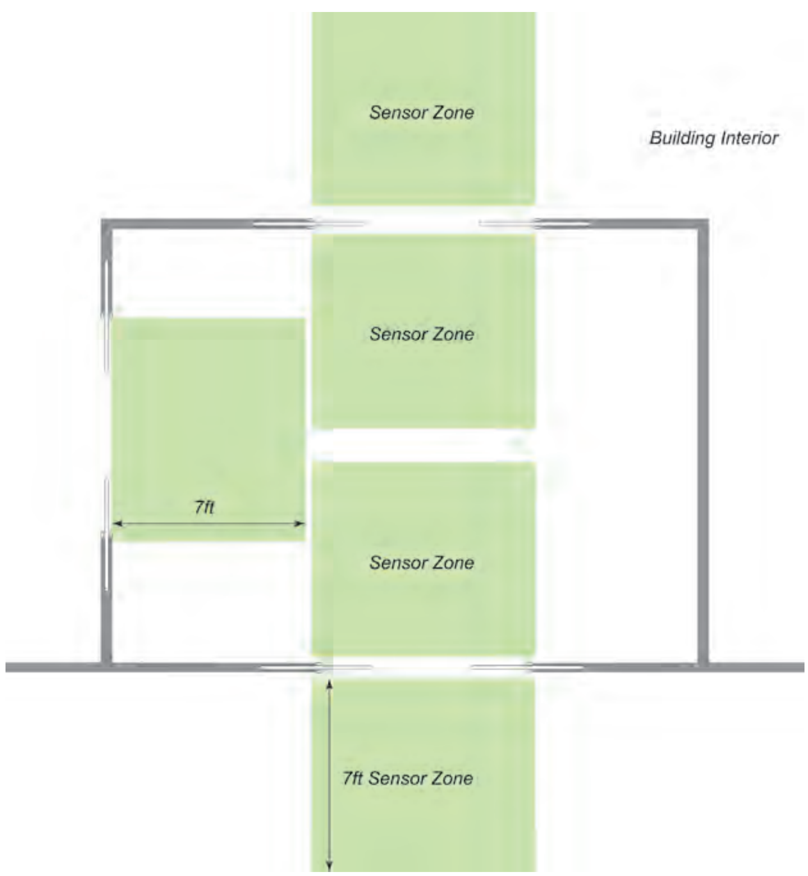
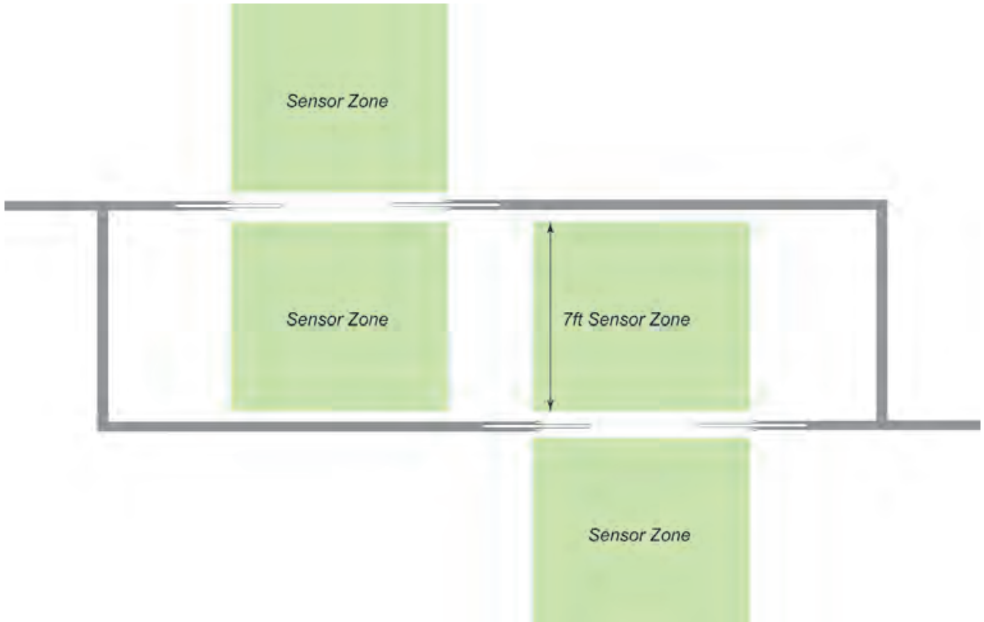


(b)

Figure 5-4 (EN 17) (a) 90-Degree Walkway Vestibule Configuration, (b) Straight-Through Entrance and 90-Degree Exit Walkway Vestibule Configuration



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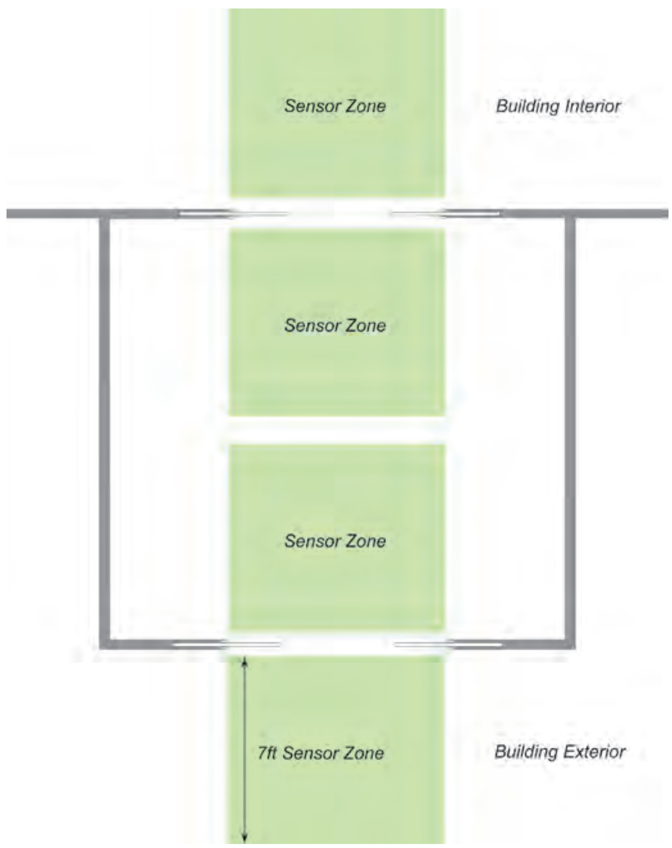


(d)

Figure 5-4 (EN 17) (c) Offset Walkway Vestibule Configuration and (d) Straight-Through Walkway Vestibule Configuration

find leaks in the infiltration barrier. At a minimum, the air barrier system should have the following characteristics:

- It should be continuous, with all joints made airtight.
- Air barrier materials used in frame walls should have an air permeability not to exceed 0.004 cfm/ft^2 under a pressure differential of 0.3 in. w.c. (1.57 lb/ft^2) when tested in accordance with ASTM E2178 (ASTM 2013).
- The system should be able to withstand positive and negative combined design wind, fan, and stack pressures on the envelope without damage or displacement and should transfer the load to the structure. It should not displace adjacent materials under full load.
- It should be durable or maintainable.
- The air barrier material of an envelope assembly should be joined in an airtight and flexible manner to the air barrier material of adjacent assemblies, allowing for the relative movement of these assemblies and components due to thermal and moisture variations, creep, and structural deflection.
- Connections should be made between
  - foundation and walls;
  - walls and windows or doors;
  - different wall systems;
  - wall and roof;
  - wall and roof over unconditioned space;
  - walls, floors, and roof across construction, control, and expansion joints; and
  - walls, floors, and roof to utility, pipe, and duct penetrations.
- All penetrations of the air barrier system and paths of air infiltration/exfiltration should be made airtight by appropriate methods including but not limited to gaskets, caulking, expanded foams, and tapes.

Options

EN19 *Alternative Constructions* (Climate Zones: all)

The climate zone recommendations provide only one solution for upgrading the thermal performance of the envelope. Other constructions can be equally effective but are not included in this Guide. Any alternative construction that is less than or equal to the U-factor, C-factor, or F-factor presented in Appendix A for the appropriate climate zone construction is equally acceptable. U-factors, C-factors, and F-factors that correspond to all the recommendations are presented in Appendix A.

Procedures to calculate U-factors and C-factors are presented in Chapters 18, 25, 26, and 27 of *ASHRAE Handbook—Fundamentals* (ASHRAE 2013b), and expanded U-factor, C-factor, and F-factor tables are presented in ASHRAE/IES Standard 90.1, Appendix A (ASHRAE 2013a).

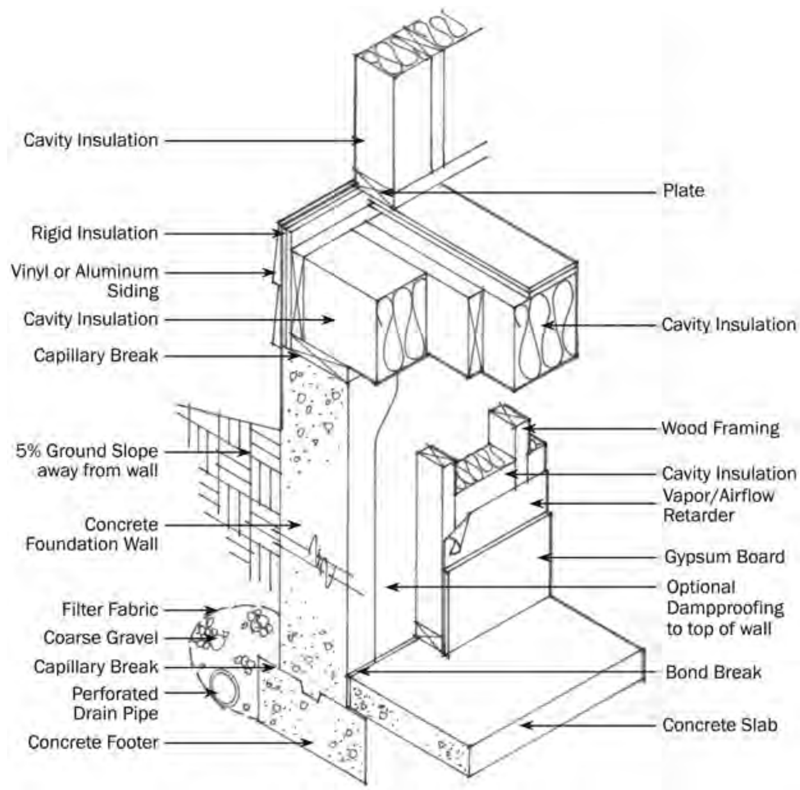
Cautions

The design of building envelopes for durability, indoor environmental quality, and energy conservation should not create conditions of accelerated deterioration or reduced thermal performance or problems associated with moisture, air infiltration, or termites.

The following cautions should be incorporated into the design and construction of the building.

EN20 *Moisture Control* (Climate Zones: all)

Building envelope assemblies should be designed to prevent wetting, high moisture content, liquid water intrusion, and condensation caused by diffusion of water vapor. See *ASHRAE Handbook—Fundamentals*, Chapter 25 (ASHRAE 2013b), for additional information. Figure 5-5a shows moisture control for wood framing that would be typical for mixed climates, and Figure 5-5b shows moisture control for concrete slabs that would be typical for warm, humid climates.



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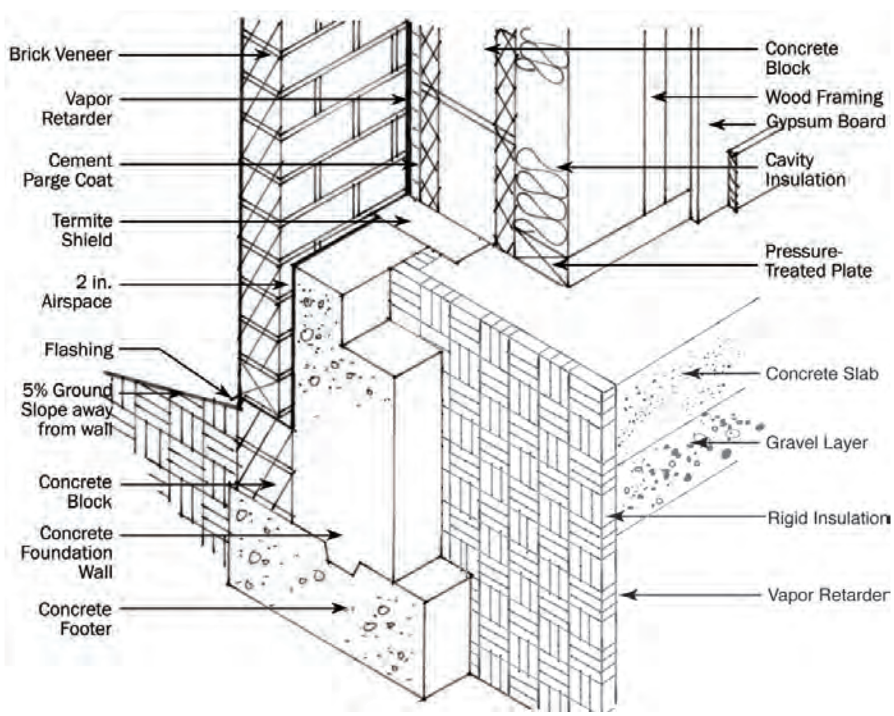


(b)

Figure 5-5 (EN20) Moisture Control for (a) Wood Framing and (b) Concrete Slabs

EN21 *Thermal Bridging—Opaque Components* (Climate Zones: all)

Thermal bridging in opaque components occurs when continuous conductive elements connect internal and external surfaces. The adverse effects of thermal bridging are most notable in cold climates where frost can develop on internal surfaces and lead to water droplets when the indoor temperature increases. The solution to thermal bridging is to provide thermal breaks or continuous insulation. Common problem areas are parapets, foundations, and penetrations of insulation.

The thermal bridge at parapets is shown in Figure 5-6a. The problem is that a portion of the wall construction is extended to create a parapet that extends above the roof to ensure worker safety per local code requirements. Since the wall insulation is on the outer face of the structure, it does not naturally connect to the insulation at the roof structure. To correct this, wrap the parapet with c.i. in the appropriate locations as shown in Figure 5-6b or use a structural solution in which an independent parapet structure periodically penetrates the roof insulation line to limit the thermal bridging effects.

Thermal bridges in foundations are shown in Figure 5-7a. This issue usually occurs because of construction sequences for the installation of below-grade works early in the design process. It is often an oversight to complete the connection between the below-grade and above-grade thermal protection because the installations of these elements are separate both in discipline and in time period on site. Design and constructing teams must make it clear that action to establish thermal continuity of the insulation line is a performance requirement of both parties, in order to achieve a typical solution as shown in Figure 5-7b. The insulation above grade needs to be protected with a surface or coating that is weather resistant and abuse tolerant.

Penetrations of insulation in which metal structural members must protrude from the building to support an external shade or construction (balcony, signage, etc.) need to be insulated. In these cases, the insulation should wrap the protruding metal piece when it is within the indoor cavity, and an additional length of insulation should be provided on its connection in each direction to prevent excessive heat transfer from the metal into the internal wall cavity. It should be noted that a façade consultant can model these types of situations to advise on the various lengths and thicknesses of insulation that would be needed to limit adverse impacts from condensation within the wall cavity.

VERTICAL FENESTRATION

Good Design Practice

EN22 *Vertical Fenestration* (Climate Zones: all)

Fenestration refers to the light-transmitting areas of a wall or roof, mainly windows and skylights but also including glass doors, glass block walls, and translucent plastic panels. Vertical fenestration includes sloped glazing if it has a slope equal to or more than 60° from horizontal. If it slopes less than 60° from horizontal, the fenestration falls into the skylight category. This means clerestories, roof monitors, and other such fenestration fall into the vertical category.

The recommendations for vertical fenestration are listed in Chapter 3 by climate zone. To be useful and consistent, the U-factors for windows should be measured over the entire window assembly, not just the center of glass. Look for a label that denotes the window rating is certified by the National Fenestration Rating Council (NFRC). The selection of high-performance window products should be considered separately for each orientation of the building and for daylighting and viewing functions.

To meet the solar heat gain coefficient (SHGC) recommendations for vertical fenestration in Chapter 4, use the SHGC multipliers for permanent projections as provided in ASHRAE/

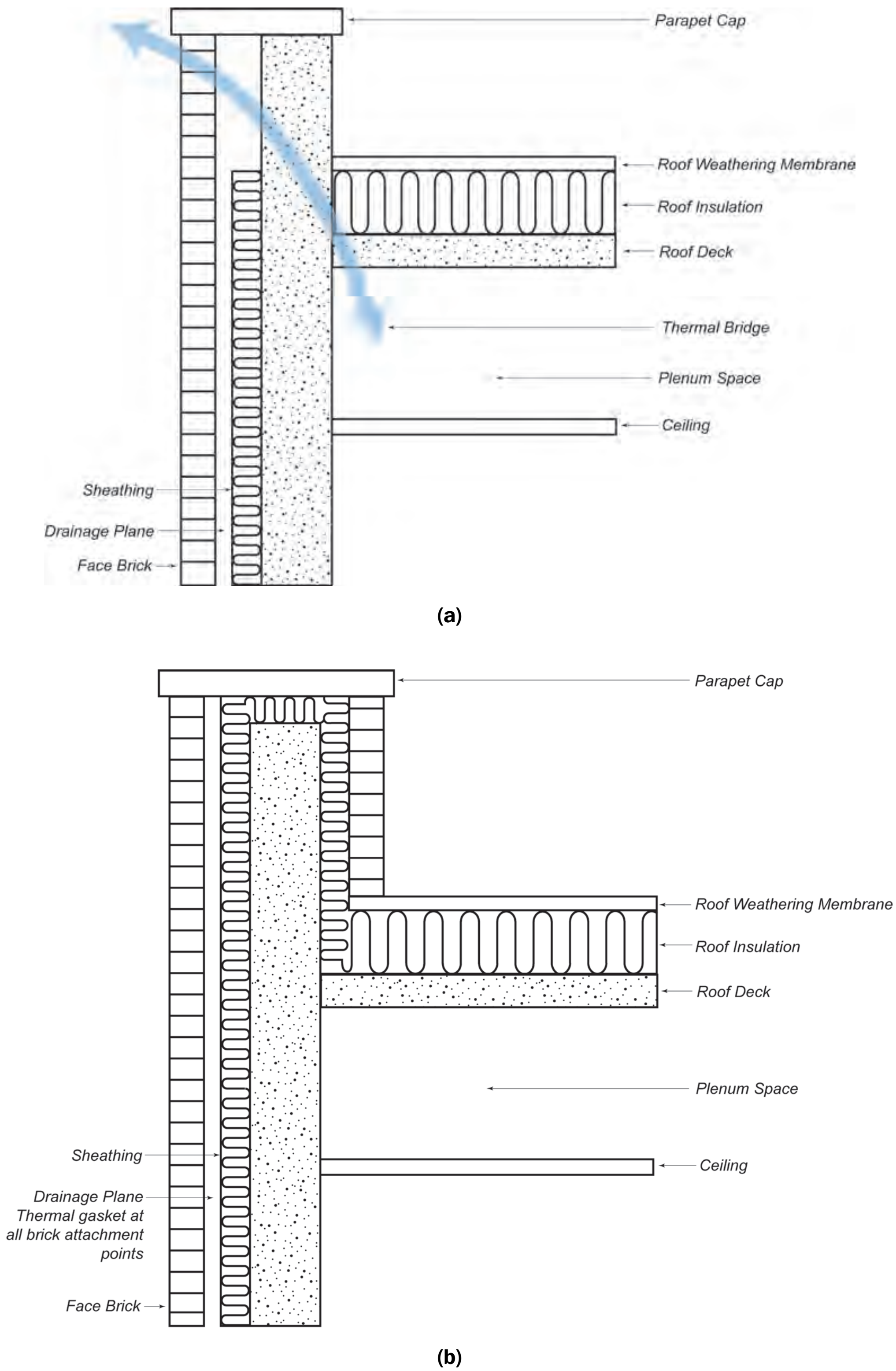
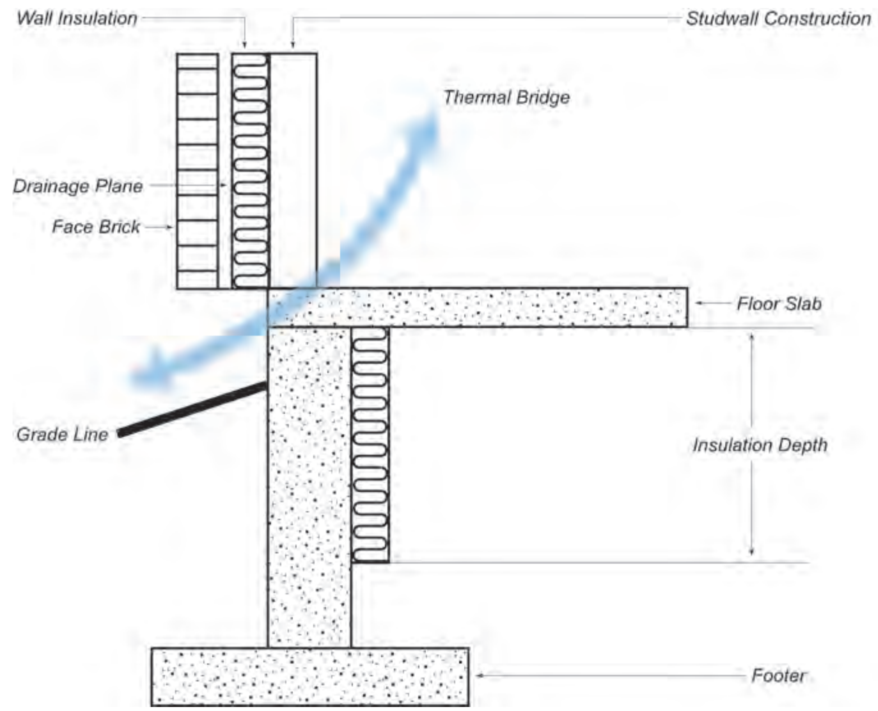


Figure 5-6 (EN21) Thermal Bridges—Parapets: (a) Problem and (b) Solution



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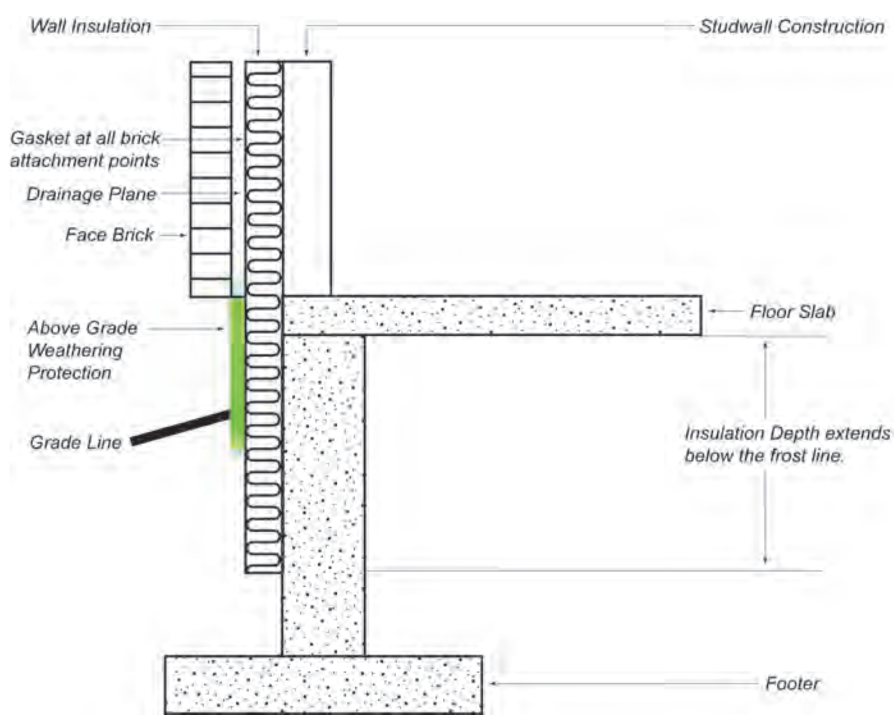

(b)

Figure 5-7 (EN21) Thermal Bridges—Foundations: (a) Problem and (b) Solution

IES Standard 90.1, Table 5.5.4.4.1 (ASHRAE 2013a). These multipliers allow for a higher SHGC for vertical fenestration with overhangs.

WINDOW DESIGN GUIDELINES FOR THERMAL CONDITIONS

Uncontrolled solar heat gain is a major cause of energy use for cooling in warmer climates and of thermal discomfort for occupants. Appropriate configuration of windows according to the orientation of the wall on which they are placed can significantly reduce these problems.

EN23 *Unwanted Solar Heat Gain is Most Effectively Controlled on the Outside of the Building* (Climate Zones: all)

Significantly greater energy savings are realized when sun penetration is blocked before it enters the windows. Horizontal overhangs at the tops of the windows are most effective for south-facing façades and must continue beyond the width of the windows to adequately shade them (see Figure 5-8). Vertical fins oriented slightly north are most effective for east- and west-facing facades. Consider louvered or perforated sun control devices, especially in primarily overcast and colder climates, to prevent a totally dark appearance in those environments.

Warm Climates

EN24 *Building Form and Window Orientation* (Climate Zones: ❶ ❷ ❸)

In warm climates, south-facing glass can be more easily shielded and can result in less solar heat gain and glare than can east- and west-facing glass. During early building configuration studies and predesign, preference should be given to site layouts that permit elongating the building in the east-west direction and that permit orienting more windows to the north and south. A good design strategy avoids areas of glass that do not contribute to the view from the building or to the daylighting of the space. If possible, configure the building to maximize

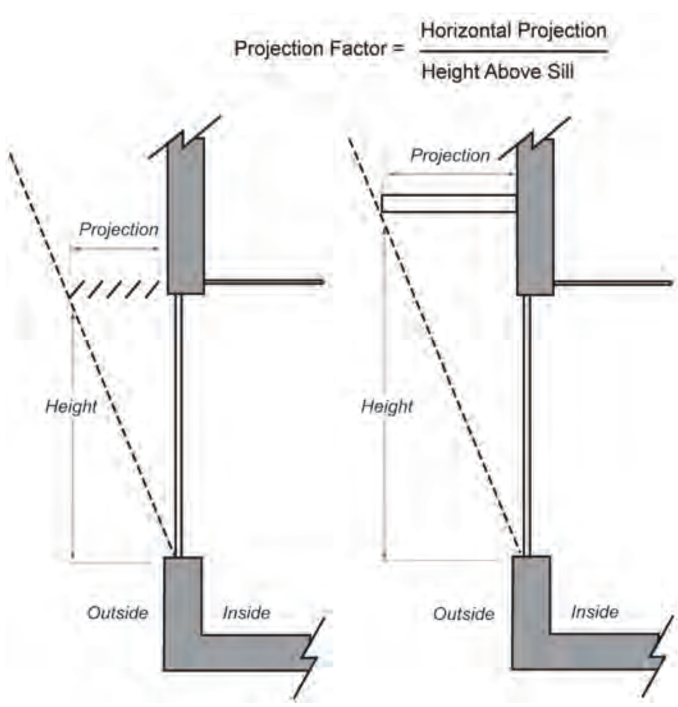

Figure 5-8 (EN23) Windows with Overhangs

north- and south-facing walls and glass by elongating the floor plan. Because sun control devices are less effective on the east and west façades, the solar penetration through the east- and west-facing glazing should be minimized. This can be done by reducing the area of glazing or, if the glass is needed for view or egress, by reducing the SHGC, or by utilizing automated operable shading systems. For buildings where a predominantly east-west exposure is unavoidable, more aggressive energy conservation measures will be required in other building components to achieve an overall 50% energy savings.

EN25 ***Glazing*** **(Climate Zones: ❶ ❷ ❸)**

For north- and south-facing windows, select windows with a low SHGC and an appropriate visible transmittance (VT). Certain window coatings, called *selective low-e,* transmit the visible portions of the solar spectrum selectively, rejecting the nonvisible infrared sections. These glass and coating selections can provide a balance between VT and solar heat gain. Window manufacturers market special "solar low-e" windows for warm climates. All values are for the entire fenestration assembly, in compliance with NFRC procedures, and are not simply center-of-glass values. For warm climates, a low SHGC is much more important for low energy use than the window assembly U-factor. Windows with low SHGC values tend to have low center-of-glass U-factors because they are designed to reduce the conduction of the solar heat gain absorbed on the outer layer of the glass through to the inside of the window.

Cold Climates

EN26 ***Window Orientation*** **(Climate Zones: ❹ ❺ ❻ ❼ ❽)**

Only the south glass receives much sunlight during the cold winter months. If possible, maximize south-facing windows by elongating the floor plan in the east-west direction and relocate windows to the south face. Careful configuration of overhangs or other simple solar control devices allows for passive heating when desired but prevents unwanted glare and solar overheating in the warmer months. To improve performance, operable shading systems that achieve superior daylight harvesting and passive solar gains and also operate more effectively when facing east and west directions should be used. Unless such operable shading systems are used, glass facing east and west should be significantly limited. Areas of glazing facing north should be optimized for daylighting and view and focus on low U-factors to minimize heat loss and maintain thermal comfort by considering triple glazing to eliminate drafts and discomfort. During early building configuration studies and predesign, preference should be given to sites that permit elongating the building in the east-west direction and that permit orienting more windows to the south.

EN27 ***Glazing*** **(Climate Zones: ❹ ❺ ❻ ❼ ❽)**

Higher SHGCs are allowed in colder regions, but continuous horizontal overhangs are still necessary to block the high summer sun.

EN28 ***Thermal Bridging—Fenestration*** **(Climate Zones: ❸ ❹ ❺ ❻ ❼ ❽)**

In colder climates, it is essential to select a glazing unit to avoid large amounts of condensation. This requires an analysis to determine internal surface temperatures, because glass is a higher thermal conductor as compared to the adjacent wall in which it is mounted. There is a risk of condensation occurring on the inner face of the glass whenever the inner surface temperature approaches the interior dew-point temperature.

Careful specification is also necessary to ensure that the framing of the glazed units also incorporates a thermal break.

In a typical fenestration situation where thermal bridging arises, the detailing of how a piece of well-insulated glazing abuts the opaque façade is important, whether it is through a metal mullion system or whether it just frames into the wall. Windows that are installed out of

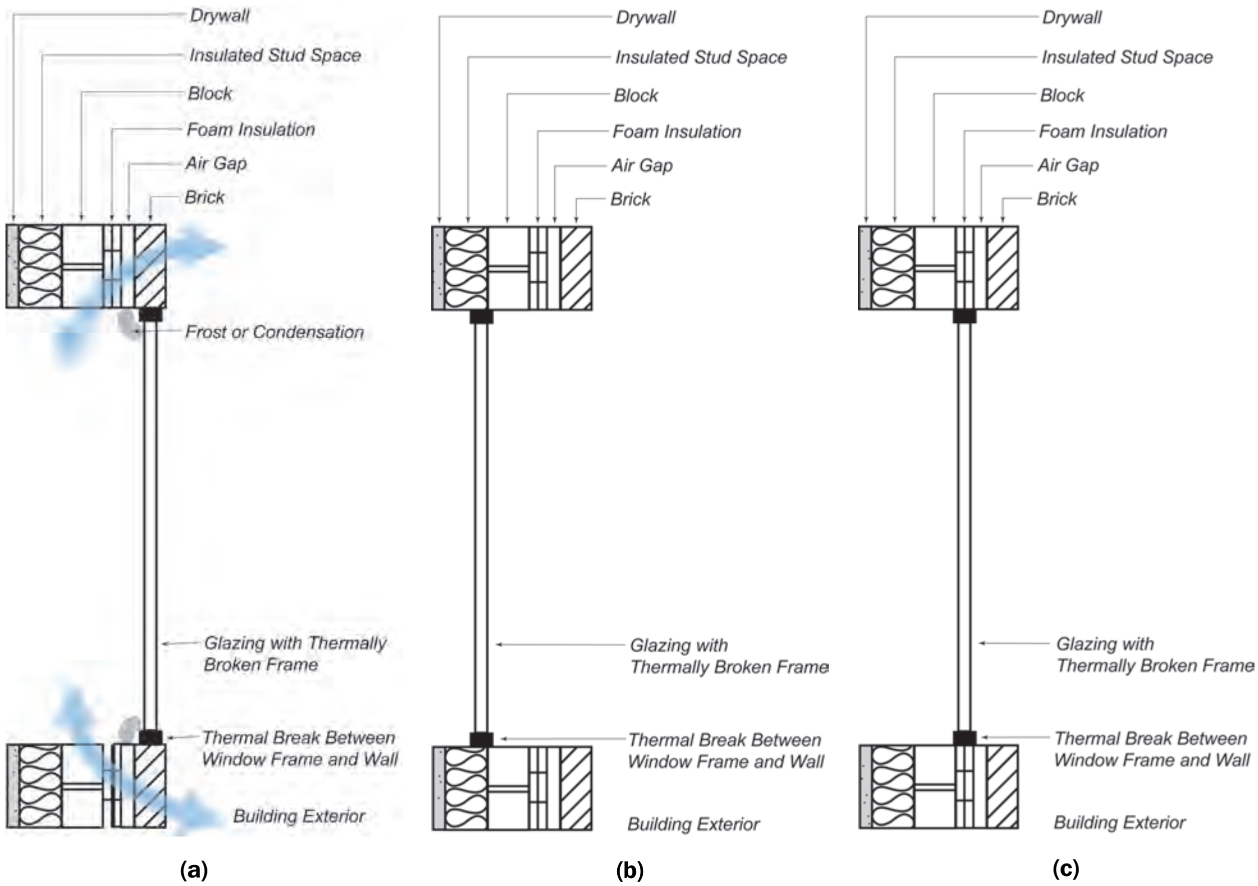

Figure 5-9 (EN28) Fenestration with the Frame Thermal Break in Alignment with Wall Insulation Options: (a) Problem, (b) Solution 1, and (c) Solution 2

the plane of the wall insulation defeat the thermal break in the window frame (see Figure 5-9a). In cold climates this causes condensation and frosting. The normal solution is not to rebuild the wall but to blow hot air against the window to increase the interior surface temperature of the frame and glazing, which increases the temperature difference across the glazing and reduces the interior film coefficient thermal resistance from 0.68 to 0.25 h·ft^2·°F/Btu.

Fenestration should be installed to align the frame thermal break with the wall thermal barrier. There are two solutions shown in Figure 5-9. Solution 1 illustrates the glazing aligned with the insulated stud space (Figure 5-9b). Solution 2 illustrates the glazing aligned with the exterior foam insulation (Figure 5-9c). Either of these solutions will minimize the thermal bridging of the frame due to glazing projecting beyond the insulating layers in the wall.

WINDOW DESIGN GUIDELINES FOR DAYLIGHTING

Good Design Practice

EN29 *Visible Transmittance (VT)* (Climate Zones: all)

Using daylight in place of electrical lighting significantly reduces the internal loads and saves cost on lighting and cooling power. In the U.S., it is estimated that 10% of the total energy generated per day is consumed by electrical lighting during daylight hours.

The amount of light transmitted in the visible range affects the view through the window, glare, and daylight harvesting. For the effective use of daylight, separate the daylight glazing from the view glazing. The daylight glazing should be limited to 5% of the floor area and use higher VT glazing (0.60 to 0.70) than the view glazing.

VTs below 0.50 can make the glazing appear noticeably tinted and make the outside view dim to occupants. However, lower VTs may be required to prevent glare, especially on the east and west facades or for higher window-to-wall ratios (WWRs). Lower VTs may also be appropriate for other conditions of low sun angles or light-colored ground cover (such as snow or sand). Use adjustable blinds to handle intermittent glare conditions.

High continuous windows are more effective than individual (punched) or vertical slot windows for distributing light deeper into the space and provide greater visual comfort for the occupants. Try to expand the tops of windows to the ceiling line for daylighting, but locate the bottoms of windows no lower than 30 in. above the floor. Daylighting can be achieved with higher WWRs, which can lead to higher heating and cooling loads.

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DAYLIGHTING

Daylighting should be considered in two forms, sidelighting and toplighting. In grocery stores, sidelighting is typically limited to the front of the store at the entries and in administrative spaces. Sidelighting can also be very effective in back of house by using high windows. Toplighting can take the form of skylights, rooftop monitors, clearstories, or tubular daylighting devices (TDDs) and can effectively light the general sales areas.

Daylighting (toplighting – skylights or rooftop monitors) is required per ASHRAE/IES Standard 90.1 (ASHRAE 2013a) in climate zones 1 through 5 in spaces greater than 2500 ft^2 directly under a roof with ceiling heights greater than 15 ft. While not all sales floor spaces are designed with these high ceilings, many open ceiling spaces will exceed the 15 ft threshold.

Therefore, in climate zones 1 through 5, for areas greater than 2500 ft^2 directly under a roof with a ceiling height above 15 ft, follow the prescriptive requirements of Standard 90.1 and install skylights that daylight at least 50% of the sales floor area. In all climate zones, rooftop monitors, clearstories, and TDDs may be used to achieve the minimum 50% of sales floor daylighted.

For all climate zones, when skylights or rooftop monitors are installed, follow all of the recommendations presented in this section.

From an energy standpoint, the energy savings from reduced loads as the electric lights are dimmed in response to daylight must be greater than any increase in heating/cooling loads resulting from using skylights or rooftop monitors to bring the daylight into the space. The following recommendations for skylights and rooftop monitors are for spaces of at least 2500 ft^2 in area.

Daylighting should be thought of in two components: direct sun, which should be avoided in general retail spaces due to the high light levels and the high contrasts created by direct sunlight, and diffuse daylight, which can be very beneficial in brightening the sales area, allowing a reduction in the energy used by the electric lighting system.

GENERAL RECOMMENDATIONS

When integrating daylighting is in your design, please consider the following actions:

- Use rooftop monitors, clearstories, or TDDs instead of skylights to light the general sales floor area for better control of heat gain/loss.
- Include windows in offices, conference rooms, and break rooms to promote a connection to the outdoors. Windows should be divided into daylighting glazing, which should not be more than 5% of the floor area and should be located high at the ceiling line, and view glazing.
- Use light-colored matte finishes on ceilings (80%+ reflectance) and walls (60%+ reflectance) to promote interreflections and better use of electric light and daylight.
- For controls, include daylight dimming on the sales floor, time switches (time clocks) in corridors, and vacancy sensors (manual ON, automatic OFF) with daylight override in offices, break rooms, and conference rooms.
- In stockrooms where the lighting-power density (LPD) allowance is low at 0.6 W/ft^2 and occupancy sensors are required, the occupancy patterns and HVAC impacts will have a significant impact in determining if skylights will be cost-effective. If these spaces are located along an outside wall, add windows before adding skylights.

DL1 *Daylighting Early in the Design Process* (Climate Zones: all)

With grocery stores, the program and site plan are the main drivers that establish the shape and footprint of the building. Planning criteria often result in creating deep floor plates that work well for rooftop monitors or skylights. Additionally, and often an afterthought, daylight through windows into offices, break rooms, and conference rooms can save energy and provide employees a connection to the outdoors.

Tubular Daylighting Devices

Tubular daylighting devices (TDDs) are one daylighting option now being used with greater frequency. The devices harvest daylighting through roof-mounted domes connected via a tubing system to a diffuser that looks very similar to a ceiling light fixture. The result is an effective daylighting source with less heat transfer than traditional skylights or roof monitors that can provide significant annual energy and operating cost savings.

Roof-Mounted Domes for Tubular Daylighting Devices
Photographs reproduced with permission of Giant Eagle, Inc. (left) and Solatube International (right)

Giant Eagle, Inc., a mid-Atlantic region grocery store chain, was looking to create a daylighting design that would cut operational costs and enhance the customer shopping experience. They installed 335 TDDs in an 88,000 ft^2 facility to displace electric lighting during daylight hours. The design strategy used both the TDDs and exterior wall windows.

The reliable distribution of natural light provides perfect color rendition, making products more appealing and making the store feel brighter, cleaner, and more comfortable for shoppers. The chain expects to pay less for electricity at this location as compared to the previous location, which was considerably smaller. The store has been so satisfied with the system that they claim that if they had the chance to start over, they would nearly double the number of devices installed to cover the entire sales floor, offices, prep areas, and the back room.

Stater Bros. Markets, a grocery store chain operating throughout southern California, installed 164 TDDs in their Chino Hills store. The TDDs are used to fully daylight the entire 43,235 ft^2 sales floor (including produce, pharmacy, and checkout aisles) and back-of house stocking spaces. Panel-level photosensor controls provide multiple controlled lighting zones using a single photosensor that is integrated into the daylighting system.

The resulting daylighting not only serves to fully displace the electric lighting during the majority of daylight hours throughout the year but also enhances the appearance of the products being sold. The devices also provide accent lighting for key architectural details and brighten the store's perimeter walls. The daylighting system is expected to cut the store's annual lighting energy costs by nearly half. An added benefit is the ability to keep the store lighted in the event of a black-out situation.

Tubular Daylighting Device Diffusers on the Grocery Sales Floor
Photographs reproduced with permission of Giant Eagle, Inc.

Daylight strategies impact the design at different levels of scale in each phase of design and can be characterized by four categories.

Predesign. During predesign, the daylight strategy's focus is on building configuration studies and the shaping of the floor plate. The goal is to maximize access to windows and daylight in offices, break rooms, and conference rooms by orienting fenestration in a predominantly north- and south-facing direction.

In the sales floor area, rooftop monitors and skylights should be placed to evenly light the space to at least 125% of the design footcandles on the equinox. For spacing recommendations, see the sales floor section of DL3.

Schematic design. During the schematic design phase, daylight strategies are about interiors, focusing on optimizing daylight penetration, defining ceiling height, layout of skylights, and wall and ceiling reflectances. The planning focus is directed toward coordinating space types where daylight is desired.

Design development. During the design development phase, the daylighting strategies' focus is on envelope design to optimize quantity and quality of daylight while minimizing solar gains. The interior design focus is on surface reflectivity and optimizing sales display layout to align with visual and thermal comfort requirements.

Construction documents (CDs). Coordination of electrical lighting includes the placement of photosensors and occupancy sensors for controlling automated daylight switching and dimmable ballasts.

DL2 *Daylighting Analysis Tools to Optimize Design* (Climate Zones: all)

This Guide is designed to help achieve energy savings of 50% without energy modeling; however, daylighting modeling programs make evaluating energy-saving trade-offs faster and daylighting designs far more precise.

Annual savings will have to be calculated with an annual whole-building energy simulation tool after the daylighting design tools have been used to determine the footcandles in the spaces and after toplighting has been appropriately sized. Most daylighting analysis tools do not help with heating and cooling loads or other energy uses; they predict only illumination levels and electric lighting use.

DL3 *Space Types, Layout, and Daylight* (Climate Zones: all)

The goal is to locate skylights in sales and storage spaces to offset the energy use of electric lighting with daylight. The potential of energy saving through daylighting varies and depends on program and space types, which can be broadly characterized by the following three categories of occupied spaces.

Sales floor. Significant energy can be saved in sales spaces due to the high electric LPDs; however, daylighting needs to be carefully coordinated with the desired store image. The following tips apply to the sales floor:

- Space skylights 1.4 times the floor-to-ceiling height from the edge of one skylight to the edge of the next skylight (if the ceiling height is 16 ft, the skylights would be spaced 24 ft apart).
- Dim the electric lighting in response to the available daylight by using open-loop photosensors located in the rooftop monitors or skylights. Closed-loop sensors may also be used and should be placed at the same height as the light fixtures.
- Provide a skylight area of 3% to 5% of the roof area with a skylight VT of at least 0.40 (or provide a minimum skylight effective aperture of at least 1%).
- If using skylights, the glazing material or diffuser should have a measured haze value greater than 90% when tested according to ASTM D1003 (ASTM 2013).

Storage areas. While the storage electric LPD is generally low, daylighting large storage areas can be an effective design option because during daylight hours the electric lighting can be turned off. The following tips apply to storage areas:

- Space skylights 1.4 times the floor-to-ceiling height from the edge of one skylight to the edge of the next skylight (if the ceiling height is 16 ft, the skylights would be spaced 24 ft apart). Switch the electric lighting in response to the available daylight by using open-loop photosensors located in the rooftop monitors or skylights. Closed-loop sensors may also be used and should be placed at the same height as the light fixtures.
- Provide a skylight area of 3% to 5% of the roof area with a skylight VT of at least 0.40 (or provide a minimum skylight effective aperture of at least 1%).
- If using skylights, the glazing material or diffuser should have a measured haze value greater than 90% when tested according to ASTM D1003.
- If storage spaces are located along an outside wall, add windows before adding skylights.

Offices, break rooms, and training rooms. From an energy performance standpoint, the first priority is to locate offices, break rooms, and training rooms on the perimeter, preferably in a north- and south-facing configuration. The following tips apply to open office spaces:

- Locate workstations next to windows within the primary and secondary daylight zones to maximize daylight harvesting.
- Use photocontrols in primary daylight zones when installed wattage is 150 W or greater.
- Use photocontrols in primary and secondary daylight zones when combined wattage is 300 W or greater.
- Use local articulated task lights to supplement daylight and electric light.
- Use manual ON occupancy sensors with daylight override (vacancy control).

DL4 *Skylight Thermal Transmittance* (Climate Zones: all)

As an alternative to skylights, use rooftop monitors for skylighting. Never use east- or west-facing rooftop monitors due to excessive summer heat gain and the difficulty of controlling direct sunlight. Rooftop monitors with operable glazing may also help provide natural ventilation in temperate seasons. Typically, north-facing rooftop monitors have less than 20% of the heat gain of skylights. They provide less than one-third of the daylighting potential per square foot of prismatic skylights. Additional considerations include the following:

- Shade south-facing rooftop monitors and skylights with exterior/interior sun controls such as screens, baffles, or fins.
- Insulate the skylight curb above the roof line with R-5.0 rigid c.i. in conditioned spaces.
- In hot climates:
  - Use north-facing rooftop monitors for skylighting to eliminate excessive solar heat gain and glare.
  - Reduce thermal gain during the cooling season by using skylights with a low overall thermal transmittance (see U-factor recommendations in Chapter 4).
- In moderate and cooler climates:
  - Use either north- or south-facing rooftop monitors for skylighting to eliminate excessive solar heat gain and glare.
  - Reduce summer heat gain as well as winter heat loss by using skylights with a low overall thermal transmittance. Use a skylight frame that has a thermal break to reduce heat loss/gain and winter moisture condensation on the frame.

DL5 *Interactions* (Climate Zones: all)

Thermal gains and losses associated with skylights should be balanced with daylight-related savings achieved by reducing electric lighting consumption.

Skylight well height (side walls below the skylight) should be as short as possible. If a skylight well is over 2 ft high, splay skylight opening at 45° to maximize daylight distribution and minimize glare (see Figure 5-10).

Ceiling and skylight well reflectance values should be a minimum of 80% to reduce contrast between skylight and ceiling.

DL6 *Expanded Recommendations for Daylight Zones* (Climate Zones: all)

Additional considerations for daylight zones include the following:

- The daylight zone is the area of the skylight plus 70% of the floor-to-ceiling dimension in all directions from the edge of the skylight (see Figure 5-11).

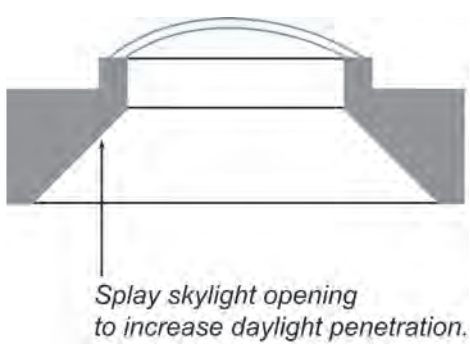


Figure 5-10 (DL5) Skylight (Horizontal Fenestration)

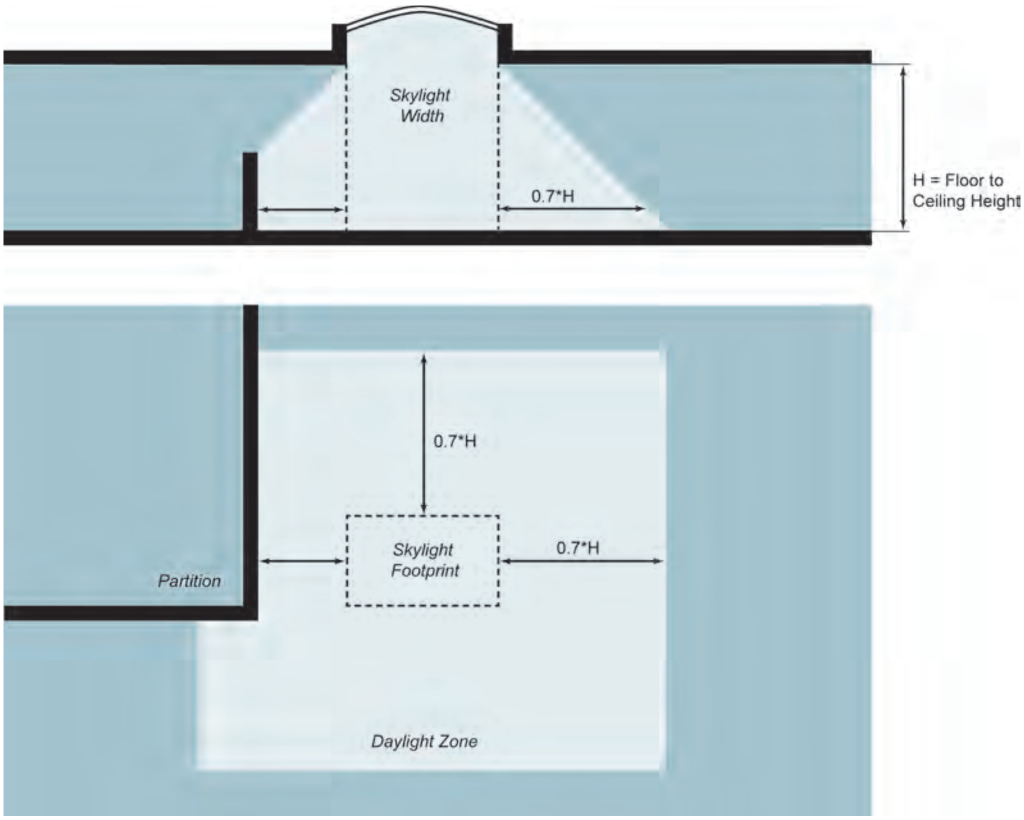


Figure 5-11 (DL6) Daylight Zone under Skylight

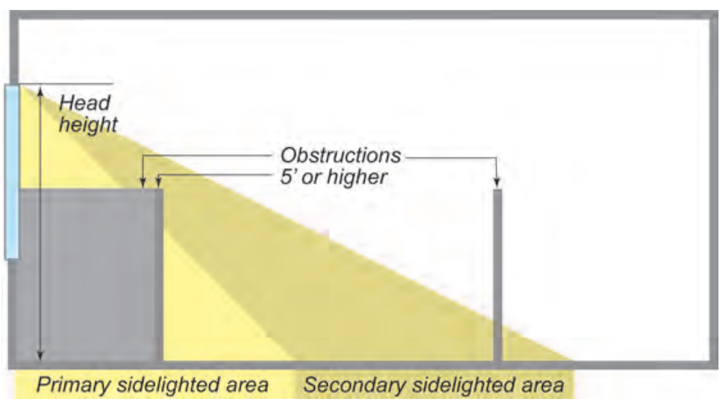


Figure 5-12 (DL6) Sidelighting Zone

- The sidelighted daylight zone is the floor area directly adjacent to vertical glazing with an area equal to the product of the sidelighted area width and the sidelighted area depth (see Figure 5-12).
  - The primary sidelighted area width is the width of the window plus, on each side, 1/2 of the window head height or the distance to any 5 ft or higher vertical obstruction.
  - The sidelighted area depth is the horizontal distance perpendicular to the glazing, which is one window head height for the primary zone and two window head heights for the secondary zone or the distance to any 5 ft or higher vertical obstruction.
- Sales floor area electric lights should automatically dim in response to daylight.

ASHRAE/IES Standard 90.1-2013 Sidelighting (Vertical Fenestration) Prescriptive Requirements

Automatic daylight responsive controls for sidelighting: In any space where the combined input power of all general lighting completely or partially within the primary sidelighted areas is 150 W or greater, the general lighting in the primary sidelighted areas shall be controlled by photocontrols.

In any space where the combined input power of all general lighting completely or partially within the primary and secondary sidelighted areas is 300 W or greater, the general lighting in the primary sidelighted areas and secondary sidelighted areas shall be controlled by photocontrols.

- In storage areas, use photosensor switching in response to daylight.
- Control luminaires in groups around skylights; when daylight zones overlap, a single control zone may be used. The daylighting control system and/or photosensor should include a five-minute time delay or other means to avoid cycling caused by rapidly changing sky conditions and a one-minute fade rate to change the light levels by dimming.
- Specify dimming ballasts that dim down to at least 10% of full output. Photosensors should include a one-minute fade rate to change the light levels by dimming and should exhibit a slow, smooth, linear response. New fluorescent lamps must be burned-in per manufacturer recommendation prior to dimming. Follow manufacturers' recommendations for setup and calibration setting.
- Specify multilevel daylight switching to provide a minimum of 5 setpoints—100% light level, between 85% and 80% light levels, between 70% and 50% light levels, between 40% and 20% light levels, and off. Follow manufacturers' recommendations for setup and calibration settings.

DL7 *Photosensor Placement* (Climate Zones: all)

Photosensors used should be specified for indoor illumination range.

An open-loop system is one where the photosensor responds only to daylight levels but is still calibrated to the desired light level received on the floor of the sales and storage areas. The best location for an open-loop photosensor is inside the skylight well.

A closed-loop system is one where the photosensor responds to the combination of electric lighting and daylight but is still calibrated to the desired light level received on the sales floor or work surface. The best location for a closed-loop photosensor is at same height as the luminaires that the photosensor is controlling. The photosensor should face the work surface or sales area.

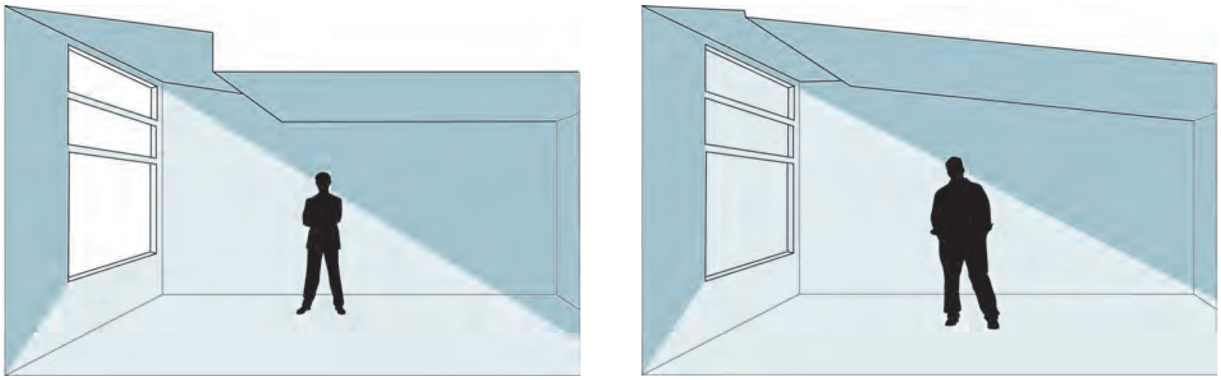
DL8 *Sidelighting—Ceiling and Window Height* (Climate Zones: all)

For good daylighting in offices, break rooms, and training rooms, a minimum ceiling height of 9 ft is recommended. When daylighting is provided exclusively through sidelighting, it is important to elevate the ceiling on the perimeter and extend glazing to the ceiling. Additional reflectance to increase lighting levels can be achieved by sloping the ceiling up toward the outside wall (see Figure 5-13). Often, this can easily be done with the typical *t*-grid ceiling found in many of these spaces.

The effective aperture for sidelighting is the area of glazing in an unobstructed wall multiplied by the VT of vertical glazing, divided by the floor area in the daylight zone.

DL9 *Calibration and Commissioning* (Climate Zones: all)

Even a few days of occupancy with poorly calibrated controls can lead to permanent overriding of the system and loss of all savings. All lighting controls must be calibrated and commissioned after the finishes are completed and the racking and products are in place. Follow manufacturers' recommendations for setup and calibration setting (most photosensors require



(a) (b)

Figure 5-13 (DL8) Daylighting Ceiling Configurations: (a) Raised Ceiling at Façade and (b) Sloped Ceiling at Façade

daytime and nighttime calibration settings). The photosensor manufacturer and the quality assurance (QA) provider should be involved in the calibration. Document the calibration and commissioning settings and calendar intervals for future recalibration. Systems should be checked for proper calibration and operation at least every two years.

DL10 *Daylight Levels* (Climate Zones: all)

Occupants expect higher combined light levels in daylighted spaces. Consequently, it is more acceptable to occupants when the electric lights are calibrated to dim or switch when the combined daylight and electric lighting exceeds 1.20 times the designed light level—i.e., if the ambient electric light level is designed for 50 maintained footcandles (fc), the electric lights should begin to dim when the combined level is 60 fc ($50 \times 1.20 = 60$). When using daylight switching, the electric lights should step down only when the initial designed light level is maintained after the step down.

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ELECTRIC LIGHTING

GOALS FOR GROCERY LIGHTING

Lighting design for grocery sales spaces should support the goals of attracting customers, facilitating merchandise evaluation, and enabling completion of the sale. These goals can be obtained in consort with lower LPDs through the use of high-efficacy light sources, lighting controls, and good design practice.

Ambient lighting. Uniform general illuminance for merchandising areas as recommended in *The Lighting Handbook* (IES 2011) ranges from 15 to 50 fc depending on the store type and merchandising strategies. Designing to the appropriate Illuminating Engineering Society of North America (IES) recommended ambient footcandle levels will establish the framework for an effective accent lighting strategy that achieves accent lighting goals without sacrificing energy efficiency.

Perimeter lighting. Illuminance ratios of no more than 2:1 of ambient light levels are recommended for the creation of effective perimeter wall lighting. Effective perimeter lighting includes adequate illumination levels for merchandise evaluation as well as for contributing to the perceived brightness within the space. Using T8, T5, or light-emitting diode (LED) asymmetric wall-wash luminaries is an energy-effective option for creating perimeter lighting. Alternatively, dedicated valance or casework with high-efficacy lighting is another energy-effective solution as long as automatic controls to turn the lighting off are included.

Accent and/or task lighting. Accent and task lighting provide additional illumination required to enhance merchandise color, texture, and detail, which aids in merchandise attraction as well as merchandise evaluation. Task lighting levels are typically 1 1/2 to 2 times that of ambient lighting. However, some tasks or product evaluation may require higher ratios, while others tasks or product evaluation can take place under ambient light levels. To create visual contrast and interest, accent lighting ratios of 5:1 (accent to general ambient and/or perimeter) should be designed as recommended in *The Lighting Handbook*. Effective accent lighting is best accomplished with point source directional luminaries such as ceramic metal halide (CMH) and LED, which also represent the most energy-efficient accent option.

Other lighting features to consider that play important roles in merchandise recognition and the perceived lighted environment include decorative lighting (pendants, sconces, etc.), internal casework, and equipment lighting, as well as wall-integrated lighting.

As shown in the plan schematic in Figure 5-14, there are different departments that may require different light levels and different lighting styles. Later in this chapter, the various space types are discussed in more detail, including basic lighting design recommendations (see EL20 to EL27).

INTERIOR LIGHTING

Good Design Practice

EL1 *Savings and Owner Acceptance* (Climate Zones: all)

Lighting first and foremost must provide proper light levels and color quality to foster sales. These goals, however, need not and should not be counterproductive to good energy efficiency practices. Accomplishing lighting quality and quantity with the least amount of needed energy also aids in a successful grocery environment and improved profitability. The LPDs in the Chapter 4 recommendation tables are based on IES recommended light levels using LED fixtures or fluorescent fixtures (using high-performance lamps and ballasts) for the general lighting and LED or CMH for the accent lighting. To maximize energy savings, this Guide does not recommend incandescent or compact fluorescent (CFL) lamps be used.

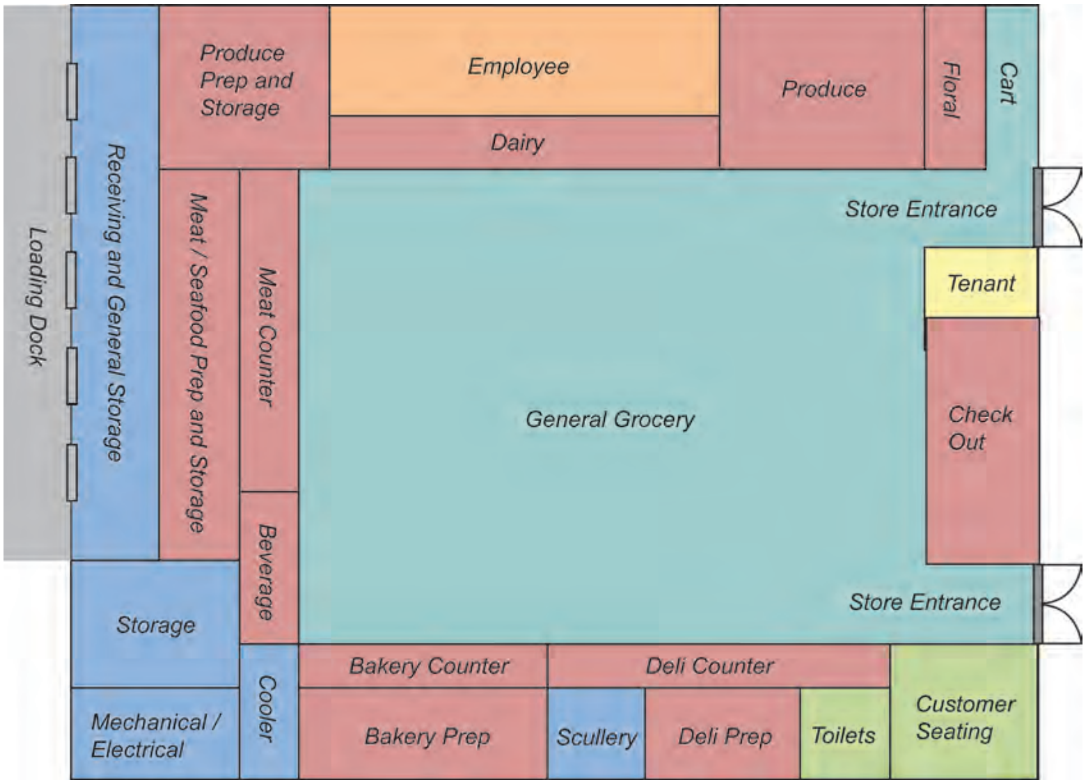


Figure 5-14 Grocery Space Planning Schematic

EL2 *Space Planning—Offices, Break Rooms, and Training Rooms* (Climate Zones: all)

Wherever possible, locate offices, meeting rooms, and break rooms on the outside wall of the building to provide employees a connection to the outdoors and daylighting to offset electrical lighting loads (see DL8). Use manual ON occupancy sensors with a daylight override to save the most energy in these spaces (vacancy switches).

EL3 *Space Planning—Sales Floor and Circulation Aisles* (Climate Zones: all)

Corporate space plans are often a given in chain-operated grocery stores. These plans usually contain circulation aisles that extend deep into the store. Store designers place focal points and/or end walls where the aisle terminates or changes direction. This design technique creates focal corridors to capture the customer's attention. Use of focal lighting to highlight a feature or end wall within a focal corridor aids in drawing customers deeper into the store.

EL4 *Lighting Walls/Perimeter Lighting* (Climate Zones: all)

Better eye adaptation, luminous comfort, and impressions of pleasantness and space can be achieved when light is distributed to the walls. Lighting vertical surfaces and walls also contributes to the visual effect of perceived brightness. Perceived brightness enforces the positive effects of a well-lit space, and without this brightness a space can look dark and underlighted, which may result in the store adding additional lighting, which in turn increases energy use. To effectively light surfaces and walls, use dedicated wall-wash luminaires. Alternatively, applications that use direct coves or valance lighting or locate ambient lighting fixtures closer to walls are also acceptable options for wall lighting. There will be occurrences where it is desirable to use accent lighting for certain perimeter wall features. When placing fixtures, always consider the final location of the merchandise to be illuminated, not the wall surface behind the merchandise (see Figure 5-15).

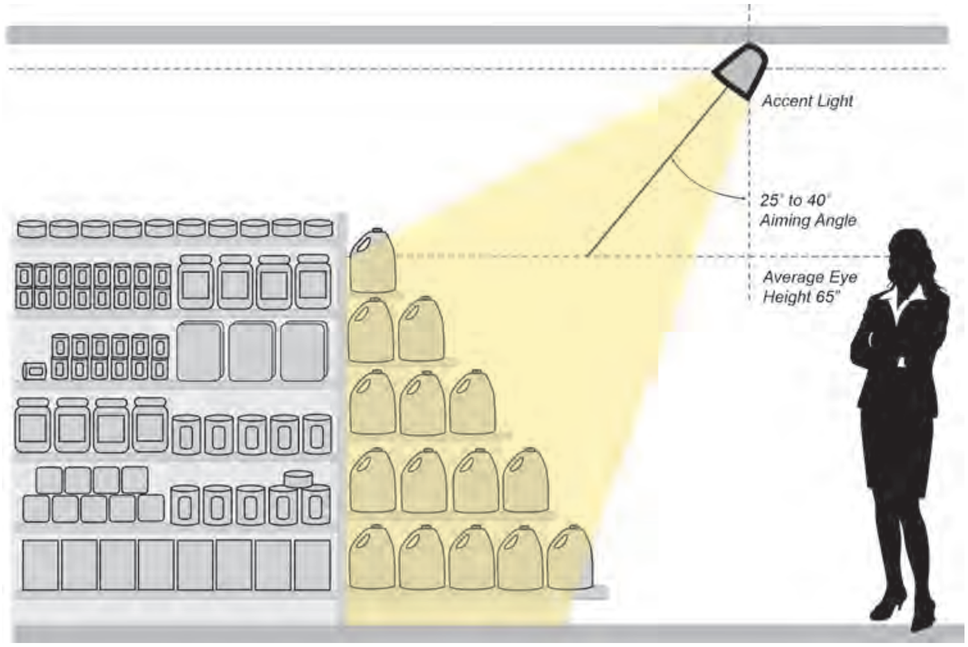


Figure 5-15 (EL4 and EL5) Accent Lighting Aimed at 40 Degrees

EL5 Additional Interior Lighting/Task and Accent Lighting (Climate Zones: all)

Accent and general lighting are included in the 1.15 W/ft^2 allowance shown in the recommendation tables in Chapter 4, and accent lighting should be specifically designed and directed to provide additional illumination for critical tasks and evaluation (such as food preparation in the deli) or to highlight merchandise (accent lighting). ***Note:*** Additional lighting required for critical task evaluation and highlighting merchandise must be accomplished within the 1.15 W/ft^2 allowance. (See EL16 for switching recommendations.)

Need for additional task illumination at selected merchandise, or within some work areas, within the store must be weighed against the added energy and operational costs associated with providing the additional lighting. The supplemental lighting should be allocated only to the tasks and merchandise product where or when general lighting is insufficient to properly and safely perform the tasks and/or evaluate the product. Food preparation areas, kitchens, and bakeries as well as other areas where reading labels or instructions are critical may require additional task illumination.

The use of accent lighting to highlight all merchandise does not create the proper contrast ratios. Use accent lighting to highlight key merchandise locations or vignettes to "feature display" light levels (a minimum of three to a maximum of ten times the general and/or perimeter merchandise lighting level in the area of the display).

Along perimeters, accent lighting should be aimed up at 40° from vertical (straight down) to reduce reflected glare off specular surfaces. The aimed accent light should not exceed 45° from vertical, and attention should be given to both the direct and reflected characteristics of the light distribution. In areas open to customer view from multiple vantage points, the accent lighting should be aimed up at no more than 40° from vertical to reduce the possibility of direct glare visible to the customer (see Figure 5-15).

EL6 Decorative Lighting (Climate Zones: all)

Decorative lighting (wall sconces and pendant fixtures) can add visual interest and focus to the space, especially at the sales transaction area. These fixtures are included in the tabulation of the base LPD, and consideration should be given to energy-efficient solutions, including CFL, CMH, and LED lighting.

Display and Accent Lighting Strategies

The use of display and accent lighting strategies within grocery store environments has become popular with owners and their designers to set their spaces apart from the competition and enhance brand identity. While applying display lighting strategies can enhance the visual ambiance and character of a space and enforce brand identity, caution must be exercised so as to maintain energy efficiency targets and code compliance.

Spaces with high base illumination *(*general lighting*)* footcandles should refrain from use of accent and display lighting. When desired, accent and display lighting is best introduced by using a layered lighting design approach as follows:

- First, circulation and general illumination are kept to a minimum, with typical targets of 15 to 25 fc.
- Next, task lighting is introduced at typical product sales areas, with typical targets of 30 to 50 fc.
- Finally, the accent/display lighting layer is added to selected feature merchandizing zones using a 5:1 ratio to surround general lighting of 75 to 125 fc.

A well-executed lighting design can use display and accent lighting yet maintain energy efficiency and meet code compliance. Based on overall strategy, the design may actually result in a lower LPD than a more traditional uniform lighting application.

Traditional Approach

Bright, uniform lighting represents a traditional approach to illuminating groceries. Using high-efficacy luminaires, the entire space is lighted to target to maintain levels of 50 to 70 fc. The high-efficiency lighting allows for high light levels while meeting code compliance and energy targets.

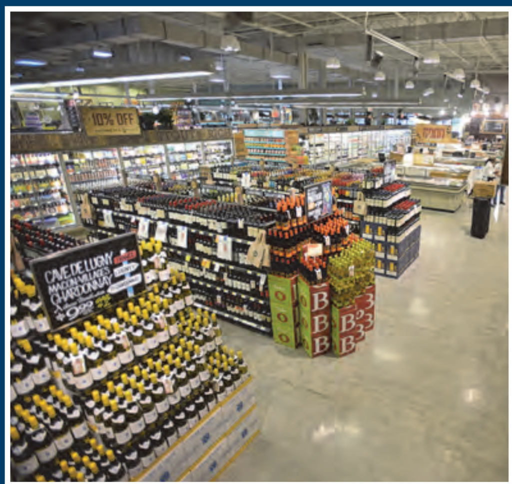
Traditional Lighting Design
Photographs reproduced with permission of Whole Foods Market (left) and NREL, credit Ian Doebber (right)

Adding accent lighting to designs with bright, uniform lighting is impractical, as the added lighting power required to achieve typical accent light targets (5:1 of general lighting) will result in excessive energy consumption and may exceed the allowed LPD required for code compliance.

Alternative Approach

As an alternative to bright, uniform lighting, some grocery store lighting designs use accent and display light strategies traditionally associated with department and specialty retail stores.

Using high-efficacy luminaires coupled with a layered lighting strategy produces dynamic lighting with accents and feature display illumination while meeting or exceeding code compliance and energy targets. In some applications, total LPD may actually be lower than that of the LPD used to produce bright, uniform lighting.



Accent and Display Lighting Strategies
Photographs reproduced with permission of Whole Foods Market

EL7 *Casework Lighting* (Climate Zones: all)

Casework lighting is not included in the tabulation of the LPD as long as it is integrated into the casework and is installed by the casework manufacturer. Lighting for casework must remain sensitive to the energy goals of the space. Strong consideration should be given to energy-efficient or low-energy solutions, including linear fluorescent, fiber optic, and linear LED sources.

EL8 *Office Task Lighting* (Climate Zones: all)

If the space planning recommendations in EL2 are followed by locating office spaces in the daylight zones, minimal if any task lighting should be needed during daylight hours. In areas next to windows, task lights should be evaluated on a needs basis and need not be automatically installed. Connect all task lights to plug strips that have integrated local occupancy sensors to turn the lights off when the space is unoccupied.

Educate employees on the need to turn off task lights when they leave the space and periodically educate employees on the goal of saving energy by turning off task lights. Confirm that task lights are controlled and are turned off during daylight hours and when the occupant leaves the space during nondaylight hours.

EL9 *Light-Colored Interior Finishes* (Climate Zones: all)

Higher surface reflectances on ceilings, walls, and floors may increase store visibility through the front windows and will increase lighting levels, perceived brightness, and energy performance within the space. However, higher reflectance may not conform to a retailer's brand image. Energy savings outlined in this Guide are based on reflectances of 80-50-20 (ceiling-wall-floor). If the reflectances are lower, then additional attention to the ambient lighting energy requirements may be necessary. Avoid direct lighting of specular surfaces (mirrors, glass, polished metals, or polished stone) in customer areas, if possible; otherwise, carefully consider the reflected light component and its effect on the customer.

An 80%+ ceiling and 70%+ wall reflectance is preferred in daylight zones (see DL2). Reflectance values are available from paint and fabric manufacturers. Reflectance should be verified by the QA provider.

In addition, take the shape and finish of the ceiling into account. A flat painted or acoustical tile ceiling is the most efficient at reflecting light, making the space feel brighter; sloping ceilings and exposed roof structures, even if painted white, may significantly reduce the effective ceiling reflectivity, making the space feel darker.

Dark colors absorb light, making the space feel darker. To maximize interreflections and maintain higher light levels, color should be used as an accent and large colored surfaces should be avoided.

EL10 *Color Rendering Index* (Climate Zones: all)

The color rendering index (CRI) is a scale measurement identifying a lamp's ability, generally, to adequately reveal color characteristics. The scale maximizes at 100, with 100 indicating the best color-rendering capability. It is recommended that lamps specified for the ambient, tasks, and accent lighting of merchandise and service areas within customer spaces have a CRI of 80 or greater to allow the consumer to effectively examine the color component of a product. Use of high-color-rendering light sources may also allow the lighting design to foster the required visual acuity with lower footcandles, which should result in a lower LPD.

EL11 *Color Temperature* (Climate Zones: all)

Color temperature is a scale identifying light source relative warmth or coolness—the higher the color temperature, the bluer the source. There are preliminary studies showing that higher-color-temperature light, in the 5000 K range instead of the 3500 K range, may provide better visual acuity; however, 5000 K lamps may produce an artificially cool-looking building at night. The higher 5000 K color temperature will also match the daylight from windows and skylights more closely than the lower 3500 K color-temperature sources.

Use 3000 K, 3500 K, or 4100 K light sources, as these color temperatures are available in multiple light sources and have gained customer recognition and acceptance. The decision as to which lamp color or multiple of lamp colors is used will be determined by merchandise objectives, brand identity, and operational considerations. For maximum energy utilization, use the highest-efficacy sources in each color temperature. Operational considerations may include creation of a purchasing plan to buy only one color-temperature lamp to maintain color consistency during spot and/or group relamping.

EL12 *Linear Fluorescent Lamps and Ballasts* (Climate Zones: all)

To achieve the LPD recommendations in Chapter 4, high-performance T8 lamps and high-performance electronic ballasts are used for general lighting. The use of standard T8 and energy-saving T8 lamps will result in lower ambient light levels or an increased number of fixtures or lamps to achieve recommended light levels. While the modeling for general lighting is based on high-performance T8 lamps and ballasts, under some conditions, similar performance may be obtained with T5 and T5/HO and LED designs.

T8 high-performance lamps. High-performance T8 lamps are defined, for the purpose of this Guide, as having a lamp efficacy of 90+ nominal lumens per watt (LPW), based on mean lumens divided by the cataloged lamp input watts. Mean lumens are published in lamp catalogs as the reduced lumen output that occurs at 40% of the lamp's rated life. High-performance T8s also are defined as having a CRI of 81 or higher and 94% lumen maintenance. The high-performance lamp is available in 32 W rapid start and 30 W, 28 W, and 25 W instant start lamps. Table 5-3 lists the average mean LPW of the commonly manufactured 4 ft T8 lamps.

Ballasts. Specify and install only National Electrical Manufacturers Association (NEMA) Premium ballasts. The NEMA Premium mark identifies ballasts that meet performance specifi-

Table 5-3 4 ft T8 Lamps Meeting the 90+ Mean LPW

T8 Lamp Description	Watts	Lumens		Mean LPW	Color Temperature, K
		Initial	Mean		
F32T8/RE80/HP	32	3100	2937	92	3000, 3500, 4100
F32T8/RE80/HP	32	3008	2848	89	5000
F32T8/25W/RE80	25	2458	2344	94	3000, 3500, 4100
F32T8/25W/RE80	25	2350	2241	90	5000
F32T8/28W/RE80	28	2725	2599	93	3000, 3500, 4100
F32T8/28W/RE80	28	2633	2509	90	5000
F32T8/30W/RE80	30	2850	2717	91	3000, 3500, 4100
F32T8/30W/RE80	30	2783	2653	88	5000

cations for the most energy-efficient T8 ballasts available from ballast manufacturers. Generally, a NEMA Premium ballast will use 5% less energy than a standard electronic ballast.

Ballast factor (BF) is a measure of the relative light output of the ballast. A BF of 1.0 means that the ballast is driving the lamp to produce 100% of the rated lamp lumens. Light output and wattage are related—the lower the BF the lower the wattage and the lower the light output. Normal BF ballasts are in the 0.85 to 1.0 range, with most at 0.87 or 0.88. Low-BF ballasts, with BFs below 0.85, can be used to reduce the light output and wattage of the system when the layout of the fixtures will overlight the space. High-BF ballasts, with BFs above 1.0, can be used to increase the light output of the lamp in areas where the fixture layout will underlight the space—wattage will go up proportionally to the increased BF.

One-lamp, three-lamp, and four-lamp ballasts may be used but should have the same or better efficiency as the two-lamp ballast. Dimming ballasts do not need to meet this requirement.

The higher-output 3100 lumen lamps are visibly brighter than standard T8s. Using ballasts with a BF of 0.77 may provide more comfortable lamp brightness in direct luminaires where the lamp is visible without sacrificing efficiency.

Instant start ballasts. Instant start T8 ballasts provide the greatest energy savings options and the least costly option. Additionally, the parallel lamp operation allows one lamp to operate even if the other burns out.

> ***Caution:*** Instant start ballasts may reduce lamp life when controlled by occupancy sensors or daylight switching systems. However, even if the rated lamp life is reduced by 25%, if the lamp is off due to the occupancy sensor more than 25%, then the socket life (the length of time before the lamps are replaced) will be greater. If extended socket life is desired, consider programmed rapid start ballasts.

Programmed rapid start ballasts. Programmed rapid start ballasts use approximately 5% more power than instant start ballasts, but programmed rapid start ballasts are normally recommended on vacancy/occupancy-sensor-controlled lamps and daylight control zones due to increased lamp life and dimming capabilities. In this Guide, programmed rapid start ballasts are not used to achieve the LPDs in Chapter 4; however, they may be used as long as the LPDs in Chapter 4 are not exceeded.

> ***Caution:*** Using programmed rapid start ballasts will result in slightly higher power consumption with the same light level. The wattage and light levels will need to be reduced in other areas to meet the LPD recommendations in Chapter 4.

T5 lamps and ballasts. T5HO and T5 lamps have initial LPWs that compare favorably to the high-performance T8s. In addition, T5s use fewer natural resources (glass, metal, phosphors) than T8 systems with comparable lumen outputs. However, when evaluating the lamp and ballast at the mean lumens of the lamps, T5HO lamps perform more poorly. On instant start ballasts, high-performance T8s are 13% more efficient than T5s. In addition, since T5s have higher surface brightness and caution should be used when specifying open-bottom fixtures, it may be difficult to achieve the LPD recommendation in Chapter 4 and maintain the

Table 5-4 4 ft T5/T5HO Lamps Meeting the 90+ Mean LPW

T5 Lamp Description	Watts	Lumens		Mean LPW	Color Temperature, K
		Initial	Mean		
F54T5HO (energy saver)	47–49	4800–5000	4410–4750	94–97	3000, 3500, 4100
F54T5HO (energy saver)	47–49	4600–4850	4230–4625	90–94	5000
F28T5	28	2900	2660–2750	95–98	3000, 35000, 4100
F28T5	28	2750–2840	2530–2641	90–94	5000
F28T5 (energy saver)	25–26	2900	2660–2750	102–110	3000, 3500, 4100

desired light levels using current T5 technology as the primary light source. Table 5-4 lists the average mean LPWs of the commonly manufactured 4 ft T5/T5HO lamps.

EL14 *Light-Emitting Diode (LED), also referred to as* Solid-State Lighting *(SSL)* (Climate Zones: all)

An LED is a solid-state semiconductor device that can produce a wide range of saturated colored light and can be manipulated with color mixing or phosphors to produce white light. To achieve the LPD recommendations in Chapter 4, LEDs may be used for ambient, task, and accent lighting.

White-light LED sources should be carefully evaluated for use in the lighting of retail merchandise, as color-rendering and color-temperature capabilities can vary widely by manufacturer. Products are now available that allow the adjustment of the intensity and the white color temperature, providing dynamic flexibility to the retail environment. Efficacy of high-color-rendering white LEDs is often less than that of T8 and T5 linear fluorescents. However, LED modules are available with a wide range of lumen packagers and optics, allowing LED luminaires to perform as efficiently as, and in some cases better than, T8 and T5 fluorescent luminaires. The most recent offerings of LED sources do perform as well if not better compared to lower-wattage CMH lamps and will significantly outperform most incandescent accent lighting.

There are current LED offerings that meet the ambient lighting requirements of spaces in the grocery model developed for this Guide. However, the ambient lighting for the model is based on T8 and T8 troffers and strip-light luminaires with higher efficacy than similar-footprint LED luminaires. LED luminaries, however, can be used effectively for casework, task, and display lighting. In applications and spaces where smaller-footprint downlights are desired, LED luminaires are the preferred option. Effective accent and wall-washing strategies can be achieved using LEDs, but intensity, color, and efficacy must be reviewed thoroughly. Single-color LEDs are well suited to creating interesting visual effects, as they produce color more efficiently than filtering other white light sources with relatively low-e consumption.

Careful consideration should also be given to maintenance issues. LED lamp life can offer advantages over other sources but does vary depending on the design of the LED module and the environment in which the LEDs operate. In some cases, the LED module cannot be replaced as a single lamp, like an incandescent source, for example; often the entire control board will have to be removed and replaced.

EL15 *Occupancy Sensors* (Climate Zones: all)

Use occupancy sensors in all areas where spaces may be unoccupied for significant time durations. Occupancy sensors are especially effective energy-saving tools in back-of-house spaces that are only occasionally occupied and on sales floors of grocery stores with extended operating hours. The greatest energy savings are achieved with manual ON/automatic OFF vacancy sensors or automatic ON to less than or equal to 50% power occupancy sensors. This

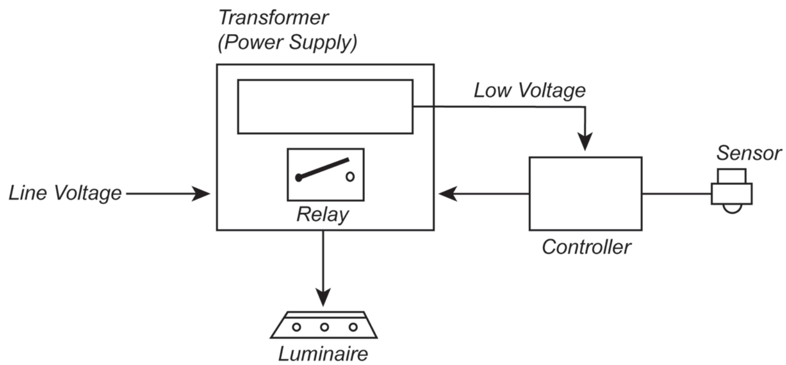


Figure 5-16 (EL15) Occupancy-Sensing Control

avoids unnecessary operation when electric lights are not needed and greatly reduces the frequency of switching. It should not be possible for the occupant to override the automatic OFF setting of the vacancy or occupancy sensor. Unless otherwise recommended, factory-set occupancy sensors should be set for medium to high sensitivity and a 15 min time delay (the optimum time to achieve energy savings without excessive loss of lamp life). Review manufacturers' data for proper placement and coverage. Figure 5-16 shows a typical occupancy-sensing control setup.

The two primary types of occupancy sensors are infrared and ultrasonic. Infrared sensors are line of sight and should not be used in rooms where the user cannot see the sensor (e.g., storage areas with multiple aisles, restrooms with stalls). Ultrasonic sensors can be disrupted by high airflow and should not be used near air duct outlets.

Caution: Confirm that the occupancy sensor is set to manual ON operation during installation. Many manufacturers ship sensors with the default setting of automatic ON.

Offices, break rooms, and other back-of-house rooms in daylight zone. In private offices, break rooms, and other back-of-house rooms in the daylight zone, dual-circuit infrared wall box sensors with integrated daylight override should be preset for manual ON/automatic OFF operation or auto ON to 50% (auto ON to 50% is achieved by having the first circuit set to auto ON and the second circuit set to manual ON). In rooms not in the daylight zone, use manual ON/automatic OFF without daylight override.

Other areas. In nondaylighted areas, ceiling-mounted occupancy sensors are preferred.

EL16 *Multilevel Switching* (Climate Zones: all)

For effective after-hours reduced lighting levels, specify luminaires with multiple lamps to be factory wired for inboard-outboard switching or inline switching. The objective is to have multiple levels of light uniformly distributed in the space. Avoid checkerboard patterns of turning every other fixture off in medium and large spaces. In open office and large open areas, avoid nonuniform switching patterns unless different areas of the large space are used at different times or for different functions.

EL17 *Exit Signs* (Climate Zones: all)

Use LED exit signs or other sources that consume no more than 5 W per face. The selected exit sign and source should provide the proper luminance to meet all building and fire code requirements.

EL18 *Light Fixture Distribution* (Climate Zones: all)

For high-ceiling sales and storage applications, use luminaires with substantially direct optics. These luminaires may use reflectors to focus light downward and louvers to minimize glare when used on the sales floor. For office/conference and other nonsales areas, use high-performance lensed 2 × 4 fluorescent or LED luminaires.

Though extensive use of totally indirect luminaires or recessed direct-indirect (basket-type) fixtures is a design option, these fixtures may not achieve desired light levels and still meet the LPD goal from Chapter 4 and, if not properly used, can create inherent brightness/contrast problems. Although it is not the recommendation of this Guide, some retailers still may elect to use them for ambient lighting. If used, it is very important to exercise proper care and attention to ensure that luminaires with high efficiency and superior optical performance are selected. When used for general lighting, fixture efficiency of at least 70% is recommended. Proper placement of luminaires, including attention to maintaining recommended uniformity, is also very important.

EL19 *Overhead Glare Control* (Climate Zones: all)

Specify luminaires that have good glare control and that are properly shielded for customer comfort. In sales areas, use fixtures with cross louvers that shield the view of the lamp at angles greater than 45°. In storage areas, open fixtures may be used, but the upper reflector should allow 10% uplight. In office/conference and other nonsales areas, avoid T5 lamps in open-bottomed fixtures.

For glare reduction in sales areas, use fixtures with semispecular or specular cut-off cones or louvers. Direct/indirect luminaires and indirect luminaires and architectural lighting are low-glare options for sales areas. However, these options may not achieve desired light levels and still meet the LPD goal from Chapter 4. In storage areas, use larger fixtures that spread the lights over a bigger area rather than smaller fixtures with the same number of lamps. As an example, use an 8 ft long fixture with two lamps in a cross section rather than a 4 ft long four-lamp cross-section fixture. Luminaires with solid low-transmittance lenses, such as “milk white” lenses, should be avoided, as they significantly reduce luminaire efficacy.

SAMPLE DESIGN LAYOUTS FOR GROCERY STORES

The watts-per-square-foot recommendations for LPD, shown in each recommendation table in Chapter 4, represent the individual LPD for each space type, but the base LPD may be combined into one average LPD for the building. However, the additional LPD allowances described in EL5 may not be combined and must be used for the specified application. The example designs described in EL20 to EL27 offer *a way*, but *not the only way*, that this watts-per-square-foot limit can be met.

The examples that follow are based on national average building space distribution. No building is average, and your building will have a different space allocation. Follow the recommendations below and adjust the space allocation to match your building.

EL20 *General Lighting in Grocery Sales Areas* (Climate Zones: all)

General lighting at 1.15 W/ft^2 provides the base level of lighting for the merchandise. Spill light from the merchandise general lighting will provide adequate lighting for the circulation paths. Included in the base allowance is decorative/focus lighting at the end of aisles and in produce/deli and other specialty sales areas—see EL21for more information.

When/where grocery sales include vertical or high-stack merchandise, attention to vertical illumination is required. The required vertical lighting may be provided by luminaires built into cases or gondolas, luminaires with optics that direct light onto the vertical plane, or general lighting fixtures strategically placed to provide vertical illumination on the merchandise. The optimal location for the general lighting is centered in each aisle and mounted approximately 10 ft above the floor. A simple solution to provide general lighting is to use continuous rows of linear LED or two-lamp fluorescent fixtures. Alternatively, use manufacturers’ photometric files and calculate noncontinuous spacing of light fixtures. In aisles with refrigerated cases with internal lighting it is not necessary to have a continuous row of light fixtures. In these aisles, space the fixtures 8 to 12 ft on center down the middle of the aisle.

General lighting can be provided by a number of different types of fixtures.

- *Direct fixtures, open lensed, and parabolic* will provide the highest footcandles for the space and will provide some positive shadowing on the product. However, if the merchandise layouts are expected to change or be reoriented from the initial installation, the fixtures may be misaligned to the merchandise layouts. Consider an alternative layout with the fixtures running perpendicular to merchandise layouts. (See Figure 5-17.)
- *Indirect fixtures, pendant, and recessed indirect* will not be merchandise-layout dependent but may look better running perpendicular to the merchandise aisles. Indirect fixtures tend to provide a flat lighting effect and may have the potential to draw the customers' attention away from the product. In such instances, task-directed and/or accent light illumination may be needed. (See Figure 5-17.)

In produce, deli, and other specialty sales areas, use a combination of low general lighting and high accent lighting (Figure 5-18). General lighting can be reduced to approximately 0.20 W/ft^2, and the remaining 0.95 W/ft^2 of the 1.15 LPD allowance shown in the recommendation tables in Chapter 4 can be used to highlight the product. See EL21 for more information.

EL21 *Accent Lighting in Merchandise Sales Areas* (Climate Zones: all)

See additional recommendations in EL5 for accent lighting. Use LED task and accent lighting to highlight key merchandise locations or vignettes to "feature display" light levels (three to ten times the general merchandise lighting level in the area of the display). The use of accent lighting to highlight *all* merchandise does not create the proper contrast ratios and should be avoided. (See Figure 5-18.)

Highlight merchandise at windows, if present, to ten times the general merchandise lighting level to attract customers from outside the store. Window display lighting should be switched down to three times the general merchandise lighting level at night to help with eye adaptation when entering and exiting the store. It is important to incorporate sun shades, canopies, or another form of sun and glare control at windows exposed to the outdoors. When this cannot be accomplished, it may be better to turn off lights near windows during daylight hours. Trying to accent on top of extremely high ambient light levels, as associated with midday sunlight, causes excessive energy use as well as potential damage to merchandise.

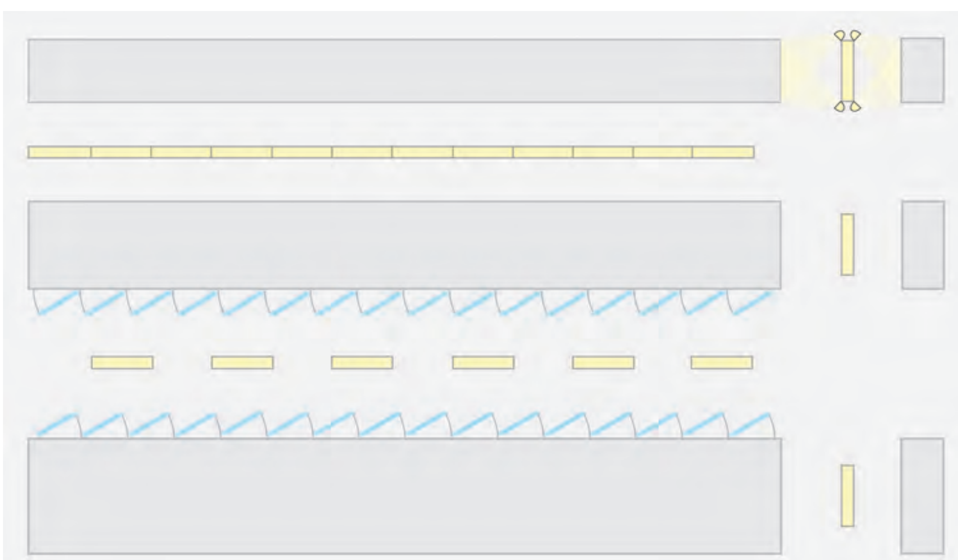


Figure 5-17 (EL20) Layout for Lighting in Merchandise Sales Area

EL22 *Perimeter Lighting in Merchandise Sales Areas* (Climate Zones: all)

Follow the recommendation in EL5 for perimeter lighting. Use directional, focal, or wall-washer luminaires to highlight key wall locations as well. It is especially important to highlight the back wall to draw customers' attention all the way to the back of the store. Perimeter lighting techniques also can be applied to fitting rooms as a tool to provide the desired vertical illumination at mirrors.

EL23 *Casework Lighting in Merchandise Sales Areas* (Climate Zones: all)

Refrigerated casework and gondola lighting is *not* included in the tabulation of LPDs as long as it is integrated into the casework and is installed by the casework manufacturer. Follow the recommendation in EL2 for external casework and gondola task and accent lighting LPDs above the base power allowance. Use accent or focal lighting (see Figure 5-19) to highlight key merchandise to "feature display" light levels (three to ten times the general merchandise lighting level). Focal lighting is also well suited to provide additional illumination for task areas requiring light levels above those of typical sales area illumination. Lighting for casework must remain sensitive to the overall energy goals of the space and should be automatically controlled to shut off during nonbusiness hours. Strong consideration for internal display lighting should be given to energy-efficient or low-energy solutions, including linear fluorescent and LED sources.

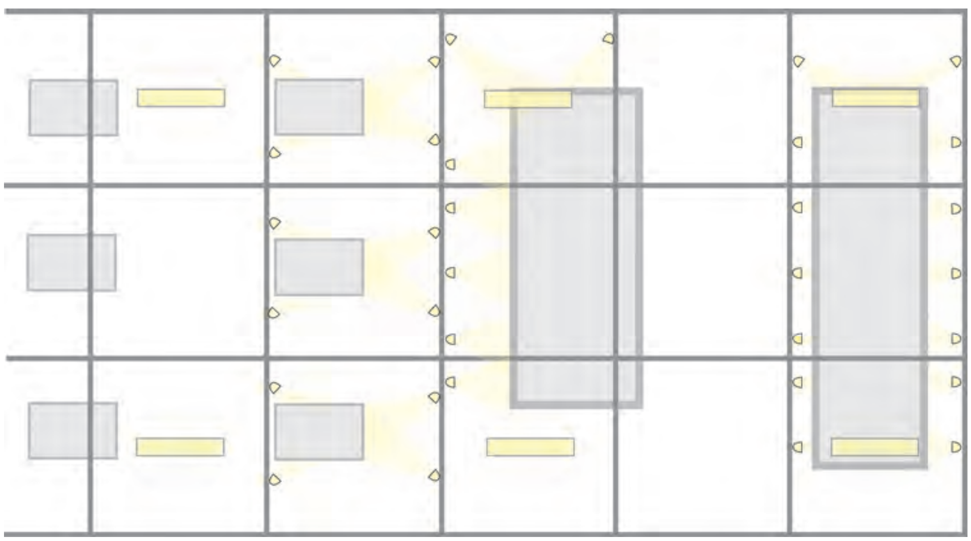


Figure 5-18 (EL20, EL21) Layout for Lighting in Specialty Sales Area

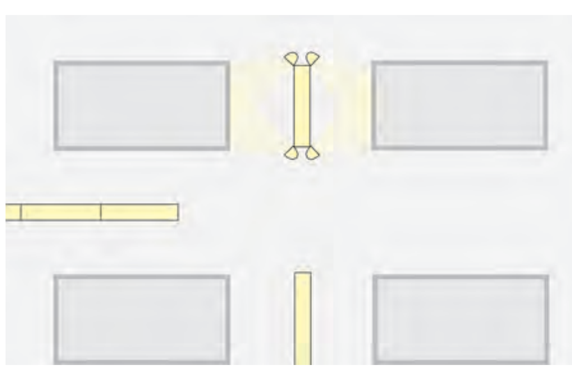


Figure 5-19 (EL23) Accent Lighting in Merchandise Sales Area

Caution: Most new refrigerated cases come installed with LED lighting as required by codes and standards. However, the quality and output of the LEDs is not a given. Furthermore, a comprehensive lighting control coupled with the LED lights is not mandatory. Store owners and designers desiring to maximize the performance and energy savings of their refrigerated cases should predetermine the quality and performance of the LED lighting in the cases. They also should use motion sensors and lighting controls to shut off or step dim (to a low 30% to 50% level light output) when there are no occupants near the cases. Additional information on lighting requirements for refrigerated display cases is detailed in RF22.

LED Lighting Color, Optics, and Characteristics

LED color and optics present new design challenges, especially when applied to grocery lighting, where color quality and light-source characteristics can either enhance or deter merchandise sales.

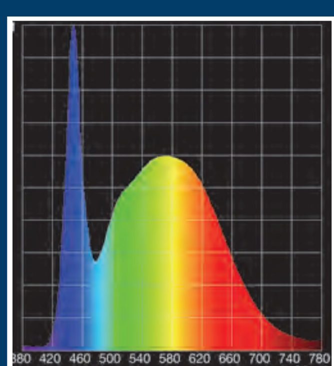
Color Rendering and Quality

There are a number of differences that rule out using photometric comparisons of lighting performance between LEDs and other lamp sources such as fluorescent or metal halide. While a higher color rendering index (CRI) for a light source indicates improved color quality when applied to traditional sources such as fluorescent or metal halide, this same CRI matrix can fall short when evaluating the color quality of LED lighting.

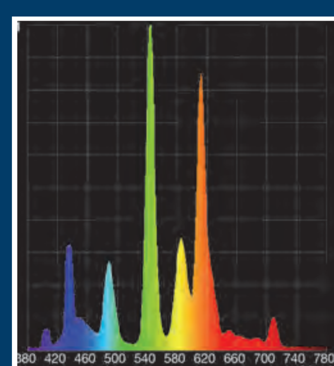
The first eight reference test colors (R1 through R8) in the following figure are used when establishing the CRI of a light source under a traditional CRI evaluation matrix. When evaluating the CRI of an LED light source, however, test colors R9 trough R14 are important to the evaluation process.

Chromaticity is another component of color quality. When evaluating chromaticity, keep in mind that the differences in spectral distribution can result in the visual appearance of an LED white light that is different than a fluorescent or metal halide source even though their chromaticity has the same kelvin rating. The spectral distribution of LED white light has a fluid spectral curve as opposed to the spikes found in fluorescent and metal-halide spectral distributions.

Test Colors for Calculating CRI



White LED Spectral Curve



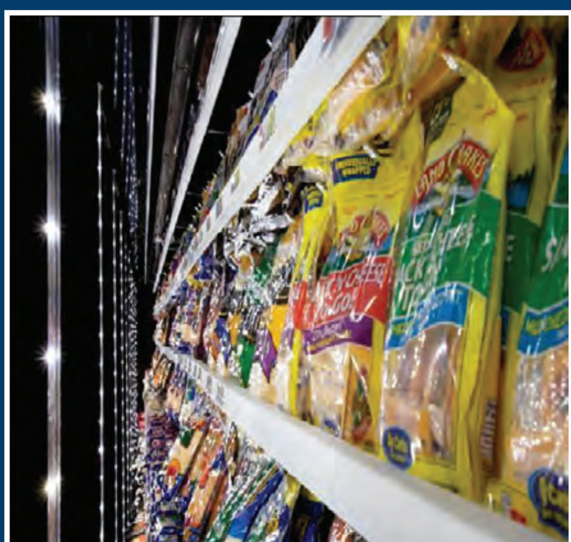
Fluorescent Spectral Curve

Figures reproduced with permission of PLANLED Inc.

In addition, the correlated color temperature (CCT) of white LED is subject to variations that cause LED modules to go through a "binning process" to set the limits of acceptable variations within a given chromaticity target. Tighter binning ensures white light uniformity; however, tight binning comes at a premium cost, and high-quality LEDs command a premium price. Therefore, white LED luminaires and lamps with these characteristics should be limited to retail applications where maximum color quality is important to the enhancement of the sales environment.

Shadows and Sparkle

LED light sources without diffuser lenses can create a sparkle effect on merchandise due to the point source characteristics. Shadows and sparkle on fresh fruits, vegetables, and some bottled goods provides enhanced rendition of the merchandise versus the same product illuminated with defuse light such as fluorescent. However, LED lighting can create unwanted shadows and sparkle (glare) when illuminating packaged and boxed goods. In these applications, using an LED luminaire with a diffuser lens will create a softer (defuse) light without harsh shadows and glare on packaging, similar to that of fluorescent lighting.



LED Sparkle

Diffuser Lens over LED Strip Light

Photographs reproduced with permission of NREL, credit Pat Corkery (left) and Hera Lighting LP (right)

Ultraviolet Radiation

While LED white light has the potential to emit ultraviolet (UV) radiation, the UV present in the LEDs is much lower than that in fluorescent and metal halide lamps. In addition, the small amount of UV produced by an LED is completely absorbed by phosphors used to produce white light. Heat within the beam of light produced by LED technologies is also significantly less than heat produced by other light sources.

High light levels and UV radiation can cause fading in products and shorten the shelf life of fresh food products. With no UV and minimal heat in the beam, LED lighting is ideal for illuminating these items that are susceptible to UV and heat damage. Use of LED lighting may contribute to longer shelf life and less damage to fresh food products.

EL24 *Lighting for Deli and Baked Goods Areas* (Climate Zones: all)

While T8 and CFL luminaires have been the primary light sources for deli/baked goods service areas and on-site kitchens, advances in LED lighting provide opportunities for LED illumination within these spaces. LED lighting offers long life and added control options and can be fine-tuned easily to meet task illumination in these spaces while lowering the overall LPD of the space. LED is the preferred source for undercounter and shelf illumination as well as those locations where small-aperture downlights are desired. With a one-to-one exchange of LED luminaires for fluorescent luminaires, 20% to 80% lower luminaire lighting power for equivalent useful light output can be expected.

Caution: LEDs subject to high temperatures can experience short life via premature failure. This is a concern in food service areas and kitchens when LED lighting is used near warming ovens, broilers, grills, or similar cooking and backing equipment. To minimize LED failure because of excessive heat within and close to this equipment, heed the following guidelines:

- Avoid placing LED luminaires directly above broilers and inside the hottest zone of hoods.
- Place luminaires a foot or two either side of the hottest area in the hood to improve luminaire performance and life.
- Provide adequate cool air flow over the luminaire or consider LED luminaires with active cooling built into the fixture.
- Review performance characteristics and performance parameters of the LED luminaire to assure that the product will perform in this environment.

These considerations are especially important for luminaires used in deli/baked goods service areas and on-site kitchens, as health department codes usually require lensed, washable, sealed luminaries, which may run hotter than open luminaires.

Comprehensive lighting controls coupled to LED lighting can provide additional energy savings, enhance lighting quality, and maximize visual acuity, which results in improved task performance. Add vacancy sensors in areas or zones that are not used full time. The sensors may turn off lighting completely when the areas are unoccupied or, at minimum, step it down to 20% to 30% of full light output when the spaces are vacant. Specifically, use vacancy sensors with the following:

- To turn off undercounter linear LED task lighting when area is not in use
- To turn off LED task lighting under range hoods when not in use
- To dim LED downlights or decorative pendants for task illumination on the service counter when occupants are not present
- To dim CFL or LED decorative lighting for dining areas when areas are unoccupied
- To dim deli general lighting using LED, T5, or T8 fluorescent luminaires when area is unoccupied or closed

Other control options include the following:

- Time clocks that turn off lights when the areas listed above are closed
- Tunable lighting controls that set and maintain target light levels thereby using less energy when daylight is present in the space

The target lighting for deli/baked goods service areas and on-site kitchens is 20–30 average maintained footcandles for general work area lighting with approximately 50–75 horizontal and 50 vertical footcandles provided on the detailed task work surfaces, such as food handling, preparation backing, and cake decorating. Sample conceptual layouts for lighting deli/baked goods service areas and on-site kitchens are shown in Figures 5-20 and 5-21.

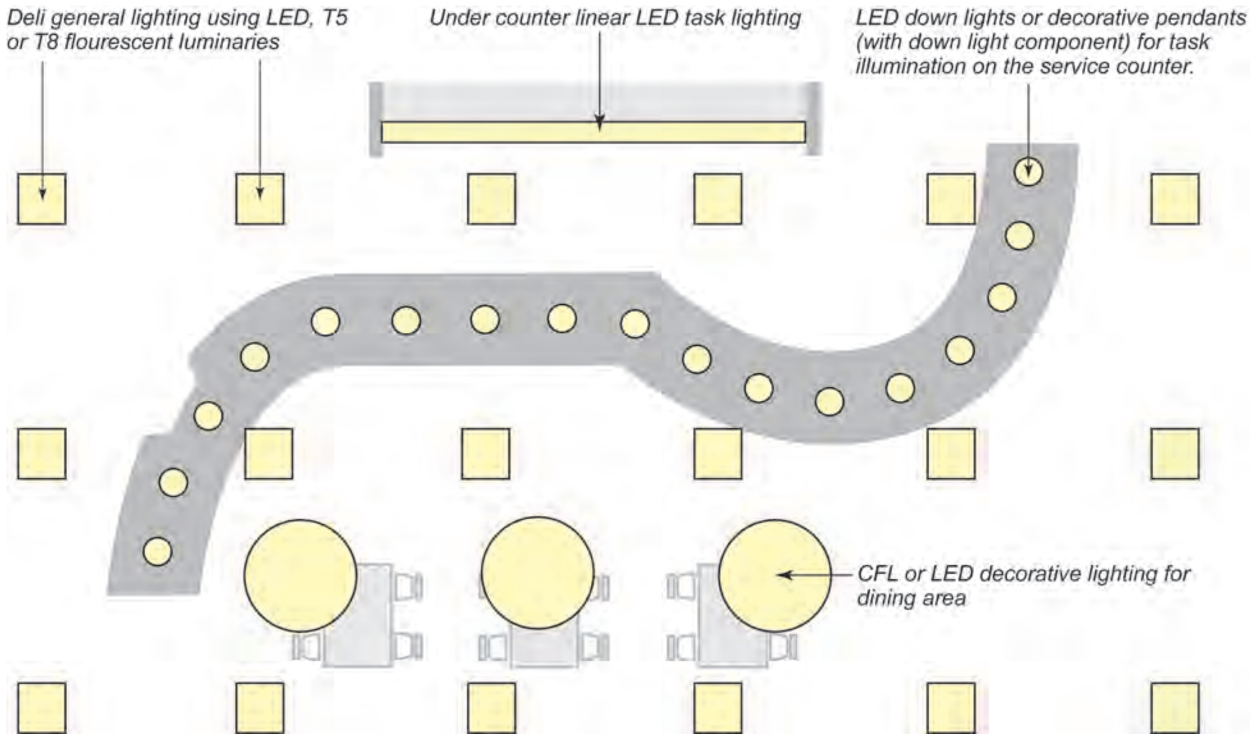


Figure 5-20 (EL24) Conceptual Layout for Deli Service Counter and Dining Area

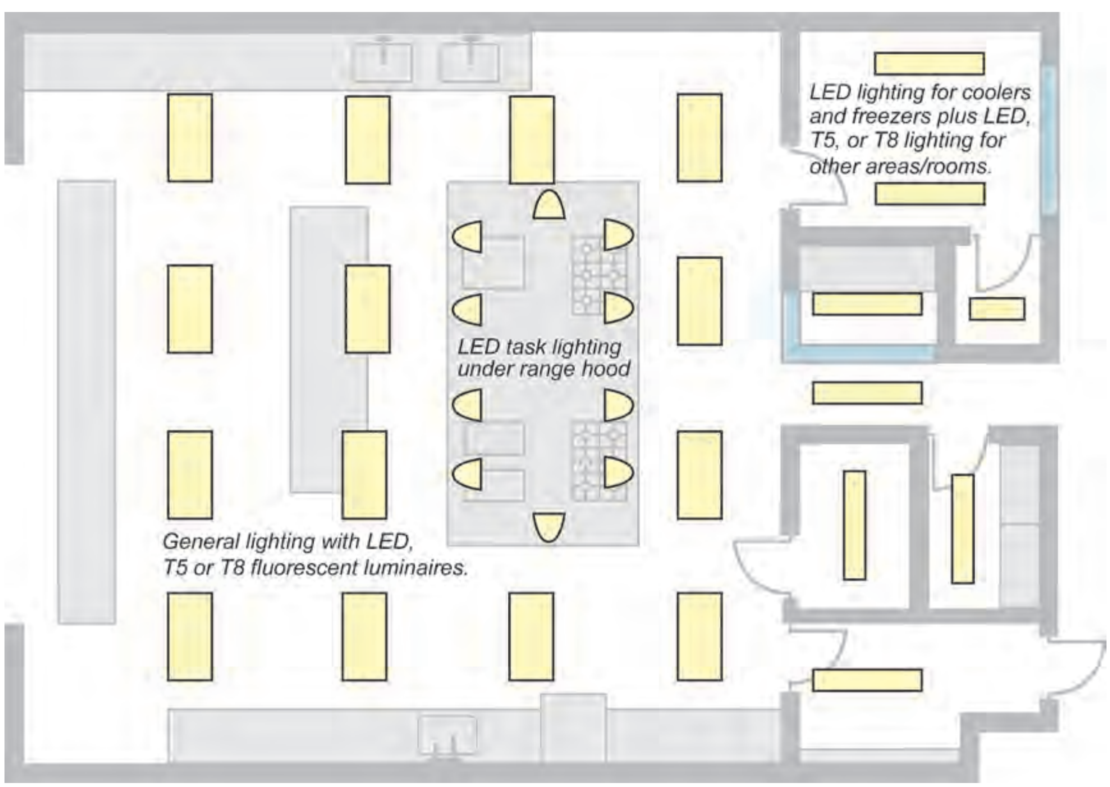


Figure 5-21 (EL24) Conceptual Layout for Grocery In-House Kitchen

EL25 *Lighting for Walk-In Coolers and Freezers* (Climate Zones: all)

LED lighting provides significant energy savings for walk-in coolers and freezers. LED lights perform well in cold environments and provide improved lighting performance in addition to energy savings. Lighting in walk-in coolers and freezers should be limited to 0.80 W/ft^2. This can be accomplished by using an LED fixture using approximately 50 W for every 64 ft^2.

Walk-in boxes found in grocery stores fall into several categories of small, medium, and large coolers or freezers, which are described in RF18. The majority of grocery stores use small or medium walk-in boxes with or without customer access doors, while large walk-in boxes are usually located in large big-box warehouse stores. Lighting equipment for small and medium walk-ins has traditionally been cold-start-compliant energy-efficient T8 and T5/HO lamps/ballasts, and the use of controls with these fluorescent systems has been minimal. Equipment in large walk-ins has traditionally been metal halide luminaires with full-time ON operation and a time clock based ON/OFF daily cycles.

Switching to LED luminaires can result in up to a 50% lower LPD and produce less heat, which reduces the load on the refrigeration equipment. Because small walk-in box ceilings are typically only 8 to 10 ft high, LED luminaires with wide-distribution optics are recommended for proper light distribution. As an alternative, use lower-wattage luminaires with closer luminaire spacing to provide proper light distribution. The adverse environment of refrigerated coolers and freezers requires LED luminaires that are designed to function within these environments. Vapor-tight luminaires that have a NEMA IP65 rating or Underwriters Laboratories (UL) wet location listing are recommended for these applications (NEMA 2004; UL 2004).

Comprehensive lighting control has not been a strategy for most cooler/freezer cases until very recently, but by coupling LED lighting to motion sensors and occupancy/vacancy control devices, maximum energy savings are realized. Specifically, use vacancy sensors with the following:

- To dim linear LED lighting at customer access doors facing the selling floor when occupants are not present
- To turn off LED ceiling-mounted luminaires for task illumination when restocking walk-ins when the walk-in is unoccupied

Other control strategies include the following:

- Time clocks that turn off lights in walk-in boxes when the store is closed
- Door trigger switches or motion sensors that circuit luminaires off when walk-in box is unoccupied

Target illumination for walk-in coolers/freezers is 10 average horizontal and 3 average vertical footcandles maintained for back-of-house storage and restocking, with approximately 50 to 75 fc for sales area presentation casework feature lighting, which is typically vertical illumination. Sample conceptual layouts for a variety of walk-in cooler/freezer lighting options are shown in Figures 5-22 through 5-25.

EL26 *Offices* (Climate Zones: all)

The target lighting in private offices is 30 average maintained footcandles for ambient lighting with approximately 50 fc provided on the desktop by a combination of the electric lighting and daylighting. Supplemental task lighting is only required during nondaylight hours. A sample layout for a private office is shown in Figure 5-26. For office controls, specify manual ON or automatic ON to 50% occupancy sensors.

Private office plans should be limited to 0.80 W/ft^2. This can be accomplished by using an LED fixture using approximately 50 W or a two-lamp T8 fixture (with a low-ballast-factor ballast) for every 64 ft^2.

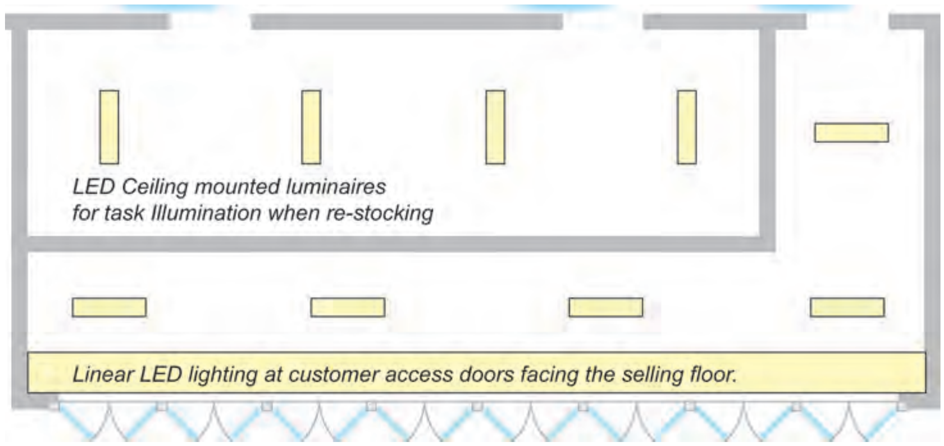


Figure 5-22 (EL25) Conceptual Layout for Back-of-House Walk-in Freezer and Walk-in Cooler with Sales Floor Customer Access Doors

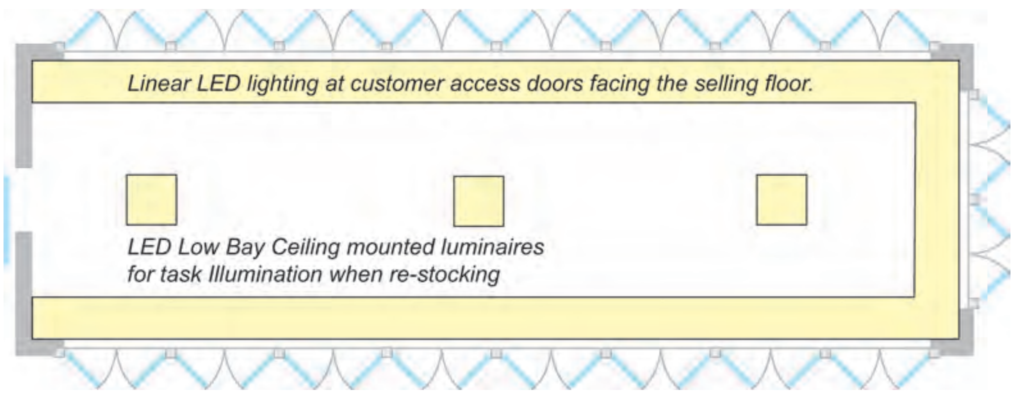


Figure 5-23 (EL25) Conceptual Layout for Large Walk-in Cooler/Freezer with Customer Access Doors

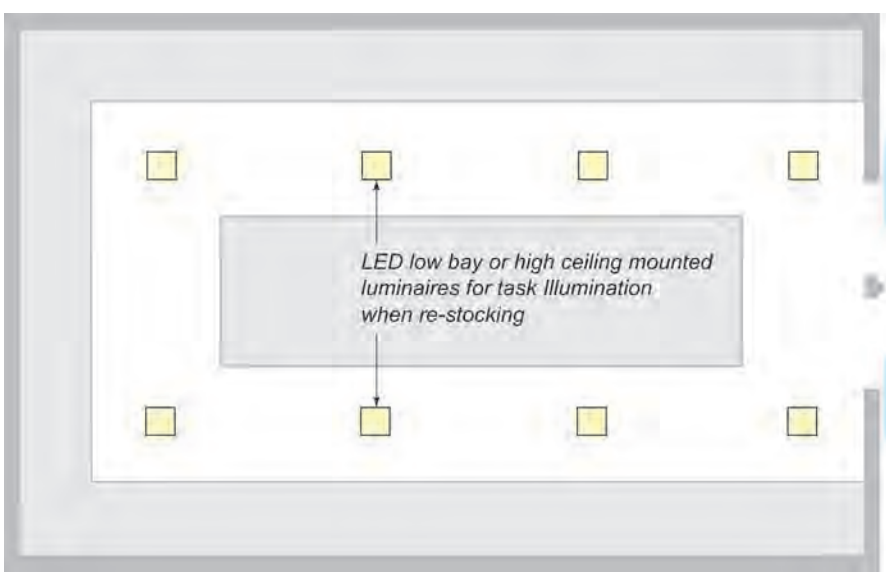


Figure 5-24 (EL25) Conceptual Layout for Large Big-Box Cooler/Freezer with High Ceilings

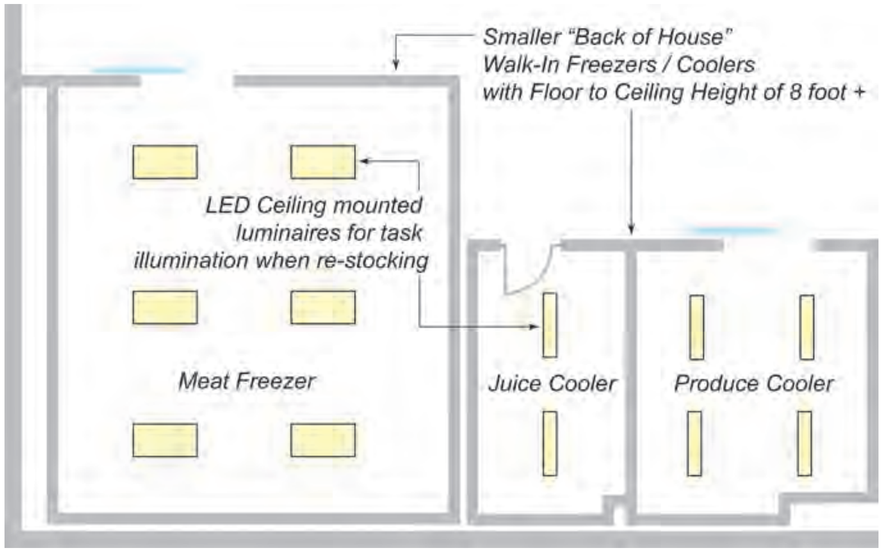


Figure 5-25 (EL25) Conceptual Layout for Small Walk-in Grocery Cooler/Freezer Modules

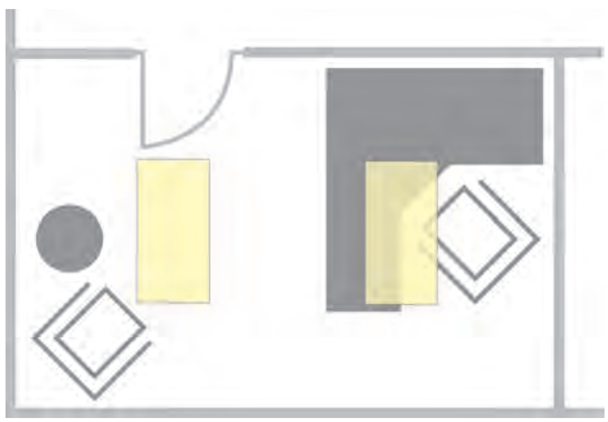


Figure 5-26 (EL26) Conceptual Layout for Office

EL27 Break/Meeting Rooms (Climate Zones: all)

The target lighting in conference rooms is 30–40 average maintained footcandles. Conference rooms serve multiple functions that may require lighting layers and multiple lighting systems. Collective use of multiple systems tends to result in higher LPDs. Therefore, use of lighting controls is important. Controls to consider are dimming controls and separate switching of multiple systems. Specify manual ON or automatic ON to 50% occupancy sensors to meet ASHRAE/IES Standard 90.1 (ASHRAE 2013) requirements.

Conference rooms should be limited to 0.80 W/ft^2. Figure 5-27 shows a typical conference room lighting layout. This can be accomplished by using an LED fixture using approximately 50 W or a two-lamp T8 fixture (with a low-ballast-factor ballast) for every 64 ft^2.

EL28 Warehousing/Active Storage Areas (Climate Zones: all)

The target lighting in storage areas is 20–25 average maintained footcandles. Storage areas account for approximately 15% of the floor area and should be limited to 0.65 W/ft^2, including circulation, which is equivalent to about one 50 W LED fixture or a two-lamp T8 fixture (with a low-ballast-factor ballast) for every 80 ft^2. Use occupancy sensors or timers where appropriate.

Figure 5-27 (EL27) **Conceptual Layout for Break/Meeting Rooms**

EL29 *Other Spaces* (Climate Zones: all)

Lighting in the remaining floor space is composed of various functions, including lighting corridors, restrooms, electrical/mechanical rooms, locker rooms, and others. Limit the connected lighting load in these spaces to no more than 0.80 W/ft^2, which is equivalent to about one 50 W LED fixture or a two-lamp T8 fixture (with a low-ballast-factor ballast) for every 64 ft^2. When/where possible, these spaces should be designed under the 0.80 W/ft^2 suggested LPD. Use occupancy sensors or timers where appropriate to achieve additional energy savings.

EL30 *Twenty-Four-Hour Emergency and Security Lighting* (Climate Zones: all)

Night lighting or lighting left on 24 hours to provide emergency egress needs when the building is unoccupied should be designed to limit the total lighting power to no more than 2% of the general LPD. It should be noted that most jurisdictions allow the application of occupancy-sensor controls on egress lighting to further reduce electricity associated with lighting an unoccupied building.

EXTERIOR LIGHTING

Good Design Practice

Per ASHRAE/IES Standard 90.1 (ASHRAE 2013), exterior LPDs are calculated using lighting zones. There are five zones as shown in Table 5-5. Most grocery stores fall into lighting zone 3.

Cautions: Calculate the LPD only for areas that you intend to light. For this Guide, areas that are illuminated to less than 0.1 fc are assumed to not be lighted and should not be counted in the LPD allowance. For areas that are intended to be lighted, design with a maximum-to-minimum ratio of illuminance no greater than 30 to 1. Therefore, if the minimum light level is 0.1 fc, then the maximum level in that area should be no greater than 3 fc.

For parking lot and grounds lighting, do not increase luminaire wattage in order to use fewer lights and poles. Increased contrast makes it harder to see at night beyond the immediate fixture location. Flood lights and wall-packs should not be used, as they cause hazardous glare and unwanted light encroachment on neighboring properties.

Limit poles to 20 ft mounting height and use luminaires that provide all light below the horizontal plane to help eliminate light trespass and light pollution.

Table 5-5 Exterior Lighting Zones

Lighting Zone	Description
0	Undeveloped areas within national parks, state parks, forest lands, rural areas, and other undeveloped areas as defined by the authority having jurisdiction
1	Developed areas of national parks, state parks, forest lands, and rural areas
2	Areas predominantly consisting of residential zoning, neighborhood business districts, light industrial with limited nighttime use, and residential mixed-use areas
3	All other areas
4	High-activity commercial districts in major metropolitan areas as designated by the local jurisdiction

EL31 *Exterior Lighting Power—Parking Lots and Drives* (Climate Zones: all)

Calculate only for paved areas, excluding grounds that are lighted to less than 0.1 fc. Per IES lighting recommendations, maintain 0.5 fc minimum (IES 2011). Limit exterior lighting power to 0.08 W/ft^2 for parking lots and drives in lighting zones 3 and 4. Limit exterior lighting power to 0.05 W/ft^2 in lighting zone 2.

Use parking lot fixtures with photosensor control so lights are not on during the day and an occupancy sensor that automatically reduces the power by at least 50% when no motion is detected. Set the time delay for occupancy sensors at a maximum of 10 min.

Caution: Parking lot lighting locations should be coordinated with landscape plantings so that tree growth does not reduce effective lighting from pole-mounted luminaires.

EL32 *Exterior Lighting Power—Walkways* (Climate Zones: all)

Exclude grounds that are lighted to less than 0.1 fc.

Limit exterior lighting power to 0.08 watts per linear foot (W/lf) for walkways less than 10 ft wide and to 0.16 W/ft^2 for walkways 10 ft wide or greater, plaza areas, and special feature areas in lighting zones 3 and 4.

Limit exterior lighting power to 0.07 W/lf for walkways less than 10 ft wide and 0.14 W/ft^2 for walkways 10 ft wide or greater, plaza areas, and special feature areas in lighting zone 2.

EL33 *Decorative Façade Lighting (Climate Zones: all)*

Avoid the use of decorative façade lighting. If façade lighting is desired, limit the lighting power to the following:

- 0.075 W/ft^2 in lighting zones 3 and 4 for the area intended to be illuminated to a light level no less than 0.1 fc
- 0.05 W/ft^2 in lighting zone 2 for the area intended to be illuminated to a light level no less than 0.1 fc

Program façade lighting to automatically turn off between the hours of midnight and 6:00 a.m. This does not include lighting of walkways or entry areas of the building that may also light the building itself.

Additional energy savings can be achieved by using light sources that are dimmable and coupled to motion sensors, as the lights can be set to a low-level output or turned off when there are no occupants in the area.

EL34 *Sources* (Climate Zones: all)

To meet the LPD recommendations and IES footcandle recommendations in EL31, all parking lot fixtures are assumed to be LEDs. All grounds and building lighting should use LED fixtures as well.

EL35 *Controls* (Climate Zones: all)

Use photosensors or astronomical time switches on all exterior lighting. If a building energy management system (EMS) is being used to control and monitor mechanical and electrical energy use, it also can be used to schedule and manage outdoor lighting energy use.

Turn off exterior lighting not designated for security purposes when the building is unoccupied by incorporating a time-clock control.

For walkways and plaza areas, incorporate an occupancy sensor control into each fixture or group of fixtures that automatically reduces the power by at least 50% when no motion is detected. Set the time delay for occupancy sensors at a maximum of 10 min.

All other exterior lighting (façades and building grounds) should also be automatically shut off at posted business closing time and remain off until posted opening time (and should not be on during daylight hours).

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PLUG LOADS

EQUIPMENT AND CONTROL GUIDELINES

PL1 *General Guidance* (Climate Zones: all)

To reduce connected wattage for plug load equipment, select equipment and appliances that are energy efficient. ENERGY STAR® rated equipment typically has significantly lower operational wattage and may include improved device sleep-mode algorithms (EPA 2015). Illuminated signage, security monitors, desktop computers, laptops, desk printers, fax machines, copying machines, refrigerators, microwave ovens, vending machines, coffee makers, and dishwashers are typical equipment types used in grocery stores that have ENERGY STAR ratings. There may be no ENERGY STAR ratings for some equipment. Compare efficiency levels for equipment not rated by ENERGY STAR, such as point-of-sale equipment (cash registers, demagnetizers, scanners, scales, and conveyor belts) and security cameras. Grocery retailers should specify energy-efficiency requirements for all equipment supplied by outside vendors, such as vending machines, beverage refrigerators, refrigerated cases and gondolas, and self-service kiosks. The ENERGY STAR website provides lists of covered equipment (EPA 2015).

PL2 *Sales Floor Plug Load Specification* (Climate Zones: all)

The first step in reducing sales floor plug load energy consumption is to reduce the connected wattage. As stated in PL1, retailers should select ENERGY STAR rated equipment whenever possible; if an ENERGY STAR rated product is not available, options should be evaluated and selected according to their efficiency.

A large portion of the connected plug load wattage in grocery stores is supplied by outside vendors; accordingly, it is essential that the grocery retailer provide specifications to ensure that vendor-supplied equipment is as energy-efficient and controllable as possible. Wherever possible, equipment should be specified to have automatic standby and/or load-managing features. Vending machines can be specified to be delamped (or at least to have occupancy-sensor-based lighting control and high-efficiency lamps); refrigerated and freezer cases, including beverage refrigerators, can be specified to have load-managing systems that optimize compressor cycles and control internal lighting based on occupancy. Self-service kiosks, such as price-check scanners, photograph printing machines, and video rental machines, should be specified to have automatic standby modes. When/where this option is not offered, equipment can be connected to sensor-controlled plug strips that will shut off equipment when not in use.

See PL7 for sales floor plug load control strategies.

PL3 *Office Plug Load Specification* (Climate Zones: all)

Laptop computers are designed to operate efficiently to extend battery life, with characteristics such as lower connected wattage and effective power management. Laptops with an ENERGY STAR rating should be selected. Desktop computers generally use significantly more energy and may not be necessary for most applications. A typical desktop computer, not including monitor, may consume 100 W; a comparable laptop, on the other hand, may consume only 30 W (Lobato et al. 2011). In applications for which laptop computers are not appropriate, mini desktops and thin clients can be substituted at comparable efficiency.

Printing services should be consolidated to minimize the number of required devices. Use of multifunction devices that provide printing, copying, and faxing capabilities reduces power demand from multiple devices.

Enable the ENERGY STAR power settings on the computer. For network computers, these can be set at a corporate level through the enterprise operating system policy. Device energy reduction recommendations include turning off the disk drive after 3 min, turning off the monitor after 5 min, and putting the computer into sleep mode after 15 min. The computer sleep mode may be adjusted depending on the recovery response needed. In no case should the sleep mode setting be longer than 4 h.

Many products today, such as coffee pots, are equipped with an automatic OFF feature. Units with such an energy-saving feature should be specified wherever possible.

See PL7 for grocery office plug load control strategies.

PL4 *Bathroom Fixture Plug Load Design Considerations* (Climate Zones: all)

Specify photovoltaic-powered faucet, soap dispenser, and toilet sensors for bathrooms. Specify efficient no-heat hand dryers (1500 W maximum).

PL5 *Security System Plug Load Specification* (Climate Zones: all)

Security system plug loads can be significant and should not be overlooked when seeking to reduce overall plug load energy consumption. High-efficiency LED liquid crystal display (LCD) models should be specified for security monitors.

Security cameras, both interior and exterior, and associated equipment can also represent a significant load and should be carefully selected. Select security cameras designed to minimize parasitic loads. Consider operational requirements and factor in the energy use of necessary functionality, such as zooming. In general, static, nonheated cameras will use the least amount of energy. If camera motion is required, carefully consider the energy use required by the motors to pan and tilt. If heaters must be used (most likely in exterior applications), ensure that they operate only as needed to prevent moisture from building on the camera lens. Using infrared cameras makes it unnecessary to light the viewing area. If infrared cameras are not appropriate for an application and lighting is needed, select low-light cameras and light the viewing area with low-illuminance, uniform lighting whenever possible to optimize lighting energy use and camera performance.

Occupancy-based control of security lighting is encouraged; the viewing area need not be illuminated unless there is motion to capture. Consider security lighting needs early in the design process to coordinate with architecture and site planning and reduce the need for security-specific lighting.

PL6 *LED Alternatives to Neon and Fluorescent Illuminated Signs and Graphics* (Climate Zones: all)

The baseline for signs and illuminated graphics plug load is fluorescent and neon/cold-cathode lighting, which operates during store hours and has only basic ON/OFF control coupled to a time clock.

LED light source applications for illuminated signage, graphics, and decorative elements can drastically reduce the energy consumption of these design elements. Furthermore, LED systems often provide additional benefits such as reduced maintenance requirements, lower-profile designs, and greater design flexibility. LED lighting is also conducive to comprehensive lighting control that allows for dimming, standby modes, and automated ON/OFF functions. LEDs can be used to backlight and/or edge-light illuminated graphics (images and signs) and can be inserted into diffuse flexible tubes to emulate neon or cold-cathode tubular lighting.

Used as direct replacements for fluorescent, cold-cathode, or neon lighting, LED-illuminated signs and graphics in static applications can produce desired design objectives with 40% to 90% less energy.

Another application of LED technology to consider is the use of dynamic liquid crystal displays (LCDs) with edge-lit LED lighting. The latest generation of 40–80 in. LED LCD monitors are very energy efficient, meeting or exceeding ENERGY STAR requirements. An average ENERGY STAR compliant LED LCD monitor will consume slightly less than one-third of the energy of an equivalent-sized fluorescent light box. The LED LCD monitor provides dynamic displays that may result in fewer units needed for a given message or visual effect, thereby producing significant additional energy savings. For example, the LED LCD monitor can serve as a sign as well as provide advertising and informational messages. With

Table 5-6 LED Lighting System Energy Savings

Type of Sign/Graphic	Base System LPD	LED Alternative LPD	Energy Saved
Surface-script neon/cathode signs	7.5–10.0 W/lf	2.5–3.5 W/lf	55% to 75%
Face-illuminated channel letters	15.0–20.0 W/lf	2.1–3.0 W/lf	85% to 90%
Indirect-illuminated channel letters	15.0–20.0 W/lf	2.1–3.0 W/lf	85% to 90%
Light-box signs and graphics	7.5–8.5 W/lf	2.5–3.0 W/lf	40% to 70%
Decorative-script neon/cathode signs	6.5–8.5 W/lf	2.8–3.5 W/lf	46% to 67%
Luminous walls and decorative surfaces	3.5–4.5 W/lf	2.5–3.0 W/lf	15% to 44%
Dynamic signs and audio/visual graphic boards*	7.5–8.5 W/ft^2 (static light box)	5.4–7.4 W/ft^2 (dynamic monitor)	2% to 36%

* Based on the latest generation of ENERGY STAR compliant large-screen monitors; however, because dynamic monitors can potentially replace three or four static light-boxes total, energy saved can approach 65% to 90%.

conventional static signs and light boxes, three, four, or more individual units may be required to communicate the same series of messages. Furthermore, the ability to place the monitors in standby mode or turn them off when the store is closed and have them automatically turn back on when power is restored is highly desirable and can maximize energy savings.

Table 5-6 provides guidelines for potential expected energy savings for LED lighting systems versus conventional fluorescent, cold-cathode, and neon counterparts.

PL7 Plug Load Control Strategies (Climate Zones: all)

Plug load control strategies can be as important as, or even more important than, minimizing connected wattage in reducing overall plug load energy consumption. At a minimum, it is essential that all plug loads be turned off or put into standby outside of operating hours and when otherwise not in use. Many vendor-supplied items, such as vending machines, beverage refrigerators, and self-service kiosks, can be specified to have automatic standby modes; ENERGY STAR rated computers are often equipped with low-power standby modes. Plug load equipment without automatic standby capability should be plugged into controllable power strips and/or electrical outlets such that they can be scheduled OFF outside of operating hours by the building automation system (BAS) or even by a simple timer switch; since most equipment consumes measurable (and sometimes significant) amounts of energy even when in standby mode, external power control is an advisable strategy for all equipment. The objective should be that all plug loads that are not attached to perishable merchandise should be turned off when the store is closed.

Consideration for plug load control should be made early in a project so that electrical circuiting can be used to group common power loads that have common ON-time requirements. This allows more economical control from a BAS by switching multiple plug loads with common circuits. Circuiting should also be considered for possible use with electrical demand-control strategies (reducing power consumption on a call from a utility provider). Grouping similar types of power loads on common circuits will ease the implementation and reduce the investment cost for demand control.

Using a motion-activated power strip or outlet is a good option for intermittent loads where a user needs to engage with the equipment and the start-up time is relatively short. Typically vacancy-based or time-out-based power strips and outlets are easier to deploy and more reliable at saving energy. These devices can also be plug strips where an operator presses a button and the device stays on for a set amount of time before shutting off. Vacancy sensors also require an operator to turn on the power, but a motion sensor will shut the unit off when there is no activity. Power strips with these sensors can be used to control vending lighting and compressors and to power down monitors, including security monitors, and other items typically plugged in at employee workstations such as fans, chargers, and task lighting. ENERGY STAR rated vending machines include this type of control or can be retrofitted with add-on equipment.

Grocery Store Plug Load Strategies

The following table provides a list of plug loads typically found in a grocery store and options for controlling the loads. The objective is to effectively use the loads when they are needed and to have them off when they are not needed.

Plug Load	Purchase ENERGY STAR Product	Motion-Activated Switch or Strip	Time-Out/Vacancy-Based Switch or Strip	Load Sensing to Master	Time Clock on Store Hours	Manual Switch	Specify PRODUCT with Auto OFF	No Control Strategy Available	Comments
Two-way radio charging station							x		
Air compressor, cake decorations			x				x		
Battery charger, single battery							x		
Battery charging station, portable printer/label machine							x		
Battery charging station, wireless scanning tool							x		
Battery charging station, pallet jack or forklift							x		
Beverage refrigerator	x				x				Set thermostat to 39°F to 45°F
Blood pressure monitor					x				
Break room refrigerator	x								Maximum one 18 ft^3 per 50 employees; use as small a unit as possible
Bug lamp					x				
Charger for floor washer								x	
Coffee grinder station					x				
Coffee maker, break room					x		x		Auto OFF is a standard safety feature on many makes/models
Coin exchange customer kiosk					x				
Computer monitor, checkout area				x		x			Control with “lane occupied” switch
Conveyor belt				x		x			Control with “lane occupied” switch
Credit card scanner				x		x			Control with “lane occupied” switch
Cathode ray tube (CRT) TV	x								Replace with LED monitor
Customer convenience barcode scanner					x				
Customer water purifier/dispenser					x				

Plug Load	Purchase ENERGY STAR Product	Motion-Activated Switch or Strip	Time-Out/Vacancy-Based Switch or Strip	Load Sensing to Master	Time Clock on Store Hours	Manual Switch	Specify PRODUCT with Auto OFF	No Control Strategy Available	Comments
Demagnetizer				x					
Desktop computer	x						x		Set sleep mode to 5 min; consider laptops
Display lighting, coffee grinder station			x						
Drinking fountain					x				
DVD rental station					x				
Electric wheelchair							x		
Employee badge reader								x	
Floor cleaner								x	
Label writer	x			x					
Laminator			x						
Modem								x	
Plug-in fans		x							
Point-of-sale (POS)/ cash register terminal							x		Specify with built-in low-power/ sleep mode
POS barcode scanner and scale				x					Tie to “lane occupied” light or sleep mode of POS device
Printer	x						x		Set sleep mode to 5 min
Receiving door fly fan							x		Tie switch to door position sensor
Reverse osmosis systems store water use meter								x	
Reverse osmosis systems water system								x	
Scale					x		x		
Security monitor	x	x							Use flat screens; note that many images can be displayed on a single screen
Self-checkout POS terminal					x				
Slicer							x		
Solenoid for produce sprinkler system								x	
Task light	x	x	x		x		x		
Vending machine	x	x			x				Use only LED lights

Source: Torcellini et al. (2015)

Units are available that sense a master plug or circuit. When the master is turned off or goes into a lower power state, the rest of the outlets are de-energized. An example is when a computer goes to sleep all the peripheral devices are de-energized. Where an operator has a clear need to turn on and off the power (usually to show occupancy or that a device is in service), related devices can be connected to the same circuit. An example might be the "line in use" light for point-of-sale units. Equipment that has a quick start-up time can also be attached to this same circuit.

Another approach well suited to employee office and common areas is to control selected room outlets with occupancy sensors (sensors can be shared between the electric lighting system and the controlled outlets). Education can encourage employees to plug the majority of their equipment and/or appliances into the power-controlled outlets. Timer switches should be applied for central equipment that is unused during unoccupied periods but that should be available throughout occupied periods; examples include water coolers and central coffee makers. Network power management software can facilitate central control of office equipment (computers and multifunction printing, copying, and faxing devices), allowing for off-hour standby (or shutdown) with flexibility for software updates and maintenance.

PL8 *Parasitic Loads* (Climate Zones: all)

Reduce and eliminate parasitic loads. These loads include small energy usage from equipment that is nominally turned off but still uses a trickle of energy. Transformers that provide some electronic devices with low-voltage direct current from plug alternating current also draw power even when the equipment is off. Transformers are available that are more efficient and have reduced standby losses. Wall-switch control of power strips noted in PL7 cut off all power to the plug strip, eliminating parasitic loads at that plug strip when the switch is controlled OFF. Newer power management surge protector outlet devices have low or no parasitic losses (Lobato et al. 2011).

PL9 *Unnecessary Equipment* (Climate Zones: all)

Identify and eliminate equipment that qualifies as "nice to have" but is not fundamental to the core function of the business. Chilled water coolers in northern climates and powered paper towel dispensers are examples of possible unnecessary equipment. When managing plug loads, it is useful to create an inventory of the store's plug loads and identify strategies for controlling those loads. The Grocery Store Plug Load Strategies sidebar provides a list of typical loads in a grocery store.

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KITCHEN EQUIPMENT

EQUIPMENT AND DESIGN GUIDELINES

KE1 *General Guidance* (Climate Zones: all)

Food service operations within grocery stores range from those with just a few ancillary appliances to those with a complete deli, bakery, and/or in-store restaurant. The energy and water use within a large food service operation could add up to 20% to the total utility bill for the grocery store. Despite the wide range of appliances within a food service operation and the resulting energy intensity of a particular building, the appliance selection process and best-practices design strategies apply to all food service facilities. To impact the energy consumption of the kitchen, it is best to include the deli/bakery/restaurant manager in the design process. Opportunities to conserve energy include the following:

- Select ENERGY STAR equipment as a minimum standard for designs that include any of the eight appliance categories currently available, including reach-in refrigerators and freezers. For other categories, refer to the Commercial publications from Consortium for Energy Efficiency (CEE 2015) and Appliance Performance Reports from Food Service Technology Center (FSTC) (2015a).
- Select exhaust hood styles and designs that allow a reduction in the exhaust and makeup airflow rates. Where feasible, incorporate separate exhaust fans for hoods that operate over appliances that will be used during different times of the day. Doing this improves the return on investment (ROI) of variable-speed or demand-controlled kitchen ventilation (DCKV) systems.
- Consider transferring available "outdoor" air from the sales floor to the food service area as a contribution to the makeup (replacement) air requirement of the exhaust hoods. Select walk-in freezers and coolers with high-performance thermal envelopes and refrigeration systems. The refrigeration system should comply with Section 312 of the *Energy Independence and Security Act of 2007* (EERE 2015).
- The application of LED lighting in both exhaust hoods and walk-in refrigeration units is recommended. (See also EL25.)

KE2 *Energy-Efficient Kitchen Equipment* (Climate Zones: all)

Specify energy-efficient appliances, including dishwashers, solid-door freezers, fryers, hot-food holding cabinets, ice machines, refrigerators (solid and glass door), and steamers. In addition, select low-flow hot-water fixtures to minimize both water and energy use. ENERGY STAR qualified commercial food service equipment has been tested in accordance with appropriate ASTM Standard Test Methods, and the specifications for ENERGY STAR certification ensure that the "cooking" performance (from a food safety perspective) is equal to or better than non-ENERGY STAR appliances.

The Commercial Kitchens Initiative (CKI) and ENERGY STAR websites provide good lists of efficiency strategies and ENERGY STAR rated commercial kitchen equipment (CEE 2014; EPA 2015). The goal is to provide clear and credible definitions in the marketplace as to what constitutes highly efficient energy and water performance in cooking, refrigeration, and sanitation equipment and to help streamline the selection of products through a targeted market strategy. There are also resources from FSTC (2015a), including links and guidance on efficient design for commercial kitchens. FSTC, supported by California public-goods energy efficiency funding, is the industry leader in commercial kitchen energy efficiency and appliance performance testing. The Center has developed over 35 standard test methods for evaluating commercial kitchen appliance performance.

While there are only eight categories of commercial kitchen equipment in the ENERGY STAR program, there are over 35 ASTM International standard performance test methods that provide a recognized method to test the capacity, cooking performance, and energy use and efficiency of appliances. Using a specification that requires the manufacturer to provide test results from an ASTM Standard Test Method ensures that appliances submitted for approval during construction meet the project's design energy goals. Table 5-7 lists ASTM International performance test method standards that may be relevant to food service operations within grocery stores (FSTC 2015b). Performance specifications based on these Standard Test Methods are a recommended design strategy.

Table 5-7 Commercial Food Service Appliance ASTM Standard Test Methods

ASTM #	Appliance Type
F1275-03	Griddles
F1361-05	Open deep-fat fryers
F1484-05	Steam cookers
F1496-99(2005)	Convection ovens
F1521-03	Range tops
F1605-95(2001)	Double-sided griddles
F1639-05	Combination ovens
F1695-03	Underfired broilers
F1696-96(2003)	Single-rack hot-water sanitizing, door-type commercial dishwashing machines
F1704-05	Commercial kitchen exhaust ventilation systems
F1784-97(2003)	Pasta cookers
F1785-97(2003)	Steam kettles
F1786-97(2004)	Braising pans
F1787-98(2003)	Rotisserie ovens
F1817-97	Conveyor ovens
F1920-98(2003)	Rack conveyor, hot-water sanitizing, commercial dishwashing machines
F1964-99(2005)	Pressure and kettle fryers
F1965-99(2005)	Deck ovens
F1991-99(2005)	Chinese (wok) ranges
F2022-00	Booster heaters
F2093-01	Rack ovens
F2140-01	Hot-food holding cabinets
F2141-05	Self-serve hot deli cases
F2142-01	Drawer warmers
F2143-04	Refrigerated buffet and preparation tables
F2144-05	Large open-vat fryers
F2237-03	Upright overfired broilers
F2238-03	Rapid-cook ovens
F2239-03	Conveyor broilers
F2324-03	Pre-rinse spray valves
F2379-04	Powered open warewashing sinks
F2380-04	Conveyor toasters
F2472-05	Staff-served hot deli cases
F2473-05	Water bath rethermalizers
F2474-05	Commercial kitchen ventilation/appliance systems
F2519-05	Commercial kitchen filters and extractors
F2644-07	Commercial patio heaters
F2990-06	Commercial coffee brewers

Demand-Controlled Kitchen Ventilation System

A study was conducted in a grocery store in North California to evaluate a demand-controlled kitchen ventilation (DCKV) package installed on the commercial kitchen ventilation system. The store's kitchen included the three cooking lines with wall-mounted canopy exhaust hoods with a total exhaust ventilation rate of 23,800 cfm. The three lines were a rear deli line with a 9500 cfm exhaust hood, a front deli line with a 5800 cfm exhaust hood, and a Chinese cuisine line with two 4250 cfm side-by-side exhaust hoods.

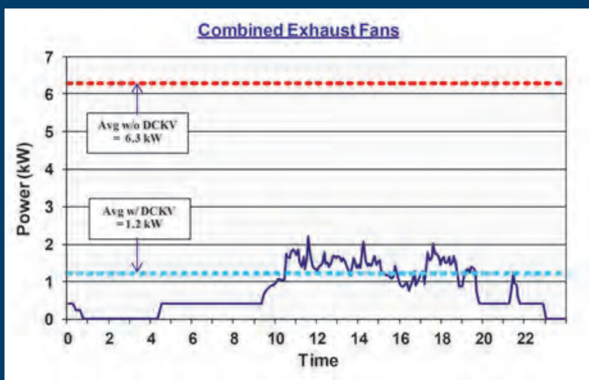
A DCKV system reduces fan energy consumption by slowing exhaust and makeup air fans when full speed is not needed. The speed of the motors is modulated with variable-frequency drives (VFDs) based on input received from both a temperature sensor mounted in the exhaust duct and an infrared beam that spans the length of the exhaust hood. When heat, smoke, or steam trigger either the sensor or the beam, fan speed is increased as needed.

Demand-Controlled Kitchen Ventilation System Overview
Image reproduced with permission of Melink Corporation, Intelli-Hood®

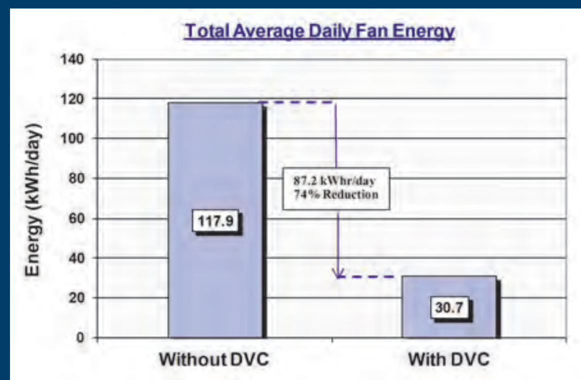
The kitchen was originally equipped with manual fan control and individual wall switches for each hood. It was retrofitted with a DCKV system configured for fully automatic operation that would turn on the fans upon the first detection of heat or smoke and turn them off at the end of the day after the equipment cooled.

As illustrated in the following graphs, the DCKV system reduced average daily fan energy consumption by 74% (118 to 31 kWh/day). This represented an annual reduction of 31,370 kWh and $3770 at the time of the study. The reduction in makeup air heating and cooling energy load provided additional savings for a combined total yearly operation cost savings of $9310, or a payback of less than two years.

Additional details on this study and the results (Fisher-Nickel 2006) are available on the Fisher-Nickel website at www.fishnick.com/publications/appliancereports/hoods/Supermarket_Melink_Report.pdf.



Typical-Day Power Profile



Daily Fan Power Comparison

Graphs reproduced with permission of Fisher-Nickel, Inc.

KE3 Exhaust and Ventilation Energy Use **(Climate Zones: all)**

Emphasis should be on designing exhaust ventilation systems with proper layout of cooking equipment and proper hood design to minimize total airflow while still providing adequate exhaust flow for complete capture and containment of the cooking effluent. After minimizing ventilation needs, consider variable-speed exhaust hood flow systems, otherwise known as demand-controlled kitchen ventilation (DCKV) systems. The design and specifications of a kitchen hood system, including the exhaust hood, ductwork, exhaust fan, and makeup air, need to be addressed by the food service consultant and/or the mechanical engineer, which requires collaboration and communication between these two disciplines. Additional opportunities include exhaust air energy recovery using dedicated exhaust heat recovery ventilation (HRV) units. However, energy recovery from grease-laden exhaust air may be considered too expensive when wash-down and/or fire protection systems are included. There are also limitations within the *International Mechanical Code* (IMC) with respect to the installation of an HRV in kitchen exhaust systems (ICC 2015).

Augmenting relevant design information provided by the *ASHRAE Handbook—HVAC Applications* chapter on kitchen ventilation (ASHRAE 2011), commercial kitchen ventilation design guides developed by the FSTC provide additional guidance for energy efficiency:

- *Design Guide 1: Improving Commercial Kitchen Ventilation System Performance—Selecting and Sizing Exhaust Hoods* (SCE 2004) covers the fundamentals of kitchen exhaust and provides design guidance and examples.
- *Design Guide 2: Improving Commercial Kitchen Ventilation System Performance—Optimizing Makeup Air* (CEC 2002) augments Design Guide 1, with an emphasis on the makeup-air side of the equation.
- *Design Guide 3: Improving Commercial Kitchen Ventilation System Performance—Integrating Kitchen Exhaust Systems with Building HVAC* (SCE 2009) provides information

that may help achieve optimum performance and energy efficiency in commercial kitchen ventilation systems by integrating kitchen exhaust with building HVAC.

- *Design Guide 4: Improving Commercial Kitchen Ventilation System Performance—Optimizing Appliance Positioning and Hood Configuration* (PG&E 2011) discusses the influence of appliance positions under a hood on the exhaust requirements.

KE4 *Minimize Hot-Water Use* (Climate Zones: all)

The FSTC publishes a hot-water system design guide for commercial kitchens that provides key information to restaurant designers and engineers on how to achieve superior performance and energy efficiency with their systems. The information is relevant to the design of hot-water systems in grocery stores with lengthy distribution piping. This design guide, *Improving Commercial Kitchen Hot Water System Performance—Energy Efficient Heating, Delivery and Use* (Fisher-Nickel 2010), describes the design process and reviews the fundamentals of commercial water heating, including the following topics:

- Reducing hot-water use of equipment while maintaining performance
- Increasing the efficiency of water heaters and distribution systems
- Improving hot-water delivery performance
- Incorporating "free heating" technologies, such as waste heat recovery and solar preheating

KE5 *Position Hooded Appliances to Achieve Lower Exhaust Rates* (Climate Zones: all)

Research sponsored in part by ASHRAE shows that the position of appliances under a hood can make a significant difference in the required exhaust rate—up to 30% (ASHRAE 2005). Some key recommendations are as follows:

- Position heavy-duty equipment (such as underfired broilers or wok ranges) in the middle of the cook line.
- If a heavy-duty appliance is on the end, incorporating a side panel or end wall is imperative.
- Fryers and broilers should not be placed at the end of a cook line. Ranges can be located at the end of a cook line because under typical operating conditions the plume strength is not as high as that of broilers.
- Locate double-stacked ovens or steamers at the end of the hood. This has a plume control effect that tends to assist in capture and containment. However, this does not replicate the value of adding a partial side panel or end wall. Both strategies should be considered.

Repositioning of appliances requires approval of the kitchen manager and/or kitchen consultant. If these recommendations are followed, let the kitchen hood manufacturer and design mechanical engineer know why these decisions were made, and reference the ASHRAE research (ASHRAE 2005) or FSTC Design Guide 4 (PG&E 2011). The resulting design exhaust and makeup air rates should be less than those of a conventional design.

KE6 *Operating Considerations* (Climate Zones: all)

While not directly part of the design process, providing staff training to encourage best practices within the food service operation is critical to sustaining an energy-efficiency operation through measures including but not limited to the following:

- Institute a start-up and shut-down schedule for all cooking and holding equipment, including the exhaust hood.
- Properly adjust standing pilot lights on appliances such as range tops and broilers. Properly adjust air shutters on gas appliances. Flames should be hard and blue, not soft and yellow.

- Repair or replace damaged door gaskets on cooking and holding appliances and replace missing control knobs.
- Install occupancy sensors or timers in areas of infrequent use, for example, closets, storage rooms, break rooms, and restrooms.
- Maintain refrigerator doors by replacing worn gaskets, aligning doors, enabling automatic door closers, and replacing worn or damaged strip curtains.
- Clean evaporator and condenser coils and ensure proper airflow. Straighten damaged fins and remove objects that block air to the coils.
- Check and properly set thermostats on all refrigeration equipment.
- Install strip curtains or plastic doors on walk-in cooler and walk-in freezer exterior doors.
- Set the thermostat of storage water heaters at the minimum temperature consistent with avoiding microbiological growth in the tank (typically 135°F) or the minimum required to supply temperature for the end use (e.g., 140°F for dishwashing machines).
- Control the hot-water recirculation pump with a time clock so that the pump is turned off when the facility is closed.
- Improve capture and containment performance of exhaust hoods so that exhaust airflow rates may be reduced, saving fan energy and makeup air conditioning loads. Ensure cooking appliances are pushed as close as possible to the back wall, maximizing hood overhang. Use end panels where practical (most cases).
- Install a low-flow pre-rinse spray valve (≤1.2 gpm) at the dishwashing machine.
- Have a professional check and maintain the dishwashing machine. Properly set rinse temperature and pressure and overflow-bypass and replace worn rinse-arm nozzles.
- Consider installing a time clock to control the ice machine and restrict operation to off-peak utility hours. Peak hours are typically noon to 6:00 p.m.

Contact your local energy and water utility to conduct an audit at your facility. The survey should include a thorough examination of lighting, refrigeration, sanitation equipment, cooking equipment, and HVAC equipment. In many cases, local utilities provide their customers with free audits. In the event an audit is not provided, the customer may need to find a third party program or an energy consulting firm.

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REFRIGERATION EQUIPMENT

CONDENSER SELECTION AND CONDENSER CONTROL

Good Design Practice

RF1 ***Condenser Choices*** **(Climate Zones: All)**

Identifying the best condensing means is important but is far from obvious in many areas. The two recommended choices are air-cooled and evaporative-cooled condensers. Hybrid condensers and precooling are discussed in a sidebar in Chapter 3. Rather than specify which type of condenser to select, this document points to how each type of condenser can be most effectively selected and controlled.

Many factors drive condenser choice, including physical constraints, noise restrictions, and water cost and availability. The choice of condenser should be approached with an open mind and with the use of appropriate analytical tools, engineering expertise, and operating experience. Hourly full-system simulation with weather and utility rates is very useful in evaluating performance and cost-effectiveness for all condenser options.

Air-cooled condensing systems may use less energy throughout the year. Condenser size (see RF2) is defined in terms of the condenser temperature difference (TD). Hotter climate zones have lower recommended TD values and colder climate zones have higher recommended TD values. Note that the TD recommendations are made in conjunction with the specific efficiency recommendations (see RF3). The maximum fin spacing of 10 fins per inch (FPI) is recommended to minimize fouling with airborne debris and to facilitate cleaning. Fewer fins per inch may be preferred in some areas, but there is an associated reduction in capacity that should also be considered.

Evaporative-cooled condensing systems may reduce overall utility costs due to demand charges and higher energy costs during peak summer periods. The benefit of evaporative-cooled condensing is greatest in peak weather when electric costs are typically higher (particularly for summer peaking utilities).

Both types of condensers should use floating head pressure to a minimum condensing temperature of 60°F or lower, use a variable-setpoint control strategy, and control all fans in unison with variable-speed control. Floating head pressure with these setpoints and strategies has comparatively greater benefit for air-cooled condensing due to the different responses of air-cooled and evaporative-cooled condensers at lower ambient conditions (see RF4).

RF2 ***Condenser Sizing Criteria*** **(Climate Zones: All)**

Condenser size is defined in terms of the condenser TD. The TD recommendations are made in conjunction with the specific efficiency recommendations.

Condensers should be designed using system design heat rejection determined using compressor capacity, not system load. Fully redundant compressor capacity (i.e., the backup compressor) can be excluded. The design approach or TD values (the difference between saturated condensing temperature and entering ambient temperature) shown in Tables 5-8 through 5-10 provide efficient condenser sizing in conjunction with the other condenser control and floating head pressure recommendations.

Air-cooled condenser design TDs vary by climate zone, based on the estimated cost-effectiveness in each climate zone. It may seem surprising that some TDs are higher than the established practice of 10°F TD for low-temperature and 15°F TD medium-temperature systems. As a historical point, these TDs have been used for many decades and have largely been a function of maintaining compressor life in hot weather, when compressors are less reliable and less efficient. On an incremental basis, with efficient compressor systems and condenser control using variable-speed control of all fans in unison and with an efficient condenser spe-

Table 5-8 Typical Selection Table for Air-Cooled Condensers

Case Type	Design TD
Low temperature	8°F to 12°F
Medium temperature	12°F to 18°F

Table 5-9 Typical Selection Table for Evaporative-Cooled Condensers

Design Wet-Bulb Temperature	Design TD
<68°F	23°F
68°F to 75°F	20°F
>75°F	16°F

Table 5-10 Typical Selection Table and Capacity Factors for Evaporative-Cooled Condensers

Refrigerants 12, 22, 500, and 502														
Condensing Pressure, psig		Condensing Temperature, °F	Wet-Bulb Temperature, °F											
R-12	R-22		50	55	60	65	68	70	72	75	78	80	85	90
76.9	133.5	75	1.46	1.66	1.96	2.51	3.11	3.45	4.26					
84.1	145.0	80	1.26	1.41	1.64	2.03	2.44	2.69	3.19	3.93	4.02			
91.8	155.7	85	1.10	1.22	1.39	1.67	1.94	2.13	2.45	2.94	3.02	3.63		
99.8	168.4	90	0.93	1.02	1.14	1.32	1.47	1.59	1.75	2.00	2.38	2.78	3.34	
108.3	181.8	95	0.80	0.87	0.95	1.08	1.16	1.22	1.32	1.45	1.61	1.79	2.56	3.09
117.2	195.9	100	0.71	0.76	0.82	0.89	0.93	1.00	1.03	1.12	1.23	1.33	1.72	2.50
126.6	210.8	105	0.63	0.66	0.70	0.76	0.79	0.83	0.86	0.93	1.00	1.05	1.27	1.61
135.4	226.4	110	0.56	0.59	0.62	0.66	0.70	0.71	0.75	0.79	0.84	0.88	1.01	1.19
146.8	242.7	115	0.49	0.52	0.55	0.58	0.60	0.62	0.64	0.67	0.70	0.73	0.81	0.92
157.7	259.9	120	0.41	0.45	0.48	0.51	0.53	0.54	0.55	0.57	0.60	0.62	0.68	0.75

Factors courtesy of SPC Cooling Technologies, Inc.

cific efficiency, the cost-effective sizing is often a higher TD than the traditional 10°F and 15°F TDs. There is nothing wrong or necessarily inefficient with using the condenser sizes determined from these traditional TDs, particularly in warmer climates. However, detailed hourly analysis that considers actual local climate, utility rates, impact of heat recovery, and store operations is the most accurate way to optimize condenser sizing, efficiency, and cost-effectiveness for a particular store.

The range of TDs in Table 5-9 reflects the fact that for a given heat of rejection capacity, the TD of an evaporative-cooled condenser changes with wet-bulb temperature. This can be found in the capacity factors in Table 5-10 and is simply the consequence of moist-air properties, where in general colder air holds less heat energy. The varying application TD doesn't necessarily provide a result in a smaller condenser, since the sizing factor changes at lower wet-bulb temperatures. As noted above for air-cooled condensers, this is a recommendation that can be improved upon with hourly simulation of the conditions at a particular store.

The sizing assumptions for both air-cooled and evaporative-cooled condensers do not include consideration of space heat recovery. The condenser must still be sized to maintain reasonable condensing temperatures during peak weather conditions, but heat recovery may change the economics of condenser design substantially. For example, in Seattle, Washington, with mild temperatures year round, a conventional grocery store with heat recovery could be operating in heating mode for the majority of the year. The most cost-effective condenser size can be far different in a store without heat recovery, with potentially lower size, cost, and refrigerant charge. Hourly modeling can help to optimize the incremental or marginal selection of a condenser.

Table 5-11 Specific Efficiency Assumptions

Specific Efficiency Rating Basis	Evaporative-Cooled Condenser	Air-Cooled Condenser
SCT °F	100	
Wet-Bulb Temperature °F	70	
Dry-Bulb Temperature °F		
TD °F		10
Specific Efficiency, Btu/h·W	180	85

RF3 *Condenser Specific Efficiency* (Climate Zones: All)

Specific efficiency is a relatively new term and is not found directly in condenser catalogs. It is calculated by dividing the condenser capacity at an assumed condition by the condenser input power. An important point is that the specific efficiency rating point is unrelated to the application conditions for a particular store; rather it is simply a point selected that allows air-cooled condensers to be compared with each other. The specific efficiency rating basis used in Table 5-11 is a 10°F TD between saturated condensing temperature (SCT) and entering dry-bulb temperature.

Both air-cooled and evaporative-cooled condensers are available with a very wide range of fan power for a given capacity. Air-cooled condensers tend to use direct-drive motors and have smaller motors, but still with a substantial range in power for a given capacity. *Specific efficiency* is the term used to define condenser fan power versus capacity and describes the heat rejection capacity at an assumed specific efficiency rating point divided by the input power for the condenser fans and, for evaporative-cooled condensers, the spray pump. Specific efficiency rating conditions are unrelated to the application conditions. The rating conditions for air-cooled and evaporative-cooled condensers are necessarily different, since one is based on wet-bulb temperature and one is based on dry-bulb temperature. Table 5-11 shows the rating assumptions and recommended specific efficiencies.

Air-cooled condenser manufacturers publish capacity adjustments for altitude but not information on motor power at altitude. The typical air-cooled capacity adjustment for 5000 ft altitude is approximately 12%, which is roughly similar to the air density change from sea level. Because fan power and density are nominally proportional (i.e., based on affinity laws), it could be assumed that the same specific efficiency basis is reasonable at higher altitudes. Air-cooled condensers require far greater air volume than evaporative-cooled condensers and thus generally have higher fan power.

Caution: Comparison of specific efficiency can be made directly among different air-cooled condensers and directly among different evaporative-cooled condensers; however, air-cooled condenser specific efficiency and evaporative-cooled condenser specific efficiency cannot be directly compared because

- air-cooled condensers respond to dry-bulb temperature and evaporative-cooled condensers respond to wet-bulb temperature and
- any set of specific efficiency rating conditions could have been chosen for the rating basis, which would have resulted in completely different numeric values.

Furthermore, simulation modeling is recommended for comparing air cooling to evaporative cooling to account for all aspects and assumptions regarding the two condensing options, including the hourly weather.

RF4 *Floating Head Pressure* (Climate Zones: All)

Floating head pressure often provides more refrigeration system energy savings than other energy-saving measures. As a rule of thumb, one degree of reduction in condensing temperature saves 2% compressor energy. While systems must be designed to operate properly and maintain temperatures at the highest ambient conditions that may occur, maximum energy savings is achieved when the head pressure is controlled to take advantage of the lower hourly temperatures throughout the year. In practice, there are numerous factors to consider to operate reliably with the lowest energy usage, including both compressor and condenser energy.

Aside from the relatively few hours (if any) in a year that the compressors and condensers run near their maximum capacities, there is a constant opportunity to employ controls to optimize the total power used by the compressors and condenser fans. For lack of a better description, this is called *floating head pressure*. Floating head pressure is the overall effort to control to the lowest total energy use of compressors and condensers throughout the year. There are three elements to floating head pressure that are discussed in the how-to tips that follow:

- Minimum condensing temperature (RF5)—How low can the condensing temperature (discharge pressure) go at minimum weather conditions?
- Condenser fan control (RF6)—How are the condenser fans themselves controlled?
- Setpoint determination (RF7)—How is the control setpoint determined?

RF5 *Minimum Condensing Temperature* (Climate Zones: All)

The refrigeration system should be designed to operate down to 60°F SCT or lower where weather permits. Grocery store refrigeration systems routinely operate down to 70°F SCT, and there is experience operating at lower than 60°F SCT in colder areas of the country. System design considerations include compressor operating envelopes, oil separator sizing, defrost design, and, most importantly, the ability of the control system to provide stable condenser pressure at the lowest pressure conditions.

The value of a minimum condensing temperature lower than 60°F may be small in warm climates where there are few hours in the year with ambient temperatures below 50°F. Hourly analysis readily shows the incremental value of successively lower minimum SCT setpoints.

When evaporative-cooled condensing is used, the liquid receiver should be insulated to reduce the effect of ambient temperatures. The dry-bulb temperatures surrounding the receiver (which are always higher than the wet-bulb temperature and often much more so inside a compressor room) can often inhibit floating head pressure and cause liquid to be backed up into the evaporative-cooled condenser. Insulating the receiver reduces heat gain and with a flow-through horizontal receiver is typically sufficient. With a vertical or surge (bypass) receiver, insulation is often not sufficient since the liquid-vapor interface is still and will come to equilibrium with the surrounding temperature even if the receiver is insulated. For vertical and surge receivers, a small solenoid should be added to vent vapor to the suction line, with control based on sensing subcooling in the condensate (drop leg) temperature from the condenser. If subcooling exceeds 2–4 degrees (some difference is needed to allow for sensor error), the solenoid should be opened briefly until subcooling is reduced. Essentially this control acts to keep the liquid in the receiver at the condition the condenser is able to maintain, thereby gaining full benefit from floating head pressure. This strategy could also be used on air-cooled systems if the compressor room is maintained at a temperature higher than the condensing temperature. Control systems should trend condensing temperature and condensate temperature along with calculated subcooling as a tool for system commissioning and ongoing maintenance.

RF6 *Condenser Fan Control* (Climate Zones: All)

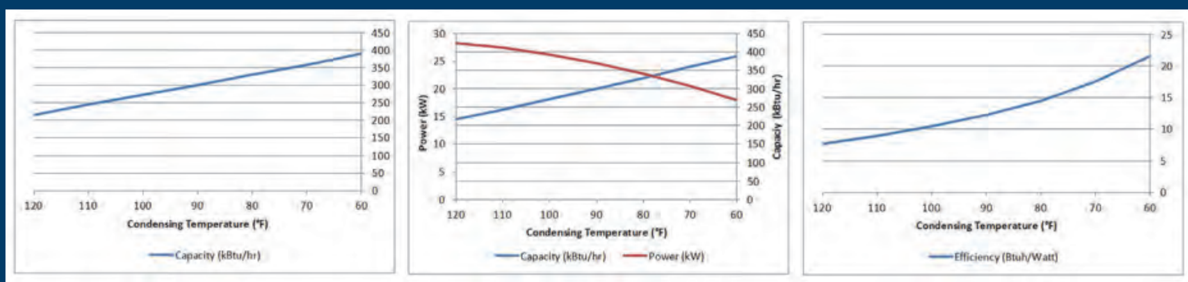
For both evaporative and air cooled condensers all fans should be controlled in unison with variable speed, rather than fan cycling. The use of all condensing surface, all of the time, is the most efficient means of condenser capacity utilization. Condenser heat rejection capacity (at given entering air and condensing temperature conditions) is nominally directly proportional to fan speed and airflow whereas power varies with the cube of fan speed.

Maintaining stable condensing temperature is most important (and most challenging) when operating at or near the minimum condensing temperature. Stability, rather than the low condensing temperature itself, is more likely to inhibit successful floating head pressure. This is particu-

Condensing Temperature

The graphs that follow show the effect of condensing temperature on capacity, power, and compressor efficiency. The tables show more specifically the numbers for two specific compressors, one low temperature and one medium temperature.

The fact that capacity varies more than mass flow demonstrates that much of the capacity gained from floating head pressure is mostly a function of refrigerant properties rather than the compressor pumping more refrigerant.



Effects of Condensing Temperature on (a) Capacity, (b) Power, and (c) Compressor Efficiency
Table data source: Copeland

Copeland ZB45KCE Scroll Compressor

	Refrigerant Massflow (lb/hr)			
	10°F SST	20°F SST	30°F SST	40°F SST
70°F SCT	840	1,040	1,280	
80°F SCT	835	1,040	1,270	1,560
90°F SCT	830	1,030	1,260	1,550
100°F SCT	815	1,010	1,250	1,530
110°F SCT	805	1,000	1,240	1,520
120°F SCT	790	985	1,220	1,510

	Refrigeration Capacity (MBH)			
	10°F SST	20°F SST	30°F SST	40°F SST
70°F SCT	55.5	68.0	82.5	
80°F SCT	52.0	63.5	77.5	93.0
90°F SCT	48.3	59.5	72.0	86.5
100°F SCT	44.6	55.0	65.5	80.0
110°F SCT	40.8	50.0	61.0	73.0
120°F SCT	36.7	45.2	55.0	66.5

	Compressor Power (kW)			
	10°F SST	20°F SST	30°F SST	40°F SST
70°F SCT	3.43	3.44	3.44	
80°F SCT	3.88	3.90	3.90	3.87
90°F SCT	4.40	4.41	4.41	4.39
100°F SCT	4.98	4.99	4.99	4.98
110°F SCT	5.65	5.65	5.65	5.65
120°F SCT	6.40	6.40	6.40	6.35

Copeland ZF(D)18KVE Scroll Compressor

	Refrigerant Massflow (lb/hr)			
	-40°F SST	-30°F SST	-20°F SST	-10°F SST
70°F SCT	248	325	415	525
80°F SCT	248	325	415	525
90°F SCT	246	323	413	520
100°F SCT	242	319	409	515
110°F SCT	237	314	403	510
120°F SCT		307	396	500

	Refrigeration Capacity (MBH)			
	-40°F SST	-30°F SST	-20°F SST	-10°F SST
70°F SCT	22.5	28.6	35.4	43.3
80°F SCT	22.2	28.2	34.8	42.4
90°F SCT	21.8	27.6	34.1	41.4
100°F SCT	21.3	27.0	33.2	40.3
110°F SCT	20.6	26.2	32.2	39.1
120°F SCT		25.2	31.1	37.6

	Compressor Power (kW)			
	-40°F SST	-30°F SST	-20°F SST	-10°F SST
70°F SCT	3.20	3.40	3.61	3.84
80°F SCT	3.56	3.81	4.07	4.33
90°F SCT	3.91	4.21	4.51	4.80
100°F SCT	4.28	4.64	4.98	5.30
110°F SCT	4.71	5.10	5.50	5.85
120°F SCT		5.70	6.15	6.55

Compressor Capacity, Mass Flow, and Power versus Condensing Temperature
Table data source: Copeland

larly evident with fan cycling control—when the last fan cycles on and off, the system experiences large oscillations in liquid-line pressure, upsetting expansion valve performance, temperatures, and overall system stability. Variable-speed control still requires consideration of stability at minimum conditions. There are numerous design options, depending on the type of condenser, the manufacturer's preferences, and control system capability, including the following:

- Control fan operation as low as zero speed (occasionally feasible).
- Control fan operation to a minimum speed (e.g., 10%–15%) followed by fan cycling after reaching minimum speed with a single variable-frequency drive (VFD) for all fans

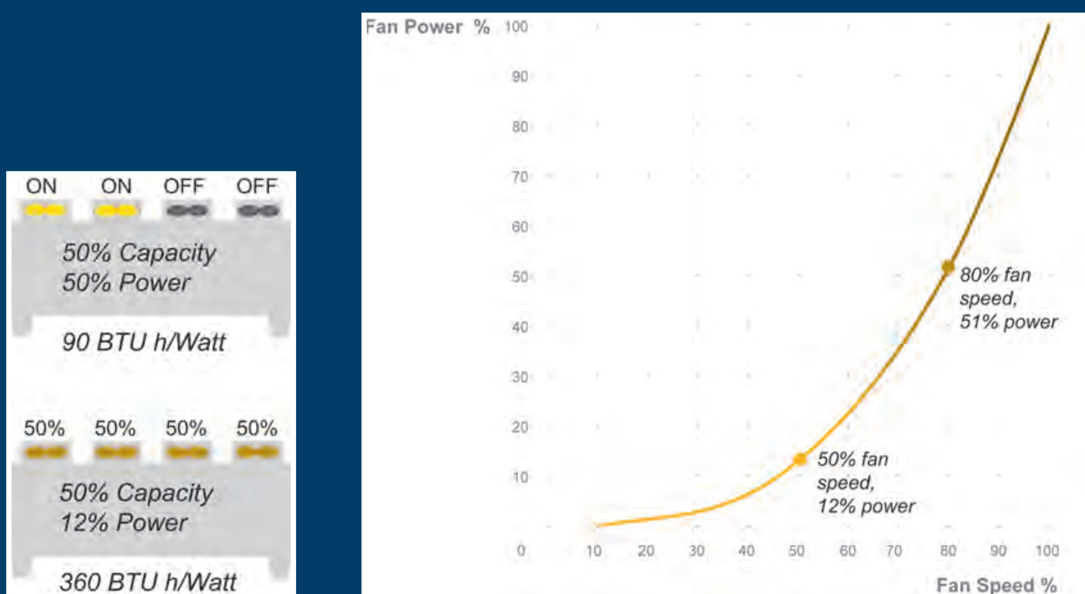
designed to allow fans to cycle or cycling multiple VFDs dedicated to motors or banks of motors.

- If fans are cycled off, cycle the fans back on as soon as they can be employed (at reduced speed) and match required speed to avoid pressure discontinuities and hunting.
- Operate all fans at minimum speed and "ride" on the condenser holdback valve setting. Many locations require a holdback valve to maintain pressure even with fans completely off (e.g., during cold, windy weather), so this often may be the most practical choice. At minimum speed, fan power is negligible.

Airflow versus Fan Power

The affinity laws for pumps/fans are used with pumping and HVAC systems to express the relationship between variables involved in pump or fan performance. The affinity laws define physical principles of flow, pressure drop, and power, and specifically the "third-power" relationship between airflow and fan power.

The following illustration is of the third-power relationship between capacity and power and shows the difference for an example condenser that has a specific efficiency of 90 Btu/h·W at 50% capacity with half the fans off (the same as the specific efficiency at full capacity) and at 50% speed with all fans running, resulting in a specific efficiency of 360 Btu/h·W, a 400% increase.



Condenser Capacity versus Power for Fan Cycling and Variable Speed

The performance values in the following table are actual results for one condenser series based on laboratory testing by the condenser manufacturer. Note that the capacity is somewhat better than proportional with fan speed, which can be attributed to better heat exchange with the air passing through the condenser.

Condenser Part-Load Performance from Laboratory Testing

Speed	100%	90%	80%	70%	60%	50%	40%	30%
Capacity	100.0%	93.0%	86.0%	78.4%	69.9%	60.8%	50.9%	39.9%
Power	100.0%	68.8%	48.7%	34.0%	22.7%	14.0%	7.8%	3.8%

RF7 *Setpoint Determination* (Climate Zones: All)

The final aspect of condenser control and optimizing system energy is setpoint determination. The essential objective is balancing the compressor and condenser power to obtain the lowest total power. From only the perspective of reducing compressor power, the condenser would simply run at 100% capacity to balance at the lowest head pressure possible at the ambient temperature. Of course, the condenser uses energy as well, which creates the trade-off between compressor power and condenser power. As shown in the Airflow versus Fan Power sidebar, fan power vs. condenser capacity is nonlinear, following a third-power relationship. In addition, like all heat exchangers, increased condensing capacity has a diminishing return in terms of the heat exchanger approach. For example, if doubling condenser capacity (and power) reduces the approach (the TD) by 20°F to 10°F (which is a reduction of 10°F in condensing temperature), an additional doubling of condenser capacity would only reduce the approach and condensing temperature by 5°F, producing only half the benefit at the compressor. Both of these nonlinear relationships complicate the goal of balancing condenser fan control versus compressor power. Simply put, the goal is to use as much condenser capacity as possible without increasing condenser power more than the gain achieved in compressor power.

The most common control strategy used to control floating head pressure for optimum power use is ambient-following logic, where the condenser control setpoint is determined by adding an "offset" value to the current ambient temperature to determine the target SCT setpoint. This offset is typically called the *control TD*. For evaporatively cooled condensers, wet-bulb temperature is used, and for air-cooled condensers, dry-bulb temperature is used.

A simplified example of ambient following control is shown in Figure 5-28. The condensing temperature setpoint follows ambient temperature, bounded by a minimum setpoint limit defined by the system design minimum pressure capability (e.g., 60°F in Figure 5-28) and typically a maximum setpoint limit as well (e.g., 95°F) at which it is desirable for the fans to run at 100% to limit maximum system pressures, regardless of energy optimization.

When using an energy simulation, the optimum control TD value is determined by iterating the simulation control TD value to obtain the lowest total combined power. To allow for real-world control variations, the control TD is then raised slightly. In actual plant operations, typically lacking detailed guidance from energy analysis, the control TD setpoint is commonly optimized using a condenser fan speed "sweet spot" of a 60%–80% target when not at minimum SCT. An average speed of 60%–80% is normally close to the ideal operating point, using

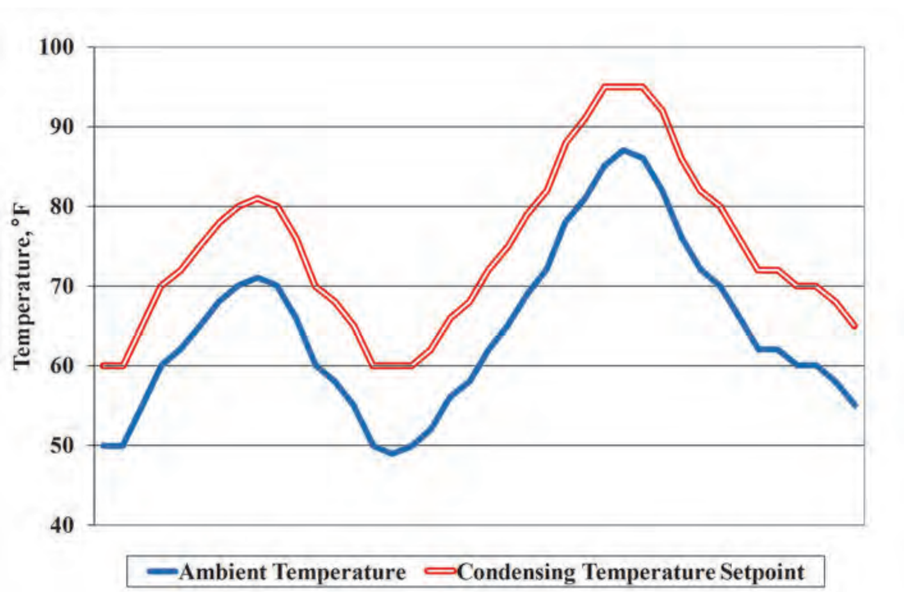


Figure 5-28 (RF7) Ambient Following Condensing Temperature Setpoint

a large fraction of the condenser capacity and still providing a sizable reduction in condenser fan power.

Caution: A long-standing problem with evaporative-cooled condensers in grocery stores is the tendency to backflood liquid in the condenser, increasing refrigerant charge and raising head pressure—essentially losing most of the potential gains from floating head pressure. This is often the result of locating the receiver tank in a warm compressor room and the use of bypass receiver piping.

COMPRESSORS

RF8 *Suction Grouping* (Climate Zones: all)

Grocery stores have a wide range of product temperature requirements that correspond to the different types of foods and products being stored and sold. This results in a wide range of required evaporating temperatures. This dynamic allows designers to maximize compressor efficiencies by grouping evaporators that have similar evaporating pressure requirements and assigning a dedicated compressor (or compressors) to those evaporators.

An example of an inefficient system is one where all the compressors operate in parallel at a single suction pressure low enough to satisfy all medium- and low-temperature evaporators. A more efficient system will have separate groups of compressors for the low-temperature and for the medium-temperature evaporators. In theory, the more the compressor lift can be reduced by means of creating additional suction groups, the more efficient the system can become. In practice, however, because an excessive number of suction groups is not practical or cost-effective, the recommendation is to create additional suction groups when the saturated suction temperature (SST) is more than 5°F below the warmest saturated evaporator temperature (SET) in that group. Applying this rule of thumb in a practical way typically results in approximately two suction groups for low-temperature systems and three suction groups for medium-temperature systems, although it may make sense to have more than this. Tighter suction grouping is even more feasible when linear capacity staging is achieved on compressors.

Designers should also note that based on the specific dynamics of a system, it may not make sense to create a separate suction group purely based on the temperature dynamics of the evaporators. There also needs to be sufficient load at the different temperature groups to justify splitting the suction. For example, it would not be practical to split the suction of a rack to operate a single case with a small load—unless it was several degrees colder than any other load on that suction group—in which case a satellite compressor (single compressor for a single load) may be warranted. Sound judgment on the system designer's part is necessary to understand the benefits of splitting suction and when it may make sense.

RF9 *Indirect Cooling Design* (Climate Zones: all)

For stores using indirect cooling, carbon dioxide (CO_2) phase-change (also called *volatile*) heat exchange is recommended for both low-temperature and medium-temperature cases and walk-ins. CO_2 is more efficient than glycol for medium-temperature indirect cooling because of better heat exchange, lower piping heat gains, and far lower pumping power requirements.

Low-temperature systems may use two-stage or cascade cooling (e.g., low-temperature CO_2 compressors condensing into or discharging into the medium-temperature level.) While cascade cooling may be somewhat less efficient than CO_2 indirect cooling, particularly in smaller stores, this may be a very practical and cost-effective way to utilize CO_2 refrigeration.

RF10 *Compressor Group Staging and Capacity Control* (Climate Zones: all)

Continuously variable capacity is recommended for each suction group. To achieve this, there are several methods offered by compressor manufacturers, including variable-speed and high-speed unloading methods, on both scroll and reciprocating compressors. Depending on the number of compressors operating per suction group, some methods will be applicable and others will not. At least one compressor will need the ability to vary capacity linearly either by varying the speed of the compressor or by applying another type of variable-capacity technology. Ideally, the compressor chosen for variable capacity would be sized such that all the other compressors on the same suction group could operate in an ON/OFF mode only. In this case, the variable-capacity compressor would provide linear capacity only between the steps of capacity provided by the nonvariable-capacity compressors.

If the variable-capacity compressor is too small or too large to fully achieve linear capacity at all possible load conditions for the suction group, some of the nonvariable-capacity compressors should be provided with additional mechanical unloading to allow them to operate at steps of capacity less than 100%. If this additional unloading still does not allow for a continuously variable capacity, then either a second compressor with the ability to achieve linear capacity needs to be provided or the number and/or size of the compressors on the suction group needs to be adjusted.

Load shifting between split suction groups is another means of capacity control that has historically been used in certain systems to help the staging of the compressor racks. However, for the sake of improving compressor staging, load shifting should not be necessary when continuously variable capacity is achieved per suction group. However, there may be other reasons designers use load shifting between suction groups.

Although the standard rack design for grocery store refrigeration systems today is parallel racks (i.e., typically three or more uneven compressors operating in parallel to serve several parallel loads), there are other options to consider that may prove to be equally effective and potentially more efficient. Since continuously variable capacity has well-known benefits, the concept could be applied to one large compressor instead of several smaller compressors. Reducing the number of compressors to one (or two for redundancy) is not a new concept and, in fact, is the concept applied in most hydronic pumping system designs. Reciprocating compressors are known for maintaining efficiency while operating at lower speeds, which also increases compressor life. Parasitic losses are also less profound in larger compressors with respect to the work that they can perform.

Other philosophies can be embraced outside of the typical parallel rack design that are being driven by a desire by some users to achieve a more "standard" solution that is more easily repeatable. Some may choose to use a standard "off-the-shelf" chiller to provide medium-temperature cooling for indirect systems. Pumping glycol should be avoided due to the high pumping energy discussed in RF9 since CO_2 is an available, and more efficient, option. Care should also be taken when selecting packaged systems to ensure that they are well suited for the application and the load. (See the Sizing Concepts section of Chapter 3 for more information.)

RF11 *Floating Suction* (Climate Zones: all)

Beyond efficient design of suction groups, additional energy can be saved by "floating" (raising) suction pressure during system operation. As does floating head pressure down, floating suction pressure up helps reduce overall compressor lift during operation. Floating suction can normally be applied to suction groups designed at 25°F or lower SSTs.

During the hours of the day or seasons of the year when the system load is reduced, there should be an opportunity to increase the SST of one or more suction groups while maintaining required product temperatures. To realize this savings, a means to monitor the project temperature (normally done indirectly by monitoring the discharge or return airstreams at the evaporators) is necessary to ensure that product temperatures are not compromised as evaporating temperatures float. Logic is also required in the compressor controller to evaluate product temperatures and adjust SST setpoints for compressor cycling.

RF12 *Mechanical Subcooling* (Climate Zones: all)

Subcooling provides substantial energy savings in several ways. Fundamentally, subcooling one system with another system at a higher suction temperature saves energy based on the difference in system efficiency. Heat removed from a low-temperature liquid line increases the refrigerating capacity on a Btu-for-Btu basis. Specifically, the Btu capacity gain on the low-temperature system is equal to the Btu load gain on the higher-suction-pressure system. However, since the higher suction pressure system operates at a higher coefficient of performance (COP), there is an overall efficiency gain.

Subcooling low-temperature systems with a medium-temperature system (e.g., +20°F SST) has been common practice. However, to maximize energy savings, subcooling is recommended for all systems with the subcooling provided from a compressor group at +30°F to +35°F SST. This is particularly effective for a centralized rack system but should be considered on distributed systems as well by using satellite compressors and continuous capacity modulation. This high-temperature suction group should also be used for food preparation areas, in lieu of +20°F to 25°F suction groups, for further energy savings.

Subcooling provides a second benefit that is not as obvious. By maintaining subcooling entering the expansion valve (eliminating flash gas), head pressure can float to lower pressures without poor expansion valve operation and resultant erratic case temperatures. Often overlooked, an essential (and challenging) requirement is that stable subcooling must be maintained continuously at all times of the year. This is challenging because the subcooler load at minimum load and minimum condensing temperature is a small fraction of the subcooling load at peak summer conditions. Electronic valves should be used. Even with electronic valves, multiple valves with staging controls may be necessary to maintain stable control. Electronic expansion valves specially designed for subcooling combine superheat control and temperature-control features.

An additional feature to consider is the use of a liquid-line outlet regulator to maintain constant liquid pressure to the expansion valves. As long as subcooling is maintained consistently, the outlet regulator will not result in flash gas. Expansion valves fed with near constant temperature and pressure liquid have very little seasonal variation and result in more stable operation of the entire system.

RF13 *Loop Piping* (Climate Zones: all)

There are energy benefits in utilizing loop piping rather than single-circuited "homerun" piping due to the decreased heat gain in the suction lines and any subcooled liquid lines. For each suction group, loop piping (also called *trunk piping*) should be used. Loop piping means one suction line and one liquid line run refrigerant from each compressor group and then branches off closer to cases in a manner that minimizes total piping. Multiple loops of trunk piping per suction group can be used if required based on load proximity to the compressor or to avoid allowable pressure drop or velocity violations. Liquid loops from separate suction groups can and should be merged given they are feeding systems on the same compressor rack and require the same temperature of liquid.

RF14 *Piping Insulation* (Climate Zones: all)

Refrigeration piping insulation has typically used flexible closed-cell rubber or closed-cell plastic insulation material. Suction-line insulation has typically been based on the insulation required to prevent condensation and external frost build-up rather than being based on an economic thickness calculation, with a common wall thickness of 1/2 in. and sometimes 3/4 in. on low-temperature suction lines. Also, as a practical matter, with individual circuit piping to central compressor racks, the wall thickness affects the area taken up by piping.

While no specific insulation thicknesses are recommended, a greater insulation thickness should be considered with trunk piping. With far fewer and larger pipes, it is possible to consider insulation sizing based on economic thickness for energy savings, increasing thickness to 2 in. on low-temperature suction lines and 1 in. or greater on medium-temperature and subcooled liquid lines. Indirect CO_2 or glycol piping should have similar insulation levels.

Heat gain in direct-expansion (DX) system refrigerant lines and indirect system fluid lines can have significant disadvantages in both systems. Heat gain on a DX suction line is not a direct load; the heat gain does not increase mass flow but only decreases density—which causes compressors to run more to maintain mass flow. This is a different effect than heat gain has on an indirect cooling pipe, which results in a direct addition of refrigeration cooling load.

There can also be some heat gain in subcooled DX liquid lines, which increases mass flow requirements; however, there will always be more heat gain in indirect supply lines due to the larger pipe size and much colder temperatures.

Vapor retarders should also be considered in areas most susceptible to insulation failure due to harsh conditions such as ultraviolet (UV) exposure or extreme humidity and heat. With many users installing refrigeration systems on roofs, cold suction piping and cold hydronic piping (sometimes below freezing temperatures) are exposed to warm, humid outdoor air, which can adversely affect system efficiencies in the event of insulation failure. Vapor retarders can protect insulation from UV damage as well as keep moisture and ice from destroying the efficacy of the insulation. There are several options for vapor retarders, such as polyester and aluminum films, rubberized membranes, mastics, and others.

Additional information on insulation and vapor retarders can be found in RF32 in the Additional Bonus Savings section of this chapter.

DISPLAY CASES AND WALK-IN BOXES

RF15 *Case Efficiency* (Climate Zones: all)

Various refrigerated display case types are common in grocery stores and fit into the following several categories:

- Open or closed (with or without doors)
- Solid or transparent doors
- Low-temperature cases (frozen) or medium-temperature cases (refrigerated)
- Remote or self-contained condensing units
- Vertical, horizontal, semi-vertical, and service over counter

For all types of cases, the energy efficiency of the system can be expressed in total kilowatt-hours (kWh) per unit per day. Units vary depending on case type and can be expressed in terms of volume (V) or total display area (TDA) as determined by ANSI/AHAM HRF-1 (AHAM 2008) and AHRI Standard 1200 (AHRI 2013), respectively.

RF16 *Display Case Types* (Climate Zones: all)

Closed cases with glass doors are recommended for all upright low-temperature display cases as well as the following medium-temperature upright cases: dairy, deli, beverage, and packaged produce. Upright packaged red meat cases and wet rack produce cases are cases that for merchandising reasons normally require an open-style case. Doors are also recommended for horizontal open tub cases for frozen food and deli products and for any other open cases that can accept doors and where merchandising will allow them.

RF17 *Display Case Door Heaters* (Climate Zones: all)

For all low-temperature glass door display cases, specify 50 to 120 W/door for the combination of frame and door heaters. Specify frame heaters at no more than 55 W/door and use with either zero-heat glass doors or low-heat (no more than 65 W/door) door heaters. Use a pulse width modulating (PWM) control device to control the door heaters based on keeping the frame temperature of the door at a small delta (recommended 7°F) above dew point. Note that dew-point-based PWM controllers require a temperature sensor on the door frame along with both relative humidity and temperature sensors monitoring the surrounding environment.

For all medium-temperature applications, use zero-heat glass doors with no door or frame heating. Ideally, when reach-in doors are installed in walk-in applications, evaporator fans should not be directed at the backs of the glass doors, as cold air blowing directly on the door interiors will increase the TD across the door pane and could lead to sweating problems. If that design option is not possible and evaporator fans are directed at the backs of the glass doors, frame heaters may be required for medium-temperature applications. In that case, specify no more than 55 W/door for frame heaters.

Display Case and Walk-In Standards

Federal standards for refrigerated cases specify minimum energy efficiency levels for refrigeration display cases. New remote display cases are regulated by these standards, which define the energy efficiency of cases in total kilowatt-hours (kWh) per unit per day. Manufacturers' options and features, such as improved lighting, can improve efficiency beyond federal standards. The following table shows an example of the information in the standards for one type of display case.

Example Minimum Energy Efficiency Requirements for Display Cases

Equipment Category	Condensing Unit Configuration	Equipment Family	Rating Temperature, °F	Operating Temperature, °F	Maximum Daily Energy Consumption, kWh/day
Remote Condensing Commercial Refrigerators and Commercial Freezers	Remote	Vertical Open	38 (M)	≥ 32	0.82 x TDA + 4.07
			0 (L)	< 32	2.27 x TDA + 6.85
		Semivertical Open	38 (M)	≥ 32	0.83 x TDA + 3.18
			0 (L)	< 32	2.27 x TDA + 6.85
		Horizontal Open	38 (M)	≥ 32	0.35 x TDA + 2.88
			0 (L)	< 32	0.57 x TDA + 6.88
		Vertical Closed Transparent	38 (M)	≥ 32	0.22 x TDA + 1.95
			0 (L)	< 32	0.56 x TDA + 2.61
		Horizontal Closed Transparent	38 (M)	≥ 32	0.16 x TDA + 0.13
			0 (L)	< 32	0.34 x TDA + 0.26
		Vertical Closed Solid	38 (M)	≥ 32	0.11 x V + 0.26
			0 (L)	< 32	0.23 x V + 0.54
		Horizontal Closed Solid	38 (M)	≥ 32	0.11 x V + 0.26
			0 (L)	< 32	0.23 x V + 0.54
		Service Over Counter	38 (M)	≥ 32	0.51 x TDA + 0.11

Note: Adapted from the standards for commercial refrigeration equipment summarized by the U.S. Department of Energy (DOE) at www1.eere.energy.gov/buildings/appliance_standards/product.aspx/productid/52 (DOE 2015)

New walk-in coolers and freezers are currently subject to federal energy efficiency requirements under Section 314 of the *Energy Policy and Conservation Act*. Under this law, the U.S. Department of Energy (DOE) defines the following:

- Minimum values for wall, roof, and floor insulation
- Door closers and methods to reduce infiltration
- Evaporator fan motor requirements
- Lighting efficacy
- Requirements for transparent doors and anti-sweat heaters

A separate performance-based requirement in these standards includes amendments made in a DOE ruling on August 4, 2014. This ruling requires compliance as of June 5, 2017, and adopts amended energy conservation standards for the main components of walk-in coolers and freezers, refrigeration systems, panels, and doors. These standards are expressed in terms of annual walk-in energy factor for walk-in refrigeration systems, R-value for walk-in panels, and maximum energy consumption for walk-in doors.

Table 5-12 Walk-In Insulation Criteria

Type	Thickness	K-Factor	U-Factor	R-Value
Coolers	3.5 in.	0.141	0.040	R-25
Freezers	5.0 in.	0.125	0.025	R-40

RF18 *Walk-In Box Construction* (Climate Zones: all)

Defining specifications for walk-in coolers and freezers is different than for cases because walk-ins are normally custom built on-site, whereas cases are completely constructed in a factory and only “set” on site. There are, however, factory-manufactured components to walk-in boxes, and there is an ability to specify how those components are built and also how they are put together or built on site.

Walk-in boxes fall into several categories:

- Small walk-in coolers/freezers without customer access doors and with typical ceiling heights of 7 to 8 ft that are generally found in back-of-house to support bakeries and deli areas.
- Medium walk-in coolers/freezers without customer access doors and with typical ceiling heights of 9 to 10 ft that are generally found in back-of-house to support meat, dairy, and produce departments.
- Small to medium walk-in coolers/freezers with customer access doors, sales floor product presentation, and backroom access for stocking. Typical ceilings for these units are 7 to 8 ft.
- Large walk-in coolers/freezes with or without customer access doors and backroom access for stocking. Typical ceiling heights are 12 to 16 ft and may go as high as 20 ft.

Floors. Walk-in box floor construction is discussed in EN12.

Insulation. The at-temperature conductivity factor (K factor), the overall coefficient of heat transfer (U-factor), and the R-value of walk-in box insulation should be constructed as shown in Table 5-12.

Infiltration barriers. Overlapping strip curtains or vinyl swinging doors are recommended for all walk-in doors to prevent air and moisture infiltration while the main door is open. Installations should cover the entire opening to within 1/4 to 1/2 in. of the floor threshold.

RF19 *Walk-In Box Doors* (Climate Zones: all)

Walk-in box doors are generally constructed of the same material used in walk-in box insulated wall panels. A magnetic compression gasket is recommended for the two sides and top edge of the door seal with a wiper-style gasket to seal between the door bottom and the threshold. Automatic door closers should be used for all doors. Swinging doors that latch upon closure are recommended for all applications to facilitate automated closing devices.

Hinges and closers. For all doors less than 48 in. in width, use cam-lift self-closing gravity hinges or a hinge set with spring-assisted operation. For all doors greater than 48 in. in width, a spring-action door closer is recommended. All doors should use a hydraulic door closer for securing the door against the gasket. This provides a “snap” closure when the door is open less than 1 in.

For produce prep areas and other high-temperature walk-ins, a double-action swinging door may be specified to facilitate higher traffic volume. In these cases, gravity hinges are recommended along with flexible wiper seals on all four sides.

RF20 *Walk-In Box Door Switches and Alarms* (Climate Zones: all)

Walk-in box freezer and cooler doors are opened more frequently and are left open for longer periods of time than necessary. Over time, self-closing doors may fail to fully latch on their own, allowing them to be left slightly open for extensive periods of time. Door alarms remind employees to keep doors shut and notify them when a door has not been completely shut or latched. This helps decrease the amount of time doors are left open, which in turn reduces energy use due to reduced box loads. (These door alarms should not be confused or combined with the very important leak detection alarms required for safety reasons.)

Each latching walk-in box door should be installed with a door switch to sense when the door is open. The recommendation is that whenever a door is opened, the refrigerant flow to the coil should be stopped (typically with a liquid-line solenoid) and the evaporator fans should be turned off to minimize air circulation—and therefore infiltration. Safeties should be in place to turn the refrigerant flow and evaporator fans back on in the event that the door is left open for too long in order to prevent product loss.

The most effective alarms have both visual (e.g., strobe light) and audible (e.g., horn) alarming functions. When a door first opens, the visual alarm should activate. After a predetermined time limit, the audible alarm should be activated. Once the door is closed, the alarms should turn off and reset. More or less aggressive alarming sequences can be applied to maximize effectiveness with respect to different owners' operations. The Examples of Walk-In Freezer and Cooler Doors, Hinges, Closers, and Alarms sidebar shows a typical door alarm and door switch.

RF21 *Walk-In Box Fan Control* (Climate Zones: all)

Dual-speed evaporator fan motors are recommended with two-speed electronically commutated (EC) motors set for 100% and 80% speed. Speed control should be based on the position of the evaporator's electronic expansion valve (EEV). When the valve is in a greater than 50% open position, the fan motors run at 100%, and when the EEV is operating at 50% or less, the fan motor ratchets back to the 80% speed mode.

Examples of Walk-In Freezer and Cooler Doors, Hinges, Closers, and Alarms

Double-Action Swinging Doors

Door with Overlapping Strip Curtains

Cam Lift Self-Closing Gravity Hinges

Hydraulic Door Closer

Spring-Action Door Closer

Horn and Strobe Door Alarms (left and center) and Leak-Detection Strobe Alarm (right)

Photographs published with permission of Target Corporation

RF22 *Display Case and Walk-In Lighting Design* (Climate Zones: all)

All display cases should be equipped with the most efficient LED light option. All glass door cases should include individual motion sensors to turn off the lights after two to three minutes of inactivity near the display case. This same control feature should be used for reach-in doors attached to walk-in boxes, such as dairy and beverage walk-ins or other point-of-sale boxes.

Display case lighting should be a maximum of 10 W/ft except for meat cases, which should be a maximum of 25 W/ft. Typically, open cases will have two rows of canopy LEDs, whereas open meat cases will also have shelf lighting. Closed cases typically have vertical LED lamps between the doors as well; they may also have lighting above and below the doors if they can meet the specified wattage requirements.

Walk-in lighting. Vaportight LED lighting fixtures with motion/vacancy sensors and/or door trigger controls are recommended. Specific details on lighting requirements for walk-in boxes are outlined in EL25.

RF23 *Display Case and Walk-In Defrost* (Climate Zones: all)

While electric and hot-gas defrost are both used frequently for many different reasons, electric defrost is recommend where time-off air-defrost is not sufficient. However, if implemented poorly, electric defrost can perform worse than hot-gas defrost from an energy perspective. It is important to always apply electric defrost with a temperature termination. This can be accomplished either by adding a separate defrost termination temperature sensor or by adding small defrost termination thermostats to the coils. The temperature rise of the coil while in defrost can be sensed after all the ice is cleared from the coil, at which point the defrost cycle can be terminated.

Electric defrost should also have a timed termination programmed into the controller in the event that the system fails to terminate by temperature. More aggressive approaches to initiate defrost are possible, but due to the critical need to always have an efficiently operating coil that is free of ice and due to the lack of a mainstream, accepted method for a demand defrost technology, defrost should be initiated based on time and defrost cycles should be allowed to be shorter when possible.

Additional discussion on the pros and cons of electric versus hot-gas defrost is provided in the Electric versus Hot-Gas Defrost section in Chapter 3.

RF24 *Display Case and Walk-in Temperature and Superheat Control* (Climate Zones: all)

The recommended temperature control in remote display cases and walk-ins is electronic modulating control integrated with floating suction pressure control.

One method to achieve this is to use electronic evaporator pressure regulators (EEPRs) on each case and walk-in line-up in conjunction with a compressor rack controller capable of optimizing suction pressure setpoint based on EEPR valve position. Specifically, if all valves are less than 100% open, the suction pressure setpoint should be slowly incremented to a higher setpoint. If one or more valves remain 100% open for more than a few minutes, the setpoint should be incremented down to provide an SST sufficient to maintain the case or walk-in temperature. In this way, the compressor group operates at the highest pressure possible while still maintaining case and walk-in temperature. (See the example in Figure 5-29.)

The EEPR valves also have the advantage of not requiring a high-pressure connection such as is common with conventional mechanical evaporator pressure regulator (EPR) valves, which is one reason the latter are commonly mounted at compressor racks. EEPRs can easily be located near the case line-up or walk-in or distributed in groups throughout the store. This flexibility of location supports more efficient and cost-effective trunk piping in lieu of “home run” piping.

An alternative method, far less common but with potentially greater savings, is to use electronic expansion valves (EEVs) for temperature control, with similar logic used in the compressor rack controller. EEVs can control temperature, in addition to their primary function of

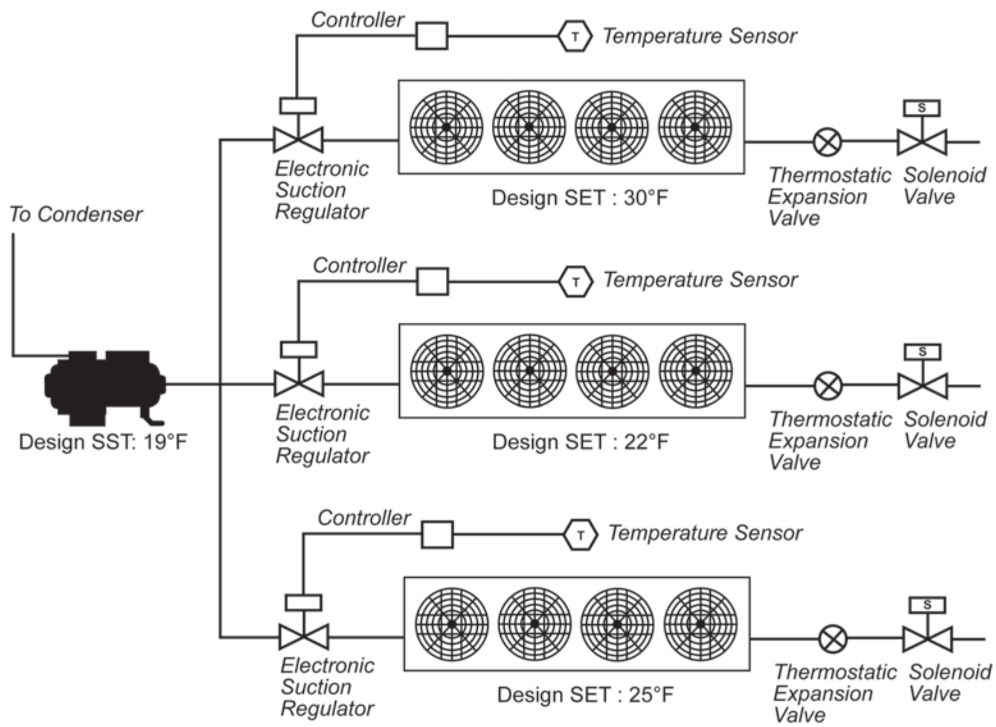


Figure 5-29 (RF24) Example Design with EEPR Valves

controlling superheat, either by cycling on/off when temperature setpoint is obtained or by modulating to a more closed position and thus a higher superheat. This strategy has the benefit of increasing superheat on noncritical case line-ups and walk-ins, thus creating a higher productive enthalpy difference and reducing mass flow that the compressors must pump. This is particularly advantageous on display cases with close-approach evaporators and when using refrigerants with substantial temperature glide.

RF25 *Unit Coolers* (Climate Zones: all)

All walk-in box unit coolers should be specified with EC motors. Additionally, a TD of 8°F is recommended to minimize compressor lift at the rack. However, many times within a suction group there will be one circuit with evaporating temperatures lower than the others operating on the same suction group. These evaporators are often considered “critical” since they are the coldest and dictate what the suction pressure of the entire suction group needs to operate at. When possible, the design TD on these unit cooler evaporators should be minimized in order to increase the actual evaporating temperature required to meet product temperatures. Doing so will allow for the entire suction pressure to rise, reducing compressor lift.

For freezer unit coolers, 4 FPI is recommended to minimize rapid capacity degradation due to ice formation. In certain instances where there are physical limitations for coil placement, FPI should be adjusted before adjusting TD.

RF26 *Liquid-Suction Heat Exchangers* (Climate Zones: all)

Liquid-suction heat exchangers (LSHXs) should be used on display case line-ups and walk-ins to provide additional subcooling. Size the heat exchangers, at design conditions, to provide at least 12°F of additional subcooling on low-temperature circuits and at least 6°F of additional subcooling on medium-temperature circuits. The LSHX should be located as close as possible to the piping exit of the walk-in or case line-up. Securely clamp the heat exchanger, particularly on the liquid line, to avoid stress on the connections from piping expansion and

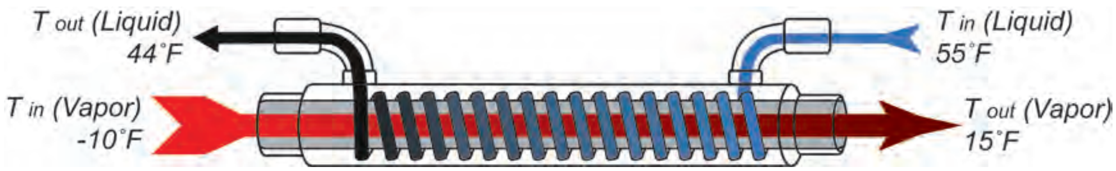


Figure 5-30 (RF26) LSHX and Typical Approach Temperatures

contraction that could lead to breakage. LSHXs are applicable to halocarbon systems; use on CO_2 DX systems has not been evaluated but may potentially be considered on the advice of the system original equipment manufacturer (OEM).

LSHXs improve efficiency and increase refrigeration system capacity when properly applied. While small heat exchangers are often supplied as standard or an option in display cases and sometimes used on walk-ins, there is more potential available by using larger heat exchangers designed to obtain significantly more heat exchange. (See Figure 5-30.)

Particularly with high-efficiency cases that have larger coils and lower suction superheat, and with refrigerants that have glide (boiling point transition) characteristics, the superheat leaving the refrigerated space often is very low and in fact there can be "wet" suction conditions with a fraction of liquid leaving the case or box. This low superheat or liquid carryover means the compressor must pump more refrigerant than necessary. A liquid suction heat exchanger adds suction superheat and subcools the liquid refrigerant, thus increasing the Btu/lb cooling capacity of the refrigerant.

HEAT RECOVERY

RF27 *Refrigerant Heat Recovery—Service Water Heating* (Climate Zones: All)

Refrigerant heat recovery for domestic hot-water heating is covered in WH6. Because water heating requires temperatures well above the refrigeration system condensing temperature, the heat is only recovered from the superheat portion of the refrigeration heat of rejection. Superheat is a small fraction of the available heat. In addition, the actual discharge temperature of a system varies with the type of refrigerant and the system application. A medium-temperature system using R-507, at average head pressure, may have an actual (superheated) discharge temperature of 115°F or lower. A low-temperature system would have a significantly higher actual discharge temperature and would be more suitable for water heat recovery.

The heat removed through desuperheating to preheat domestic hot water is a small fraction of the total heat of rejection, and the remaining available heat obtained through condensing is still available for space heating. Care must be taken to minimize discharge pressure drop, but it is possible to use a system for both service water heating (SWH) and space heating.

RF28 *Refrigerant Heat Recovery—Space Heating* (Climate Zones: All)

Refrigerant heat recovery for space heating offers large energy savings in many climates. A minimum of 25% of the design heat of rejection is recommended, even though in most instances a higher percentage is feasible and practical. Heat recovery system design varies depending on climate, type of refrigeration system (e.g., central or distributed), and the type of HVAC system. Also, the type of store affects heating needs, based on the exhaust air volume and the density of refrigerated display cases. Even with the many combinations of refrigeration and HVAC system types, numerous configurations are possible in order to obtain at least 25% heat recovery, including the following:

- Central compressor racks with direct condensing heat recovery in a central air-handling unit (AHU)
- Distributed compressor racks with direct condensing to a central air handler or adjacent rooftop units (RTUs)

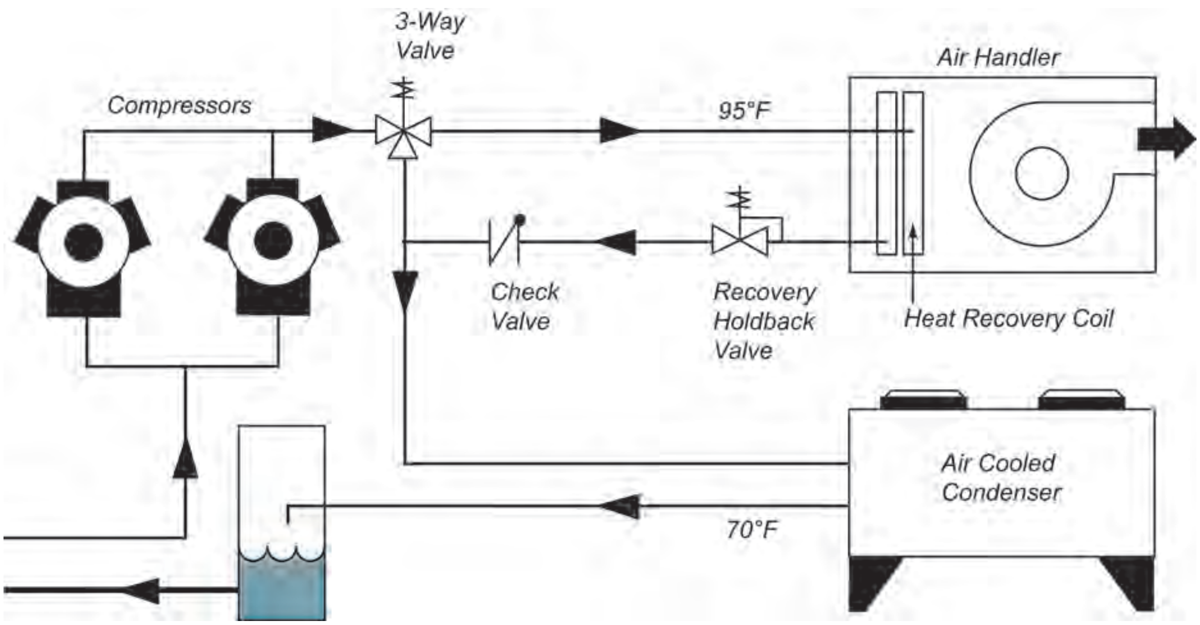


Figure 5-31 (RF29) Conventional Series-Connected Heat Recovery

- One or more refrigeration systems using water-cooled condensers to transfer heat to a water loop, which is then connected to hot-water heating coil(s) in a central AHU or multiple RTUs
- Refrigeration system(s) connected to dedicated outdoor air systems (DOASs)
- Distributed refrigeration systems connected to "unit heater" style heat recovery fan-coils in stockroom or receiving areas

A portion of the heat of rejection is available as superheat, with temperatures ranging from 120°F to 150°F (depending on conditions and refrigerant) down to the condensing temperature; this is only 15% to 20% of the available heat. To obtain more substantial quantities of heat and justify the heat recovery system costs, the system must be designed for condensing in addition to desuperheating.

RF29 *Conventional Heat Recovery Design* (Climate Zones: all)

Figure 5-31 shows a conventional heat recovery design with a three-way diverting valve, a heat recovery coil (which is essentially a condenser coil), and a holdback valve. The heat recovery coil is piped in series with the outdoor condenser such that all heat not rejected in the air handler is rejected outdoors. As shown in the figure, the holdback valve allows the pressure and condensing temperature in the heat recovery coil to be set at a value (e.g., 95°F SCT) sufficiently above the entering air temperature to establish a TD and allow condensing. The holdback valve also allows the outdoor condenser to operate at a lower condensing temperature, in fact using the same control point whether the system is in heat recovery mode or not. From an efficiency standpoint, this means the refrigerant will still benefit from floating head pressure, in terms of the cooling effect per pound of refrigerant pumped by the compressor. The penalty at the compressor can be considered an increase in power per unit mass flow.

RF30 *Other Heat Recovery Configurations* (Climate Zones: all)

The variety of refrigeration systems and HVAC systems results in numerous possible heat recovery system configurations. Refrigeration systems can be centralized or distributed and

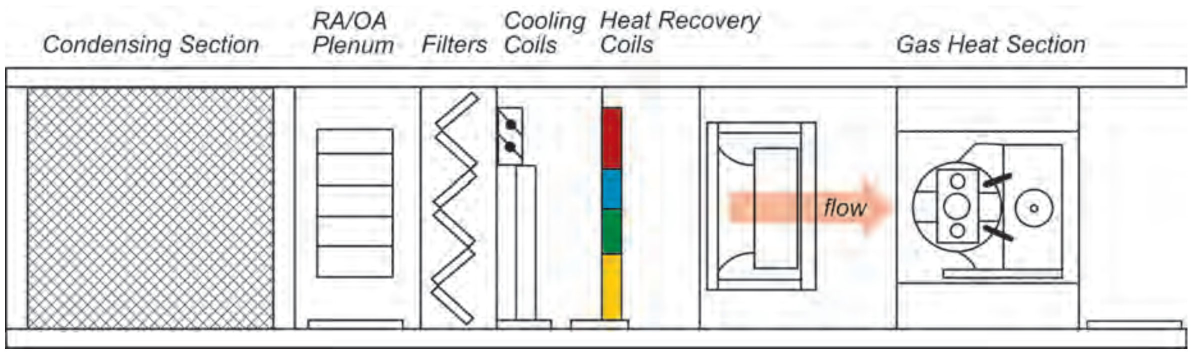


Figure 5-32 (RF29, RF30) **Heat Recovery Coil in a Large RTU**

HVAC systems can be centralized or distributed. Many configurations could be considered to use available heat, for example:

- Central rack systems connected directly to a central air handler or rooftop package.
- Multiple distributed refrigeration systems directly connected to a central rooftop package. Figure 5-32 shows a heat recovery coil in a large custom HVAC RTU of a type commonly used in grocery stores.
- Distributed refrigeration units connected to nearby distributed RTUs.
- Distributed refrigeration units connected via water loop to a central air handler or to multiple rooftop units.
- Distributed refrigeration unit connected to a horizontal condenser "unit heater" in a stockroom.

RF31 *Heat Recovery Applications* (Climate Zones: all)

A key configuration and design decision is whether heat recovery is dedicated to heating ventilation air or if heating is applied to the mixed airstream. There are many factors to consider, as follows:

- If return air is to be heated, use of low return air ducting is necessary, since the cool air (often on the order of 65°F) allows heat to be recovered without excessive penalty to the refrigeration system. This type of design is generally not feasible for stores that use multiple RTUs; however, low return air is common practice for many chains (in conjunction with AHUs or central packaged units) as a means to reduce stratification and increase store comfort.
- The heating balance point (the outside temperature below which the store requires heating) can be quite high in many conventional grocery stores with intensive refrigeration, even as high as 80°F, resulting in heating for most hours of the year.
- As stores are designed with more glass doors in lieu of open cases, the peak heating load and annual heating energy requirements will reduce, which affects the economics of heat recovery and potentially the design approach.
- Ventilation air volume and climate determine whether the available heat can be used strictly for heating ventilation air. To the extent this is free heat, using heat recovery and employing "transfer air" to supply makeup to exhaust fans may be the best combination for many stores. The many variables (ventilation air design volume, how it is controlled through the year, and hourly ambient temperatures) determine how much heat can be used.

Figures 5-33 and 5-34 show an example heat recover design with a water loop and multiple distributed refrigeration units and a central air handler with series connection to provide low TD at each refrigeration unit and high TD at the AHU to achieve a suitable air temperature rise.

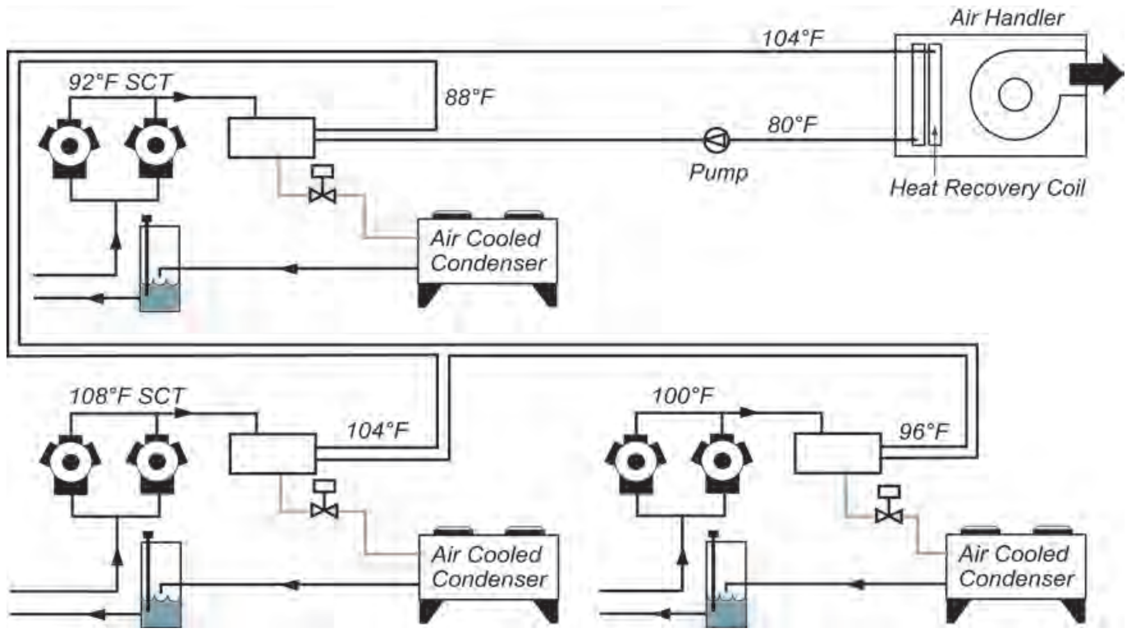


Figure 5-33 (RF31) Example Heat Recovery Design with Three Units in Series

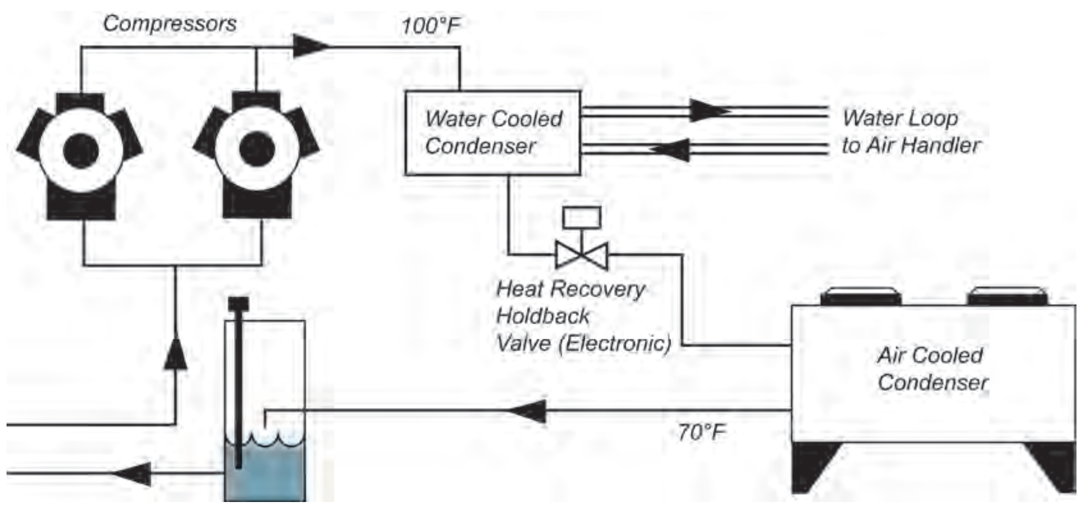


Figure 5-34 (RF31) Example Heat Recovery Unit Using Water Loop and Electronic Holdback Valve

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SERVICE WATER HEATING

SERVICE WATER HEATING SYSTEM TYPES

WH1 *System Descriptions* (Climate Zones: as indicated below)

Several different types of water heating systems are used in grocery stores. System selection depends upon fuel availability, fuel cost differentials in the location, and the magnitude of the loads to be served. Systems considered in this Guide are as follows:

Gas-fired storage water heater (Climate Zones: all): A water heater with a vertical or horizontal water storage tank. A thermostat controls the delivery of gas to the heater's burner. The heater requires a vent to exhaust the combustion products. An electronic ignition is recommended to avoid the energy losses from a standing pilot.

Gas-fired instantaneous water heater (Climate Zones: all): A water heater with minimal water storage capacity. Such heaters require vents to exhaust the combustion products. An electronic ignition is recommended to avoid the energy losses from a standing pilot. Instantaneous, point-of-use water heaters should provide water at a constant temperature regardless of input water temperature or flow rate.

Electric resistance storage water heater (Climate Zones: all): A water heater consisting of a vertical or horizontal storage tank with one or more immersion heating elements. Thermostats controlling heating elements may be of the immersion or surface-mounted type.

Electric resistance instantaneous water heater (Climate Zones: all): A compact undercabinet or wall-mounted water heater with an insulated enclosure and minimal water storage capacity. A thermostat controls the heating element, which may be of the immersion or surface-mounted type. Instantaneous, point-of-use water heaters should provide water at a constant temperature regardless of input water temperature or flow rate.

> ***Caution:*** These water heaters can save energy for small loads, such as restroom sinks, but larger systems should be avoided, as they can create demand charges from the spike in electrical load to service the hot-water heater.

Heat pump electric water heater (Climate Zones: ❶ ❷ ❸): A storage-type water heater using rejected heat from a heat pump as the heat source. Water storage is required because the heat pump is typically not sized for the instantaneous peak demand for service hot water, even in a grocery store. The heat source for the heat pump may be the interior air (such as that from a motor or switching room), which is beneficial in cooling-predominant climates; the circulating loop for a water-source heat pump (WSHP) system, also beneficial in cooling-dominated climates; or a ground-coupled hydronic loop. These types of water heaters should be considered for projects as alternatives to an electric resistance tank type water heater. Heat pump water heaters should have a COP of at least 3.0.

Where electricity is the preferred energy source for SWH, consider specifying an ENERGY STAR rated heat pump water heater for additional energy savings.

GENERAL RECOMENDATIONS

Good Design Practice

WH2 *Sizing* (Climate Zones: all)

The water heating system should be sized to meet the anticipated peak hot-water load. For a grocery store without food service, hot-water usage may be limited to hand washing by customers and employees, which is a very small load. Calculate the demand for each water heater based on the fixture units served by the heater according to local code.

While hot-water temperature requirements for the restrooms and break rooms of a grocery store vary by local and state codes within the range of 100°F to 120°F, note that production of service hot water at temperatures below approximately 135°F may result in bacterial growth within storage-type water heaters.

WH3 *Equipment Efficiency* (Climate Zones: all)

Efficiency levels are provided in this Guide for gas-fired tankless, gas-fired storage, electric resistance tankless, and electric resistance storage water heaters. Residential water heaters typically are rated for the energy factor (EF) that reflects the ratio of the heat added to the delivered hot water to the total thermal input to the heater over a prescribed schedule of hot-water delivery. Commercial tank-type water heaters, which will likely have a far different schedule of hot-water delivery, are currently rated by thermal efficiency (E_t) and standby heat loss. Standby heat losses are dependent upon tank volume and configuration in addition to jacket insulation value and are typically established by a standardized testing procedure.

For commercial gas-fired storage water heaters, the standby loss criteria is given by the following equation:

$$\text{Standby Loss (Btu/h)} \le 0.84 \times [(\text{Gas Input (Btu/h)}/800) + 110 \times \text{Tank Size (gal)}^{0.5}]$$

For gas-fired instantaneous water heaters, the EF and E_t levels are nearly the same because there are no standby losses. The incorporation of condensing technology is recommended for all gas-fired water heaters to achieve a minimum E_t of 94%.

Table 5-13 gives performance requirements for residential and commercial gas-fired water heaters of various capacities and sizes.

The levels of performance specified in this Guide for gas water heaters require that the units be of the condensing type, not only recovering more sensible heat from the products of combustion but also recovering heat by condensing moisture from these gases. The construction of a condensing water heater as well as the water heater venting must be compatible with the acidic nature of the condensate for safety reasons. Disposal of the condensate should be done in a manner compatible with local building codes.

Efficiency metrics for residential and commercial high-efficiency electric storage water heaters are also provided in this Guide. These efficiency metrics represent premium products that have reduced standby losses. Table 5-14 summarizes required EFs for residential water heaters and standby loss limits for commercial water heaters. Standby losses are expressed as a percentage of the total heat energy stored in the water heater tank that is lost over the course of an hour. For commercial electric storage water heaters, the allowable jacket losses are given by the following equation:

$$\text{Standby Loss (\%/h)} \le 0.30 + 27 / \text{Tank Size (gal)}$$

Electric instantaneous water heaters are a more efficient alternative to high-efficiency storage water heaters for very-low-volume distributed end uses because they have no tank losses.

Table 5-13 Gas Water Heater Performance

Storage Volume, gal	Capacity, kBtu/h	EF (Residential)	E_t, % (Commercial)	Standby Loss, Btu/h (Commercial)
0	Varies	0.93	0.94	NA
30	40	0.75	NA	NA
40	40	0.75	NA	NA
50	60	NA	0.94	841
65	75	NA	0.94	966
75	100	NA	0.94	1058
80	125	NA	0.94	1115
120	250	NA	0.94	1467

Table 5-14 Electric Water Heater Performance

Storage Volume, gal	EF (Residential)	E_t, % (Commercial)	Standby Loss, %/h (Commercial)
0.0	0.98	0.98	NA
30	0.96	0.98	1.2
40	0.96	0.98	0.98
50	0.95	0.98	0.84
65	0.95	0.98	0.72
75	0.94	0.98	0.66
80	0.94	0.98	0.64
120	0.93	0.98	0.53

In addition, they can be located near the end use, minimizing pipe losses. However, their impact on building peak electric demand may be significant and should be taken into account during design. Where unusually high hot-water loads (e.g., dishwashers) are present during periods of peak electrical use, electric storage water heaters are recommended over electric instantaneous water heaters.

WH4 *Minimizing System Losses* (Climate Zones: all)

Grocery store SWH requirements consist of low-volume distributed end uses, such as sinks, and high-volume end uses associated with prepared foods and bakery functions. To achieve quick hot-water response, many SWH systems use a pumped return (recirculation) to ensure immediate hot-water delivery. For low- and moderate-volume end uses, the circulation heat loss through the piping may outweigh the actual energy consumption for producing the required hot water. The problem worsens as the distance increases between the hot-water generator and the loads.

The primary water heater should be located close to the hot-water fixtures and hot-water-using equipment (e.g., dish machines) to avoid the use of a hot-water return loop or of heat tracing on the hot-water supply piping. Potential locations for gas-fired water heaters may be limited by flue and combustion air and local code requirements.

Accommodation of renewable or free heat sources necessitate a centralized SWH system, typically with a tank to accommodate the asynchronicity of heat sources with hot-water demand. These systems should be located immediately adjacent to the end use to avoid recirculation and its attendant losses. If the provision of a centralized service water heater requires the use of a recirculation pump and hot-water return pipe to ensure prompt delivery of hot water at the end use, the supply and return pipes should be insulated as discussed in WH6. Furthermore, in most grocery applications, the recirculation pump should be controlled by a time clock to disable it when hot-water end use is not active.

An alternative is to incorporate a control strategy that operates the recirculation pump only when there is a need for hot water, otherwise known as *demand circulation*.

Low-volume end uses, including handwashing sinks, janitorial closets, and implement-washing sinks in the meat and produce preparation areas likely do not justify service from a central source with required recirculation and should be served with point-of-use water heaters.

WH5 *Pipe Insulation* (Climate Zones: all)

All SWH piping should be installed in accordance with accepted industry standards. Required pipe insulation thickness should vary with the temperature of the water, the size of the pipe, and the thermal resistance of the insulation material per Table 5-15.

The insulation should be protected from damage and should include a vapor retarder on the outside of the insulation.

Table 5-15 Minimum Piping Insulation Thicknesses for SWH Systems

Fluid Operating Temperature Range, °F	Insulation Conductivity		Nominal Pipe Size		
	Conductivity, Btu·in./h·ft^2·°F	Mean Rating Temperature, °F	< 1 in.	1 to < 1 1/2 in.	≥ 1 1/2 in.
141°F to 185°F	0.25 to 0.29	125	1.5	1.5	2.0
105°F to 140°F	0.22 to 0.28	100	1.0	1.0	1.5

For insulation outside the stated conductivity range, the minimum thickness (T) shall be determined as follows: $T = r\{(1 + t / r)^{K/k} - 1\}$, where T = minimum insulation thickness (in.), r = actual outside radius of pipe (in.), t = insulation thickness listed in this table for applicable fluid temperature and pipe size, K = conductivity of alternate material at mean rating temperature indicated for the applicable fluid temperature (Btu·in./h·ft^2·°F), and k = the upper value of the conductivity range listed in this table for the applicable fluid temperature.

Data source: ASHRAE/IES Standard 90.1-2013, Table 6.8.3-1

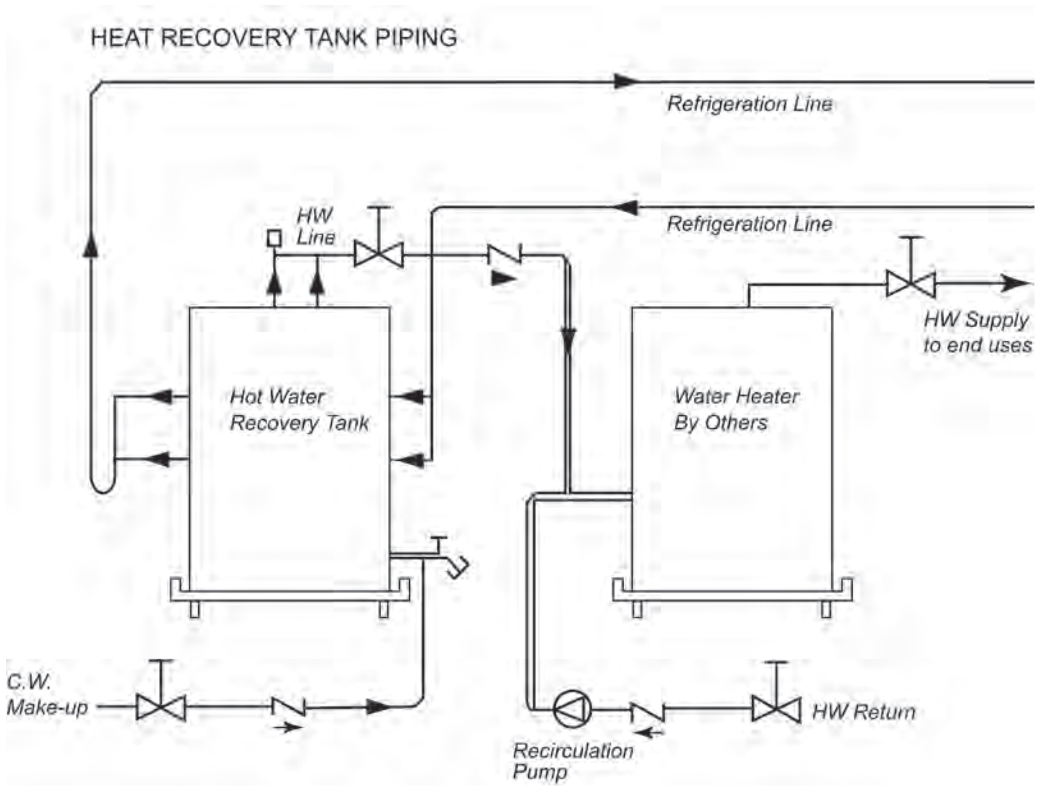

Figure 5-35 (WH6) Refrigerant Superheat Recovery Tank Piping for Service Hot-Water Preheat

WH6 Refrigerant Heat Recovery

A refrigerant-to-water tank-type heat exchanger can be installed between the host refrigeration system's compressor and condenser to recover heat for use in preheating service hot water. Incoming cold makeup water from the city supply is routed through the heat exchanger on its way to the primary service water heater. While in the heat exchanger tank, the hot compressed gas from the refrigeration condenser is cooled almost to its condensation temperature, transferring this heat to the water, hence the name *desuperheater*. The refrigeration condenser unit completes the process of condensing the hot gas and rejects this heat to the outdoor environment. Under typical conditions a desuperheater can remove 10% to 30% of the total heat that would have been rejected by the condenser.

As shown in Figure 5-35, the water path through the desuperheater is once through; there is no recirculation of water from the water heater back into the desuperheater tank.

A desuperheater reclaim system can be used with distributed point-of-use water heaters if a separate line for tempered water is run from the central desuperheater tank to each of the point-of-use water heaters. A recirculation line connected upstream of the point-of-use water heater could keep water temperature in the line close to the discharge temperature of the desuperheater tank. Heat losses from the tempered water line to the space would be substantially less than losses from a hot-water line from a central water heater to the remote end use. See also RF27.

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HVAC SYSTEMS AND EQUIPMENT

GENERAL INFORMATION

Although many types of HVAC systems could be used in grocery stores, this Guide prescriptively covers the following five system types, each of which has demonstrated applicability to this building type and the ability to meet the 50% savings criteria through extensive energy modeling:

- Mixed-air (MA) single-zone variable-air-volume (SZVAV), direct-expansion (DX) packaged rooftop air-conditioning unit (RTU) with indirect gas furnace or electric resistance heat and hot-gas reheat (HGR) for dehumidification control (see HV4).
- Mixed-air (MA) single-zone variable-air-volume (SZVAV), chilled-water packaged rooftop air-conditioning unit (RTU) served by an air-cooled chiller and indirect gas furnace or hot-water coil served by a high-efficiency boiler with heat for dehumidification control (see HV5).
- Single-zone variable-air-volume (SZVAV), direct-expansion (DX) packaged rooftop air-conditioning unit (RTU) with indirect gas or electric heat integrated with a dedicated outdoor air system (DOAS) (this Guide uses *DOAS* and *100% outdoor air system [100% OAS]* interchangeably) (see HV6).
- Single-zone constant volume (SZCV) air-source heat pump (HP), packaged rooftop air-conditioning unit (RTU) with electric resistance supplemental heat integrated with a dedicated outdoor air system (DOAS) (see HV7).
- Single-zone variable-air-volume (SZVAV) water-source heat pump (WSHP), packaged rooftop air conditioning unit (RTU) with cooling tower heat rejection and boiler backup heating for the circulating fluid loop and a dedicated outdoor air system (DOAS) for ventilation air (see HV8).

Unique recommendations are included for each HVAC system type in the climate-specific recommendation tables in Chapter 4. It is noted that in some climate zones, achievement of the 50% savings criteria is dependent on higher-efficiency components.

Good Design Practice

HV1 ***Heating and Cooling Loads*** **(Climate Zones: all)**

Heating and cooling system design loads used to size systems and equipment should be calculated in accordance with generally accepted engineering standards and handbooks such as *ASHRAE Handbook—Fundamentals* (ASHRAE 2013a). Heating and cooling loads should be calculated for the specific project, incorporating local design weather conditions, specifics of the actual building, and the impact of any energy conservation measures that reduce heating and cooling loads, including ENERGY STAR appliances, efficient lighting, and daylight-responsive lighting controls. For grocery store applications, the cooling effect associated with refrigerated display cases (case credits) and/or the additional heating load associated with self-contained refrigerated display cases (integrated heat rejection to surrounding space) can significantly impact the building heating and cooling load profile. Further, the potential "cold aisle" effects created from adjacent display cases should be considered in the design for air distribution effectiveness as well as the potential of creating heating and cooling conflicts within the system. As a result of these considerations, sales areas containing predominantly refrigerated cases should be served as a separate thermostatic control zone from sales areas without cases or back-of-house areas. HVAC units serving areas with refrigerated cases should be equipped with hot-gas bypass or heat recovery reheat to maintain appropriate apparatus dew-point temperatures required for space humidity control without overcooling the served space.

Safety factors should be applied cautiously and only to internal loads to prevent oversizing of equipment. For grocery stores that are part of a large portfolio, established precedent with measured internal loads may eliminate the necessity of safety factors. In addition, the incorporation of EC motors will result in reduced HVAC loads. Thus, careful attention to new loads is important; even precedent from a large portfolio will provide its own safety factor. If an air-conditioning unit with a constant-speed compressor is oversized and the cooling capacity reduction is limited, short-cycling of compressors could occur and the system may not have the ability to dehumidify the building properly, causing moisture problems; in addition, oversized equipment may operate less efficiently.

HVAC system and equipment sizing should incorporate the following considerations:

- Include the capacity for meeting-space loads, break rooms (that may including vending machines), information technology (IT) equipment rooms, and other miscellaneous space loads.
- Include the capacity for cooling and dehumidifying the required maximum flow of ventilation air.
- Include the capacity for heating the minimum flow of outdoor ventilation air. See HV16. This heating component must be included in the overall heating load calculation.
- Recognize that the driver for space conditions may be the ambient environment required by the refrigerated case manufacturer's submittal data to ensure case performance as opposed to the comfort limits of the occupants. Further, the design should recognize the impact of the refrigerated display cases on the interior load profiles for the sales space.
- Recognize that the refrigerated cases may reduce sensible loads significantly; further, the building HVAC system should perform the dehumidification required to achieve the ambient conditions stated in the refrigerated case submittal data. This often results in a lower sensible heat ratio (SHR) than that of other building occupancies.
- Take into account the reduction of cooling and heating loads, and the subsequent reduction of mechanical equipment size, when energy recovery is used for ventilation air.
- Ensure that mixed-air variable-air-volume (VAV) systems are configured to maintain minimum required ventilation airflow during all operating conditions in accordance with ANSI/ASHRAE Standard 62.1 (ASHRAE 2013b) and that space dew-point temperatures can be sustained.

HV2 Certification of HVAC Equipment (Climate Zones: all)

Rating and certification by industry organizations is available for various types of HVAC equipment. In general, certification is provided by industry-wide bodies that develop specific procedures to test the equipment to verify performance; ASHRAE/IES Standard 90.1 (ASHRAE 2013c) has requirements for units for which certification programs exist. Certifications that incorporate published testing procedures and transparency of results are much more reliable for predicting actual performance than are certifications that are less transparent. For types of equipment for which certification is available, preference should be given to certified products. Examples of equipment types that have recognized certifications include packaged heat pumps, packaged air-conditioning units, gas furnaces and boilers, fluid coolers, and water heaters.

For products for which certification is not available or those that have not been subjected to certification available for their type of equipment, the products should be rigorously researched for backup of performance claims made by the supplier. The project team should determine by what procedure the performance data was developed and establish any limitations or differentials between the testing procedure and the actual use.

Determining the actual operational efficiency of products may require information beyond that provided by the certification process. For example, the relative impact on annual energy performance of the various components of packaged air-conditioning equipment (i.e., supply fans, refrigeration circuits, heat rejection components) changes based on the actual requirements of the HVAC load. The certification of the packaged unit, however, standardizes the contribution of these components to the rating.

HV3 *Integration of Commercial Refrigeration with HVAC* (Climate Zones: all)

Commercial refrigeration is the largest single energy end-use in grocery stores. In maintaining food storage and presentation areas at low temperatures, the refrigeration systems continually reject a large amount of heat. The rejected heat from these systems and the additional cooling capacity added beyond that required for preservation of food can be used as an alternative heating source for the HVAC systems of the building.

The effectiveness of the integration strategy varies with climate zone and with the resulting temperature quality available for heat rejection from the refrigeration and the coincident heating requirements for ventilation and/or the occupied space. In general, heat from commercial refrigeration is recovered at a relatively low temperature to minimize the impact on refrigeration efficiency. As a result, it is most effectively used for preheating outdoor ventilation air at low ambient.

Using commercial refrigeration directly for space cooling and dehumidification may result in efficiency loss for space cooling because the refrigeration system operates at a lower SST than the space-cooling system, resulting in a lower COP for that system. Certain types of cascading refrigeration systems may present opportunities for increasing space cooling and dehumidification energy efficiency, but they require custom designs to capitalize on synergistic efficiency improvements for the entire system.

Heat recovered from the refrigeration system can be used for space heating or outdoor air preheat, as referenced above. The available heat consists of both hot-gas superheat (which can be recovered for service water preheating—see WH6) and condensing heat. Recovery of the condensing heat requires additional controls to avoid compromising refrigeration efficiency and refrigerant liquid management. The heat recovery condenser may be piped in series with, and upstream of, or in parallel with the final outdoor condenser depending on the condensing/reclaim strategy. A higher SCT may be maintained in the heat recovery condenser (using a holdback or inlet pressure-regulating valve on the outlet of the heat recovery condenser), allowing further condensation to occur at the final, lower pressure condenser consistent with the ambient outdoor temperature. A final condenser outlet holdback valve is also required to maintain the minimum SCT for the system. A holdback valve at the heat reclaim heat exchanger may be required to maintain sufficiently high condensing temperature at the heat exchanger when exterior conditions allow the condensing pressure/temperature at the main condenser to approach the minimum setpoint. The holdback valve causes an increased compressor discharge pressure and slightly decreased refrigerant mass flow, resulting in increased energy consumption and slight loss of capacity during some operating conditions. The energy savings benefit of additional heat reclaim during low ambient conditions provided by the holdback valve should be compared with the additional energy consumption generated by the increased compressor discharge pressure during those conditions. Even with the use of a holdback valve, condensing pressure is allowed to float to the minimum setpoint at the final condenser, preserving the increased energy efficiency. In stores with low internal heat gains in cooler climates, the additional compressor energy required to maintain higher SCT at the heat recovery condenser and the additional fan energy needed to overcome the pressure drop across the heat recovery condenser may be relatively small compared with the amount of heat recovered, resulting in a very high COP for provision of this heat. For WSHP systems, heat can be recovered from the refrigeration system to make up a heat deficit in the circulating water loop. The lower temperature at which the heat is recovered to this loop can be controlled to optimize refrigeration system efficiency. Detailed analysis of the annual energy balance of these systems should precede incorporation of this measure into the design of the refrigeration and HVAC systems.

Using multiple condensers to enable heat recovery, either air or water cooled, can result in an increase of refrigerant charge, entailing maintenance and regulatory issues. Closely coupled equipment helps to minimize this problem.

Additional information on heat recovery is provided in RF22 through RF27.

HVAC SYSTEM TYPES

HV4 *Distributed Mixed-Air SZVAV DX Packaged RTU* (Climate Zones: all)

The full description of this system is a distributed mixed-air (MA), single-zone variable-air-volume (SZVAV), direct-expansion (DX) packaged rooftop air-conditioning unit (RTU) with indirect gas furnace or electric resistance heat and hot-gas reheat (HGR) for dehumidification control. Variable-volume systems in grocery stores usually operate as single-zone systems with a single thermostat located in the space.

The unit supply fan either varies the air volume to the space in a linear fashion or changes the speed in discrete fan-speed steps to meet the temperature demand of the space. The cooling capacity should vary to match the fan speed through means of compressor capacity control such as digital scroll technology or variable speed application. All unit size selections should incorporate either multiple compressors and/or multistage compressors to accommodate cooling capacity to airflow matching.

The components of the RTU are factory designed and assembled and include outdoor-air and return-air dampers, filters, fans, a supply-fan VFD, a cooling coil, a heating source, compressors, a condenser, and controls.

The dehumidification necessary to meet the space conditions needed to support the refrigerated display cases (per the case manufacturer's submittal data) requires additional consideration beyond standard unit selection. This may be accomplished using the main cooling coil through selection of a capable sensible heat ratio (SHR); however, a means of reheating the supply air up to acceptable temperatures for space distribution is required. This should be accomplished first through the use of recovered heat if available or by means of a HGR recovery coil from the unit's refrigeration circuit.

Variable-volume packaged units should meet or exceed the efficiency levels listed in Table 5-16 and in the recommendation tables in Chapter 4. The cooling equipment also should meet or exceed the part-load efficiency level, where specified.

Indirect gas-fired furnaces should have at least an 80% efficiency level as required in ASHRAE/IES Standard 90.1 (ASHRAE 2013c). For systems using gas-fired boilers, condensing boilers should be used (see HV15 and HV31).

For packaged SZVAV DX systems, fan power is incorporated into the energy efficiency ratio (EER) calculation. To achieve the required level of energy efficiency, air supply and delivery systems for the packaged SZVAV units should not exceed 0.7 in. w.c. external static pressure (ESP).

A reduced design supply air temperature (SAT) of 52°F may be required to increase the percentage of dehumidification provided by the HVAC system. It can also lower system airflow and the resultant pressure drop when the duct is sized for a typical SAT of 55°F.

Table 5-16 DX Cooling Equipment Efficiency Levels

Size Category	Cooling Efficiency*
<65,000 Btu/h selected at 3 and 5 tons	16.9 SEER
65,000–135,000 Btu/h selected at 10 tons	12.41 EER 14.7 IEER
135,000–240,000 Btu/h selected at 15 tons	12.0 EER 13.0 IEER
135,000–240,000 Btu/h selected at 20 tons	12.0 EER 13.2 IEER
>240,000 Btu/h	Not recommended

* SEER = seasonal energy efficiency ratio, EER = energy efficiency ratio, IEER = integrated energy efficiency ratio.

Units should have air-side economizers in all climate zones except climate zone 1, with control based on dry-bulb temperature sensors with a dew-point temperature limit (typically 55°F). See the recommendation tables in Chapter 4 for the requirements in each climate zone. In climate zones with a large number of hours of economizer operation, units should be selected with consideration of increased fan efficiency.

For SZVAV systems, the minimum supply airflow must comply with local codes and the current editions of ASHRAE Standard 62.1 (for minimum outdoor airflow) and ASHRAE/IES Standard 90.1 (for minimum turndown before reheat is activated) (ASHRAE 2013b, 2013c).

Use the recommendation tables in Chapter 4 to determine the requirements for indirect evaporative precooling or ventilation air heat recovery. Ventilation optimization, a combination of zone demand-controlled ventilation (DCV) and system ventilation reset using the provisions of Standard 62.1, reduces outdoor air minimum flow in occupied mode.

HV5 *Distributed Mixed-Air SZVAV Chilled-Water RTU* (Climate Zones: all)

The full description of this system is a distributed mixed-air (MA) single-zone variable-air-volume (SZVAV), chilled-water packaged rooftop air conditioning unit (RTU) served by an air-cooled chiller and indirect gas furnace or hot-water coil served by a high-efficiency boiler with heat for dehumidification control. Requirements for this system are very similar to those of the packaged SZVAV DX system described in HV4 except that the cooling source, cooling distribution, and cooling-to-air heat exchange are separately specified. Efficiency of the chilled-water equipment and water heating equipment should meet the requirements in HV15. Cooling coils should be selected for a minimum of 15°F ΔT on the water side. Cooling coils should also be selected at no more than 450 ft/min air face velocity to minimize air pressure drop. The chilled-water distribution system should be designed to meet the requirements of HV30. To achieve the required level of energy efficiency for air supply, the diffusers, duct system, return air path, coils, and filters should be selected for minimum pressure drop. Air delivery system pressure drop, fan selection for mechanical efficiency, and motor selection for efficiency should result in no more than 0.72 W/cfm at design airflow. This is derived from

- 3.5 in. of total static pressure,
- 65% fan efficiency,
- 93% motor efficiency, and
- 95% variable-speed drive (VSD) efficiency.

The chilled-water source for this system could be a packaged air-cooled chiller as described in HV15. The dehumidification necessary to meet the space conditions needed to support the refrigerated display cases (per the case manufacturer's submittal data) requires additional consideration beyond standard coil selection. A coil capable of a lower sensible heat ratio (SHR) may be required; however, a means of reheating the supply air to acceptable temperatures for space distribution is required. This should be accomplished through the use of recovered refrigeration heat if available (which could include a "run-around" coil) or by the integral heating apparatus.

Economizer considerations are the same as for HV4.

HV6 *Distributed SZVAV DX Packaged RTU with DOAS* (Climate Zones: all)

The full description for this system is a distributed single-zone variable-air-volume (SZVAV), direct-expansion (DX) packaged rooftop air conditioning unit (RTU) with indirect gas or electric heat integrated with a dedicated outdoor air system (DOAS). In this system, a single packaged DX RTU serves each thermal zone. The unit supply fan either varies the air volume to the space in a linear fashion or changes the speed in discrete fan-speed steps to meet the temperature demand of the space. The cooling capacity should vary to match the fan speed through means of compressor capacity control such as digital scroll technology or variable speed application. All unit size selections should incorporate either multiple compressors and/or multistage compressors to accommodate cooling capacity to airflow matching. The components of the RTU are factory designed and assembled and include outdoor-air and return-air dampers, filters, fans, a cooling coil, a heating source, compressors, a condenser, and controls. A single thermostat controls the unit to maintain the temperature within the space, including modulation of heating or cooling delivered to the space, cycling the unit off when conditioning is not required, and changing between heating and cooling modes.

The packaged unit should meet or exceed the efficiency levels listed in Table 5-16 and in the recommendation tables in Chapter 4. The cooling equipment should also meet or exceed the part-load efficiency level, where specified.

The systems evaluated for this Guide treat recirculated air only, so that the unit fans can be cycled with load without interrupting ventilation air supply. Dehumidified ventilation air is provided continuously by a DOAS. See HV10 for additional information on DOASs. Note that this guide uses *DOAS* and *100% outdoor air system (100% OAS)* interchangeably.

Cooling compressors should have at least four steps of unloading for energy-efficient operation and to avoid coil frosting at low airflow. Unloading can be achieved using variable-speed compressors, digitally unloading compressors, multistage compressors, multiple compressors, or any combination of these.

Indirect gas-fired furnaces should be high efficiency and have at least an 80% efficiency level as required by ASHRAE/IES Standard 90.1 (ASHRAE 2013c).

To achieve the required level of energy efficiency, pressure drops in the air delivery systems for packaged variable-volume units should not exceed 0.7 in. w.c. ESP.

Units should have air-side economizers in all climate zones except climate zone 1, with control based on dry-bulb temperature sensors with an outdoor dew-point limit control (typically 55°F). In climate zones with a large number of hours of economizer operation or heating operation, units should be selected with consideration of increased fan efficiency. See the recommendation tables in Chapter 4 for the requirements in each climate zone.

Use the recommendation tables in Chapter 4 to determine the requirements for indirect evaporative precooling or ventilation air heat recovery. Ventilation optimization, a combination of zone DCV and system ventilation reset using the provisions of ASHRAE Standard 62.1 (ASHRAE 2013b), reduces outdoor air in the occupied mode.

HV7 Distributed SZCV Air-Source Heat Pump Packaged RTU with DOAS (Climate Zones: ❶ ❷ ❸ ❹ ❺)

The full description for this system is a single-zone constant-volume (SZCV) air-source heat pump (HP), packaged rooftop air-conditioning unit (RTU) with electric resistance supplemental heat integrated with a dedicated outdoor air system (DOAS). This system is similar to that in HV6; however, a packaged heat pump unit is used for each thermal zone as opposed to the DX unit described in HV6. Further, the fan systems operate at constant volume. This type of equipment is available in preestablished increments of capacity. The components are factory designed and assembled and may include outdoor-air and return-air dampers, fans, filters, a heating source, a cooling coil, a compressor, controls, and an air-cooled condenser. The heating source is provided by reversing the refrigeration circuit to operate the unit as a heat pump to be supplemented by electric resistance heating if heat pump heating capacity is reduced below required capacity by low exterior air temperatures. Indirect-fired gas furnaces can be used as an alternative heat source with heat pumps but cannot operate to supplement the heat pump output. This alternative was not evaluated for this Guide.

The systems evaluated for this Guide treat recirculated air only, so that the unit fans can be cycled with load without interrupting ventilation air supply. Dehumidified ventilation air is provided continuously by a DOAS. The DOAS may also be a heat pump system (see HV10). Single packaged heat pump units are typically mounted on the roof or at grade level outdoors.

Performance characteristics vary among heat pump manufacturers, and the selected heat pump equipment should match the calculated heating and cooling loads (sensible and latent). The equipment should be listed as being in conformance with electrical and safety standards, with its performance ratings certified by a nationally recognized certification program.

Table 5-17 Constant-Volume Air-Source Heat Pump Efficiency Levels

Size Category	Cooling Efficiency*	Heating Efficiency*
<65,000 Btu/h selected at 3 and 5 tons	13.0 SEER	9.0 HSPF
65,000–135,000 Btu/h selected at 10 tons	11.5 EER 12.5 IEER	3.4 COP at 47°F db/43°F wb outdoor air 2.4 COP at 17°F db/15°F wb outdoor air
135,000–240,000 Btu/h selected at 15 tons	10.6 EER 10.7 IEER	3.2 COP at 47°F db/43°F wb outdoor air 2.1 COP at 17°F db/15°F wb outdoor air
135,000–240,000 Btu/h selected at 15 tons	10.6 EER 10.7 IEER	3.2 COP at 47°F db/43°F wb outdoor air 2.1 COP at 17°F db/15°F wb outdoor air
>240,000 Btu/h	Not recommended	

* SEER = seasonal energy efficiency ratio, EER = energy efficiency ratio, IEER = integrated energy efficiency ratio, HSPF = heating seasonal performance factor.

For packaged constant-volume heat pump systems, the fan energy is included in the calculation of the EER for heat pump equipment, based on standard rating procedures of the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) that include an assumed external air delivery pressure drop (AHRI 2007, 2008). To achieve the required level of energy efficiency, pressure drops in the air delivery systems for packaged constant-volume units should not exceed 0.7 in. w.c. ESP. The packaged unit should meet or exceed the efficiency levels listed in Table 5-17.

Units should have air-side economizers in all climate zones except climate zone 1, with control based on dry-bulb temperature sensors with an outdoor dew-point limit control (typically 55°F). In climate zones with a large number of hours of economizer operation or heating operation, units should be selected with consideration of increased fan efficiency. See the recommendation tables in Chapter 4 for the requirements in each climate zone.

Of critical importance for this type of system is the minimum outdoor temperature at which the unit can provide the required heating capacity to meet the building heating load. Heat pump products are available that are rated to provide as much as 70% of their AHRI-rated capacity (47°F outdoor dry-bulb temperature, 70°F indoor dry-bulb temperature) at –4°F outdoor air temperature. In general, heat pump units selected for an application should be rated to provide some heating at the 99.6% heating design outdoor air temperature for the site, if available. Units meeting the above criteria should also be sized to meet 100% of the building internal heating requirement (not including outdoor air heating) at the 98% heating design outdoor air temperature.

HV8 *SZVAV WSHP Packaged RTU with DOAS* (Climate Zones: all)

The full description for this system is a single-zone variable-air-volume (SZVAV) water-source heat pump (WSHP), packaged rooftop air conditioning unit (RTU) with cooling tower heat rejection and boiler backup heating for the circulating fluid loop and a dedicated outdoor air system (DOAS) for ventilation air. This system type is also similar to that in HV6. A WSHP is used for each thermal zone.

The unit supply fan either varies the air volume to the space in a linear fashion or changes the speed in discrete fan-speed steps to meet the temperature demand of the space. The cooling capacity should vary to match the fan speed through means of compressor capacity control such as digital scroll technology or variable speed application. All unit size selections should incorporate either multiple compressors and/or multistage compressors to accommodate cooling capacity to airflow matching.

This type of equipment is available in preestablished increments of capacity. The components are factory designed and assembled and include a filter, a fan, a refrigerant-to-air heat exchanger, a compressor, a refrigerant-to-water heat exchanger, and controls. The refrigeration cycle is reversible, allowing the same components to provide cooling or heating.

In a traditional WSHP system, all the heat pumps are connected to a common water loop. A fluid cooler such as a closed-circuit cooling tower (see HV9) and a fluid heater such as a hot-

Table 5-18 WSHP Efficiency Levels

Size Category	Cooling Efficiency*	Heating Efficiency*
<65,000 Btu/h selected at 3 and 5 tons	14.2 EER	5.0 COP
65,000–135,000 Btu/h selected at 10 tons	13.9 EER	4.4 COP
135,000–240,000 Btu/h selected at 15 tons	15.0 EER	5.0 COP
135,000–240,000 Btu/h selected at 15 tons	13.6 EER	4.8 COP
>240,000 Btu/h	Not recommended	

*EER = energy efficiency ratio, COP = coefficient of performance

water boiler (see HV15) also are installed in this loop to maintain the temperature of the water within a desired range. The circulation loop should have a variable-speed pump and may include a controller to reset circulating loop temperature according to exterior and operating conditions.

The systems evaluated for this Guide treat recirculated air only, so that the unit fans can be cycled with load without interrupting ventilation air supply. Dehumidified ventilation air is provided continuously by a DOAS and ducted directly to the occupied spaces. Depending on the climate, the DOAS unit may include components to filter, cool, heat, dehumidify, or humidify the outdoor air (see HV10).

The equipment is typically located on the roof or at grade level and should consider the acoustical goals of the space and minimize fan power, ducting, and wiring.

Packaged WSHPs 4 tons and above should incorporate a two-stage or variable-speed compressor with variable-speed fans and a multistage thermostat. The packaged unit should meet or exceed the efficiency levels listed in Table 5-18.

The fan energy is included in the calculation for the EER for WSHP equipment based on standard rating procedures (ASHRAE 2012a) that include an assumed external air delivery pressure drop. Pressure drops in the air delivery system, including ductwork, diffusers, and grilles, should not exceed 0.7 in. w.c.

Per ASHRAE/IES Standard 90.1, the WSHP unit should incorporate a solenoid valve to shut off flow of circulating loop water through the unit when the compressors are de-energized (ASRHAE 2013c). The unit should also cycle fans off when no conditioning is called for.

HVAC EQUIPMENT CONSIDERATIONS

HV9 *Closed-Circuit Cooling Towers for WSHP Systems* (Climate Zones: all)

Closed-circuit cooling towers, or fluid coolers, should be selected for a maximum 10°F approach of cooling tower leaving water temperature to design wet-bulb temperature. Towers also should be selected for fan power at full load of no more than 14.0 gpm/hp. Cooling tower fan motors larger than 1 hp should be equipped with VSDs.

HV10 *Dedicated Outdoor Air Systems (100% Outdoor Air Systems)* (Climate Zones: all)

DOASs can reduce energy use by decoupling the dehumidification and conditioning of ventilation air from sensible cooling and heating in the zone.

The outdoor air is conditioned by a separate DOAS that is designed to dehumidify the outdoor air and to deliver it dry enough (with a low enough dew point) to offset space latent loads, thus providing space humidity control suitable for both occupant comfort and refrigerated display case requirements (Mumma 2001; Morris 2003). The DOAS also can be equipped with high-efficiency filtration systems. DOASs work in conjunction with additional HVAC equipment that heats or cools recirculated air to maintain space temperature, including air-cooled air conditioners, air-source heat pumps, or WSHPs.

There are many possible DOAS configurations (see Figure 5-36 for a few typical ones). Some of the considerations for choosing a DOAS are HVAC system type, the requirement for sensible or enthalpy recovery, the zoning of the system, and the availability of exhaust airstreams in relation to each other. The salient energy-saving features of DOASs are the separa-

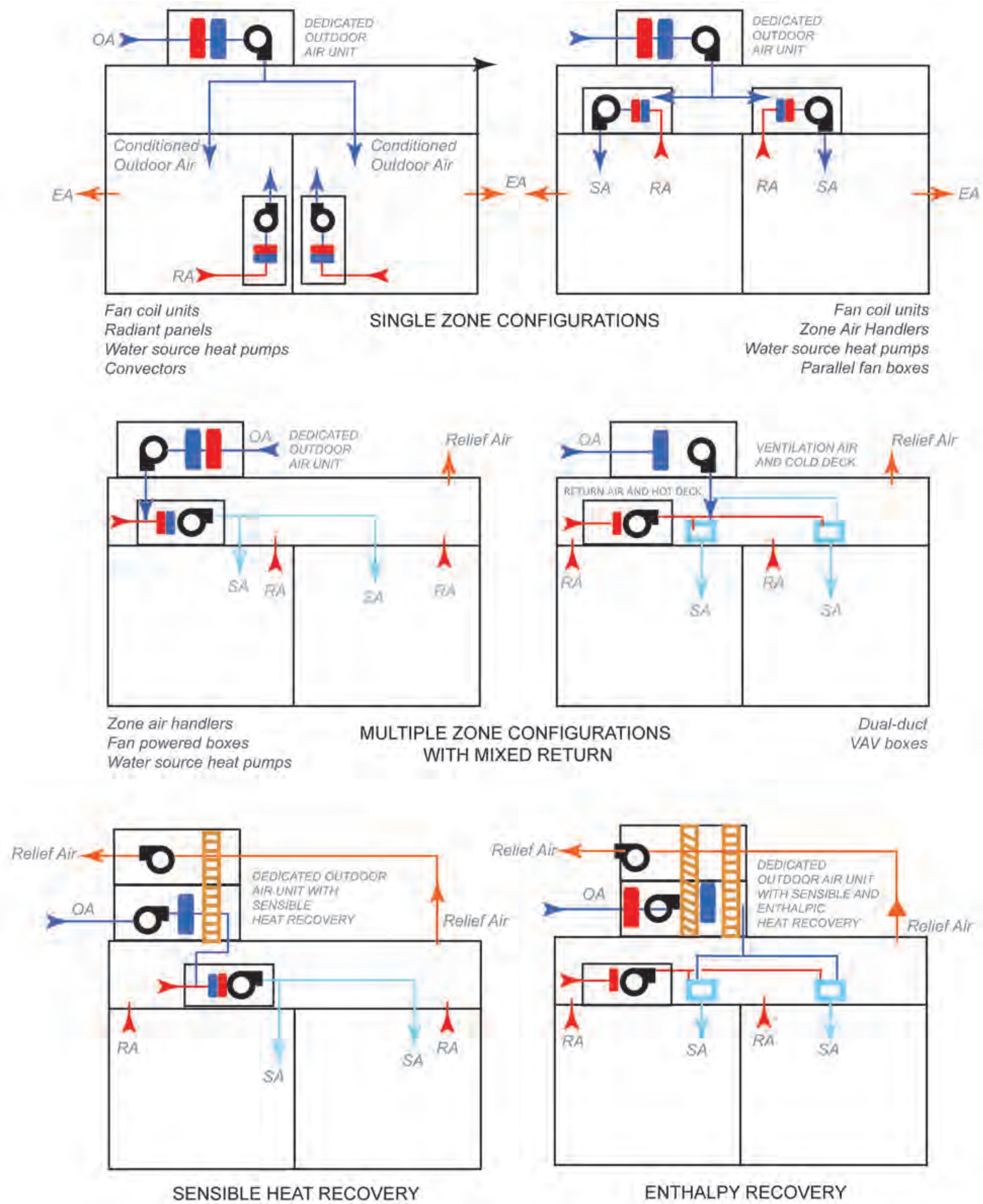

Figure 5-36 (HV10) Examples of DOAS Configurations

tion of ventilation air conditioning from zone air conditioning, the ease of implementation of energy recovery, and the ability to sufficiently ventilate the space.

DOASs are most often used in conjunction with single-zone systems, such as packaged SZVAV, packaged air-source heat pump, and WSHP systems. Multiple-zone VAV systems do not typically have a separate ventilation system, as they are often selected to handle both the building and ventilation loads. However, VAV systems can be designed with DOASs to reduce energy use.

DOASs reduce energy use in primarily three ways: 1) they avoid the high outdoor air intake airflows at central air handlers needed to satisfy the multiple spaces equation of ASHRAE Standard 62.1 (ASHRAE 2013b), 2) they eliminate (or nearly eliminate) the simultaneous cooling and reheat that would otherwise be needed to provide adequate dehumidification in humid climates, and 3) they allow the conditioning unit to cycle with load without interrupting ventilation airflow. A drawback of many DOASs is that they cannot, cost-effectively, directly provide air-side economizing because they are sized for maximum required ventilation airflow. Economizing required by ASHRAE/IES Standard 90.1-2013 should be provided by alternative methods such as water-side economizers within the HVAC system or air-side economizers directly to the space. DOAS systems configured to vary the volume of ventilation airflow to the space should be equipped with a VSD fan. See HV18.

The cooling source for a DOAS can be either an integral DX air- or water-cooled refrigeration system or an external source of chilled water. The external chilled-water source could be a packaged air-cooled chiller as described in HV15.

The supply air temperature, and thus the dew-point temperature, for the DOAS supplying the main sales area should be set sufficiently low that the ventilation airflow can maintain the desired dew-point temperature in the space. In grocery stores, required ventilation airflow per Standard 62.1 (ASHRAE 2013b) is 7.5 cfm/person at the design occupant density of 8 persons per 1000 ft^2 in the sales area, plus additional ventilation of 0.06 cfm/ft^2, giving a combined outdoor airflow at peak occupancy of 15 cfm/person. With that airflow rate and a latent load of approximately 150 Btu/person, the space moisture ratio rise from occupant latent load will be 14.7 grains/lb. With an apparatus dew point of 50°F for the DOAS, the space will be maintained at a dew-point temperature of approximately 56°F, or a level of humidity that should not result in excessive condensation in exposed refrigerated cases. Control of the DOAS leaving temperature, based on space dew-point temperature, needs to consider the maximum allowable dew-point temperature specified by the refrigerated display case manufacturer and can minimize condensation in the cases while avoiding excessive energy consumption for cooling during periods of reduced occupancy.

For stores with full kitchen operation, a separate DOAS supplying required additional makeup air for the kitchen exhaust hoods may be required. The limitations on supply-air conditions for the DOAS serving the sales area impose a significant energy penalty for providing additional air through that system to meet makeup air requirements for kitchen hoods. Should the relief air from the sales air be insufficient to meet the makeup requirements, an additional DOAS should be used. Only in extremely dry climates should the capacity of the sales area DOAS be increased to meet the additional kitchen hood makeup requirement. Air from the kitchen DOAS should be delivered in the vicinity of the exhaust hoods but in a configuration that does not negatively affect the capture of the hoods. Since this air will never be in contact with refrigerated cases, the most energy-efficient strategy for this system is control of the supply air temperature at the widest range consistent with comfort in the kitchen area. Locking out conditioning of the outdoor airstream in the DOAS when the outdoor temperature is between 50°F and 70°F should provide both comfort and energy efficiency. Heat recovery can be implemented for this DOAS in cold climates. (See HV13).

If ventilation air from the DOAS is delivered to the space separately from the conditioning air system, the ventilation air ductwork system should be designed to ensure that adequate ventilation is provided over the entire area served by the ventilation system.

Consider delivering the conditioned outdoor air cold (not reheated to neutral) whenever possible and reheat only when needed. Providing cold (rather than neutral) air from the DOAS offsets a portion of the space sensible cooling loads, allowing the terminal HVAC equipment to be downsized and use less energy (Shank and Mumma 2001; Murphy 2006). Reheating the dehumidified air (to a temperature above the required dew point) may be warranted

- if reheat consumes very little energy (using energy recovery, solar thermal source, etc.) and none of the zones are in cooling mode;
- if all of the zones are in heating mode;
- if, for those zones in cooling mode, the extra cooling energy needed (to offset the loss of cooling due to delivering neutral-temperature ventilation air) is offset by higher-efficiency cooling equipment and the reduction in heating energy needed for those zones in heating mode (this is more likely to be true on an annual basis if reheat in the DOAS is accomplished via air-to-air or condenser heat recovery); or
- if the additional cooling compounds the "cold aisle" condition often experienced in grocery store environments. (Good design features like doors on medium-temperature cases and physical separation of remaining open cases can minimize the cold aisle effect.)

The extent of open refrigerated cases in the store, and the resulting balance-point temperature of the main sales area, will be the primary determinant of the need to reheat the previously cooled supply air from the DOAS. In addition, implementing reset control strategies and exhaust air energy recovery (see HV13) can help minimize energy use.

For stores in humid climates that require a low apparatus dew-point but have low sensible loads in the sales area due to reducing lighting heat gain and open refrigerated casework, various forms of heat recovery may reduce or eliminate energy consumption for reheat. One approach is the heat recovery system entitled *sensible heat recovery* shown in Figure 5-36. This strategy recovers heat from the exhaust air to elevate the temperature of the supply air to the space. While this strategy is very effective for reducing reheat, it still wastes the energy that was required to generate the sensible cooling that was offset by the recovered heat from the exhaust. The configuration shown in Figure 5-36 entitled *enthalpy recovery* avoids most of this wastage through the use of two heat recovery devices. The first device, an enthalpy recovery wheel, transfers heat and moisture from the incoming warm moist outdoor air, upstream of the cooling coil, into the leaving exhaust stream. The second device, a sensible-only wheel, recovers heat from the exhaust upstream of the first wheel to reheat the supply air downstream of the cooling coil. In effect, warming of the supply air, to reduce space overcooling, cools the leaving exhaust air. This cooler exhaust air is more effective, through enthalpy recovery, at precooling the entering outdoor air. In effect, the "coolness" that is removed from the supply air to reduce space overcooling is, to some extent, recovered by this system to further precool the incoming ventilation air. The system requires additional fan power to overcome the pressure drop through the heat recovery devices, but for low-sensible-load stores in humid climates, this additional energy consumption results in significant cooling and reheat energy savings. Other similar heat recovery configurations are available, effectively to convert unusable sensible cooling into beneficial dehumidification.

Packaged heating equipment for DOAS applications should meet or exceed the efficiency levels listed in Table 5-19. Dehumidification efficiency and moisture removal efficiency (MRE) should meet or exceed the efficiency levels listed in Table 5-20. Cooling equipment should also meet or exceed any part-load efficiency requirements, where specified. Boilers providing hot water for hydronic coils in DOAS units should meet the efficiency requirements discussed in HV15. AHRI 920 specifies the performance rating requirements for air-cooled DX DOAS units (AHRI 2012). Gas furnaces should be high efficiency, providing a minimum 80% efficiency.

If available, exhaust-air heat recovery may be necessary for use of air-source heat pump DOASs in locations with low design heating temperatures. Most heat pump units do not operate well with low entering outdoor air temperatures. The use of exhaust heat recovery will temper the outdoor air by recovery heat from the exhaust airstream to raise the incoming ventilation air to a temperature more compatible with heat pump operation.

Table 5-19 DOAS Heating Equipment Efficiencies

Size Category	Heating Efficiency
Gas Heat	
Any size	80%
WSHP	
<65,000 Btu/h selected at 3 and 5 tons	5.0 COP
65,000–135,000 Btu/h selected at 10 tons	4.4 COP
135,000–240,000 Btu/h selected at 15 tons	5.0 COP
135,000–240,000 Btu/h selected at 15 tons	4.8 COP
>240,000 Btu/h	Not recommended

* COP = coefficient of performance

Table 5-20 DOAS Dehumidification and Moisture Removal Efficiency (MRE)

Size Category	Dehumidification Efficiency*	MRE
Air-Cooled DX		
1000 cfm	14.0 EER	7.50 lb/kWh
5000 cfm	12.0 EER	7.00 lb/kWh
10,000 cfm	11.0 EER	7.00 lb/kWh
WSHP DX		
1000 cfm	16.0 EER	8.50 lb/kWh
5000 cfm	14.0 EER	8.00 lb/kWh
10,000 cfm	13.0EER	7.50 lb/kWh

* EER = energy efficiency ratio

Systems delivering 100% outdoor air have many different configurations. The air delivery system should be configured for no more than 0.7 in. w.c. ESP drop for systems with constant ventilation supply and no more than 1.0 in. w.c. ESP drop for systems providing multiple-zone DCV. For units that do not have EER ratings per AHRI, fans should be selected for a minimum 65% mechanical efficiency and motors at no less than 93% efficiency.

HV11 *Evaporative-Cooled Condensers* (Climate Zones: ❷B, ❸B, ❹B, ❺B)

Evaporative-cooled condensers as a part of or added to packaged cooling units can be considered to improve energy efficiency in dry climates. These devices take advantage of the low ambient wet-bulb temperature in order to improve energy efficiency by coupling convective heat rejection with the evaporation of water off of wetted heat rejection condenser coils. In dry climates, up to 40% reduction in cooling energy use can result. As an alternative to true evaporative-cooled condensing, some manufacturers provide a “bolt-on” or field-retrofitted evaporative precooler for a standard air-cooled air conditioner. These devices precool the condenser inlet air, providing some of the benefit from lowered condensing temperatures at a reduced cost and reduced weight and footprint.

Generally speaking, all of the wetted components and the condenser section should be designed for corrosion resistance to ensure reasonable equipment life. Drawbacks to the system include extra first costs, extra weight that arises from the extra equipment and the water in the sump, additional controls, and the need to provide water treatment regimens

HV12 *Dehumidification* (Climate Zones: all)

In grocery stores, for full-load operation systems should be designed with a lower apparatus dew point than systems for other occupancy types. For systems that do not employ DOAS units, consider lowering the SAT for the units from 55°F to 52°F to increase the dehumidification provided by the air-conditioning system. While design to this standard may result in a lower COP for the unit, providing additional dehumidification with the air-conditioning system

Table 5-21 Total System Effectiveness with Energy Recovery

Condition	Effectiveness, %		
	Sensible	Latent	Total
Heating at 100% airflow	78	70	75
Heating at 75% airflow	83	77	82
Cooling at 100% airflow	80	71	75
Cooling at 75% airflow	84	78	82

is likely required to maintain the space dew-point conditions specified by the refrigerated case manufacturer's submittal data and could result in an overall (total building) energy savings. For DX systems, lowering the design SAT may require more precise control of compressor unloading to prevent coil icing during low part-load conditions.

Following are *some* (but *not all*) of the possible methods for providing part-load dehumidification.

- ***For single-zone air conditioner or heat pump packaged units (see HV6 and HV7).*** Packaged RTUs should utilize a DOAS (see HV10) to dehumidify the outdoor air so that it is dry enough (has a low enough dew point) to offset the latent loads in the spaces. This helps avoid high indoor humidity levels without additional dehumidification enhancements in the local air conditioning or heat pump units.
- ***For WSHPs or ground-source heat pumps (GSHPs) (see HV8 and HV37).*** The DOAS (see HV10) should be designed to dehumidify the outdoor air so that it is dry enough (has a low enough dew point) to offset the latent loads in the spaces. This helps avoid high indoor humidity levels without additional dehumidification enhancements in the WSHP units.
- ***For large packaged mixed-air VAV RTUs (see HV4 and HV5).*** VAV systems typically dehumidify effectively over a wide range of indoor loads as long as the VAV RTU continues to provide cool, dry air at part-load conditions. One caveat: use caution when resetting the SAT upward during the cooling season. Warmer supply air means less dehumidification at the coil and higher humidity in the space. If SAT reset is used, include one or more zone humidity sensors to disable the reset if the relative humidity within the space exceeds the space condition limits of the refrigerated case manufacturer's submittal data. Because these units supply both outdoor air and recirculated air, they provide the only dehumidification capability for the HVAC system. Loss of dehumidification capacity at part load due to elevated SAT results in increased condensation in refrigerated cases.

HV13 *Exhaust Air Energy Recovery* (Climate Zones: all)

Exhaust air energy recovery can provide an energy-efficient means of reducing the latent and sensible outdoor-air cooling loads during peak summer conditions. It also can reduce the outdoor-air heating load in mixed and cold climates. HVAC systems that use exhaust air energy recovery should be resized to account for the reduced outdoor-air heating and cooling loads (see ASHRAE 2012b).

Where exhaust air energy recovery is recommended in the Chapter 4 climate-specific tables, the selected device should have a total effectiveness as specified in Table 5-21. Note that in some climates energy recovery is not recommended. Control sequences for exhaust air heat recovery systems should be designed to avoid disadvantageous operation of the system—for example, heat recovery heating of the ventilation air to a temperature higher than the cooling SAT.

The performance levels in Table 5-21 should be achieved with no more than 0.70 in. w.c. static pressure drop through the wheel on the supply side and 0.80 in. w.c. static pressure drop through the wheel and required filtration on the exhaust side. Sensible energy recovery devices transfer only sensible heat. Common examples include coil loops, fixed-plate heat exchangers, heat pipes, and sensible energy rotary heat exchangers (sensible energy wheels). Total energy recovery devices transfer not only sensible heat but also moisture (or latent heat)—that is, energy stored in water vapor in the airstream. Common examples include total energy rotary heat exchangers (also known as *total energy wheels* or *enthalpy wheels*) and fixed-membrane heat exchangers (see Figure 5-37). Energy recovery devices should be selected to avoid cross-contamination of the intake and exhaust airstreams. For rotary heat exchangers, avoidance of cross-contamination typically includes provision of a purge cycle in the wheel rotation and maintenance of the intake system pressure higher than the exhaust system pressure.

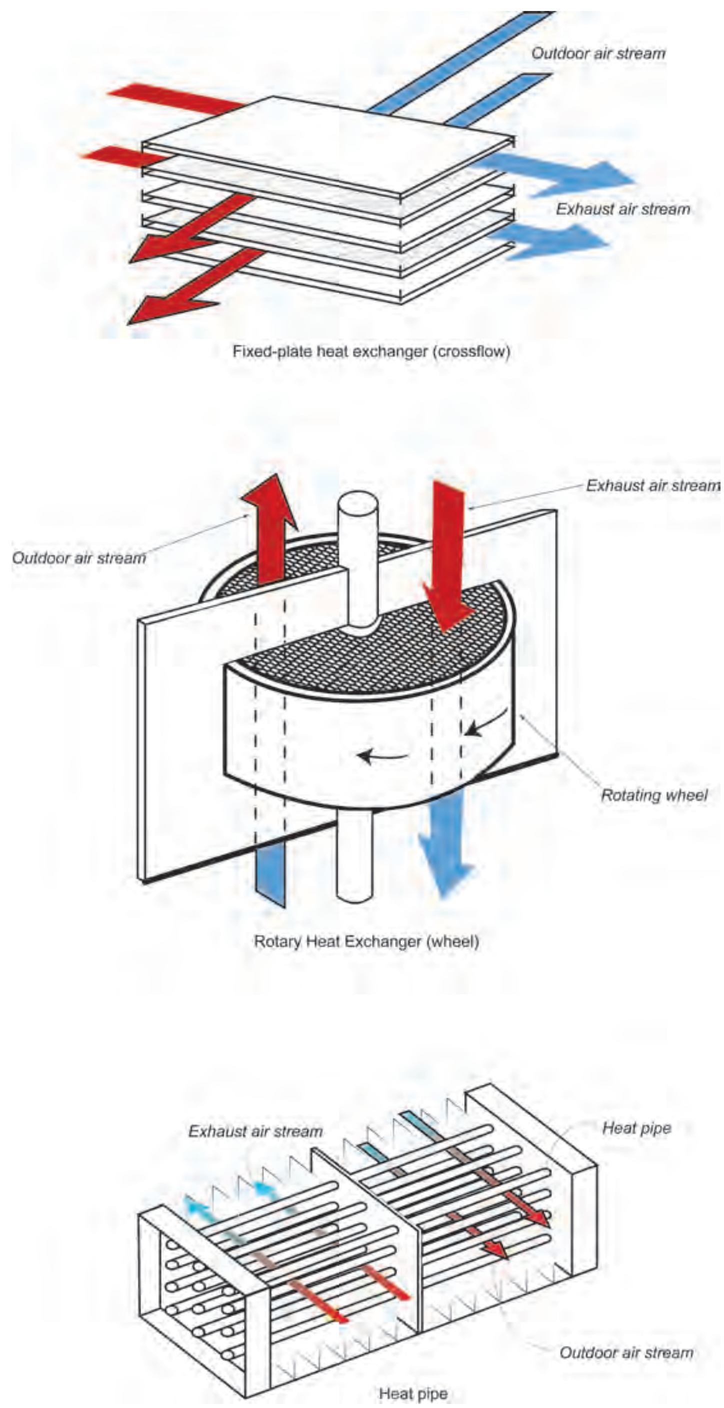

Figure 5-37 (HV13) Examples of Exhaust-Air Energy Recovery Devices

An exhaust-air energy recovery device can be packaged in a separate energy recovery ventilator (ERV) that conditions the outdoor air before it enters the air-conditioning unit, or the device can be integral to the air-conditioning unit.

For maximum benefit, the system should provide as close to balanced outdoor and exhaust airflows as is practical, taking into account the need for building pressurization and any exhaust that cannot be ducted back to the energy recovery device.

Exhaust for energy recovery may be taken from spaces requiring exhaust (using a central exhaust duct system for each unit) or directly from the return airstream (as with a unitary accessory or integrated unit). (See also HV20.)

Where an air-side economizer is used along with an ERV, add bypass dampers (or a separate outdoor air path) to reduce the air-side pressure drop during economizer mode. ERVs should be turned off during economizer mode to avoid transferring any heat between airstreams during mild outdoor conditions. Where energy recovery is used without an air-side economizer, the ERV should still be controlled to prevent unwanted transfer of heat between airstreams.

The air discharge from kitchen grease hoods often reaches 100°F, so heat reclaim from the hood exhaust is likely to be advantageous only in cold climates (climate zones 6, 7, and 8). The grease-laden characteristic of exhaust from cooking presents a technical and regulatory challenge for heat reclaim equipment. The grease entrained in the exhaust flow quickly coats the inside surface of the exhaust system, including heat exchange surfaces, rapidly diminishing their effectiveness.

Additional maintenance is required for peak system performance. Automatic water washdown of heat transfer surfaces and a combination of ozonation with ultraviolet irradiation are techniques that have been employed to reduce grease accumulation on these surfaces. Systems are available from several manufacturers and should be evaluated from the standpoint of energy cost savings, additional maintenance cost, and additional first cost.

In cold climates, follow the manufacturer's recommendations for frost prevention.

HV14 *Indirect Evaporative Cooling* (Climate Zones: ❷B, ❸B, ❹B, ❺B)

In dry climates, incoming ventilation air can be precooled using indirect evaporative cooling. For this strategy, the incoming ventilation air (the primary airstream) is not humidified; instead, a separate stream of air (the secondary or heat rejection stream) is humidified, dropping its temperature, and is used as a heat sink to reduce the temperature of the incoming ventilation air.

The source of the heat rejection stream of air can be either outdoor air or exhaust air from the building. If the air source is exhaust air, this system becomes an alternative for exhaust air energy recovery (HV13).

Sensible heat transfer between the ventilation airstream and the indirect evaporatively cooled airstream can be accomplished using plate or tubular air-to-air heat exchangers, heat pipes, or a pumped fluid loop between air coils in each stream (often called a *runaround loop*). For indirect evaporative coolers that use exhaust air as the secondary stream, the evaporative cooler also can provide sensible heat recovery during the heating season. If a runaround loop is used for heat transfer both for indirect evaporative cooling and heat recovery, the circulating fluid should incorporate antifreeze levels appropriate for the location's design heating temperature.

Indirect evaporative cooling has the advantage that humidity is not added to the interior environment and that indoor air quality (IAQ) is not impacted by microbial growth on the evaporative media, antimicrobial treatment of the evaporative water, or dusting of dissolved solids in the evaporative water. Air quality for the exhausted secondary airstream is not held to the requirements of ASHRAE Standard 62.1 (ASHRAE 2013b) like the ventilation airstream entering the occupied space is.

Indirect evaporative coolers should be selected for at least 90% evaporative effectiveness for the evaporatively cooled airstream and for at least 65% heat transfer efficiency between the two airstreams.

Indirect evaporative coolers also should be selected to minimize air pressure drop through the heat exchangers. For example, a heat pipe indirect evaporative cooler using exhaust air for the heat rejection stream should have a pressure drop through the supply side, not including filtration, of no more than 0.5 in. w.c. and no more than 0.85 in. w.c. for the exhaust sprayed coil. Inclusion of VSD fans will reduce fan energy by allowing reduction of airflow during nonpeak cooling conditions.

HV15 *Chilled Water and Hot Water Equipment Efficiencies* (Climate Zones: all)

The cooling and heating equipment should meet or exceed the efficiency levels listed in the climate-specific recommendation tables in Chapter 4. The cooling equipment should also meet or exceed the part-load efficiency level where specified. In some cases, recommended equipment efficiencies are based on system size (capacity).

There are many factors involved in making the decision whether to use gas or electricity, such as the availability of service, utility costs, operator familiarity, and the impact of source energy use. Efficiency recommendations for both types of equipment are provided in the recommendation tables in Chapter 4 to allow the user to choose.

Air-cooled chillers. Air-cooled water chillers should be certified or independently tested to produce a full-load EER of 10.1 or higher and an integrated part-load value (IPLV) of 13.7 or higher for selections less than 150 tons. For chiller selections equal to or greater than 150 tons, the chiller should provide an EER of 10.1 and an IPLV of 14.0. All ratings should be in accordance with AHRI rating methods (AHRI 2011). Chillers less than 40 tons should provide at least two steps of unloading, while those 40 tons and above should provide at least four steps of unloading or continuous unloading. Chillers should incorporate controls capable of accommodating variable evaporator water flow while maintaining control of leaving chilled-water temperature. Water-cooled chillers and cooling towers were not analyzed for this Guide. A system including a water-cooled chiller, condenser water pump, and cooling tower all with sufficient efficiency and integrated controls may give the same or better energy performance as an air-cooled chiller.

Space-heating water boilers. All gas-fired boilers specified for space heating should be of the condensing type with a minimum efficiency of 90% at 125°F return hot-water temperature. Zone heat transfer equipment should be sized based on 140°F entering hot-water temperature and as large a temperature drop through the air heating coil as possible. Boilers should be operated with a maximum leaving hot-water temperature of 140°F and should incorporate a leaving hot-water temperature reset control based on total heating load. (See HV31.)

HV16 *Ventilation Air* (Climate Zones: all)

The zone-level outdoor airflows and the system-level intake airflow should be determined based on the most recent edition of ASHRAE Standard 62.1 but should not be less than the values required by local code unless approved by the authority having jurisdiction. The number of people used in computing the breathing zone ventilation rates should be based on known occupancy, local code, or the default values listed in Standard 62.1.

> ***Cautions*:** The occupant load, or exit population, used for egress design to comply with the fire code is typically much higher than the zone population used for ventilation system design. Using occupant load rather than zone population to calculate ventilation requirements can result in significant overventilation, oversized HVAC equipment, and excess energy use.

Buildings with multiple-zone recirculating ventilation systems can be designed to account for recirculated outdoor air as well as system population diversity using the Ventilation Rate Procedure of Standard 62.1 (ASHRAE 2013b). In effect, the multiple-zone recirculating ventilation system design approach allows ventilation air to be calculated on the basis of how many people are *in the building* (system population at design) rather than the sum of how many people are *in each space* (sum of peak zone population at design). Using the Ventilation Rate Procedure can reduce the energy required to condition ventilation air in grocery stores. Refer to *62.1-2010 User's Manual* for specific guidance (ASHRAE 2010a).

An alternative to the Ventilation Rate Procedure is the Indoor Air Quality Procedure, which provides for performance-based design of ventilation systems based on actual contaminant generation in the space. See ASHRAE Standard 62.1-2013, Section 6.3, for a description of this method. This method optimizes the amount of ventilation air to meet threshold contaminant values based on calculations or tests of similar spaces to meet established threshold contaminant values (ASHRE 2013b). For facilities with very low product off-gassing levels, this procedure may result in lower outdoor air ventilation rates than the Ventilation Rate Procedure. Ventilation rates determined by product off-gassing of contaminants typically will be constant compared to variable rates determined by occupancy in the Ventilation Rate Procedure. Designers should evaluate the energy consumption consequences of these two methods of determining ventilation rates.

For all zones, time-of-day schedules in the BAS should be used to introduce ventilation air only when a zone is expected to be occupied.

For packaged single-zone air-source air conditioners and heat pumps, WSHPs, or GSHPs (see HV4, HV6, HV7, and HV8). The DOAS (see HV10) should deliver the conditioned outdoor air directly to the occupied zone if possible. Where DOAS conditioned air must be delivered to the space through the air-conditioning units, those units should be configured such that ventilation air can be delivered to the space even when the unit fan is not running, allowing the unit to cycle for load without interrupting ventilation supply.

For mixed-air packaged VAV RTUs (see HV3). Each rooftop unit should have an outdoor air intake through which outdoor air is introduced and mixes with the recirculated air prior to being conditioned and delivered to the zones.

HV17 *Economizers* (Climate Zones: 2 3 4 5 6 7 8)

Economizers, when recommended, help save energy by providing free cooling when ambient conditions are suitable to meet all or part of the cooling load. In humid climates, use enthalpy based controls (versus dry-bulb temperature based controls) with adjustable dew-point high-limit setpoint to ensure that unwanted moisture is not introduced into the space. See the climate-specific recommendation tables in Chapter 4 for economizer recommendations by climate zone.

Nondedicated outdoor air systems should be capable of modulating the outdoor air, return air, and relief air dampers to provide up to 100% of the design supply air quantity as outdoor air for cooling. (See HV10 for a discussion of DOASs.) A motorized outdoor air damper should be used instead of a gravity damper to prevent unwanted outdoor air from entering during unoccupied periods when the unit may recirculate air to maintain setback or setup temperatures. For all climate zones, the motorized outdoor air damper should be closed during the entire unoccupied period, except when it may open in conjunction with unoccupied economizer cycle operation or a preoccupancy purge cycle.

Periodic maintenance (at least annually), including recalibration of sensors, is important with economizers, as dysfunctional economizers can cause substantial excess energy use because of malfunctioning dampers or sensors (see HV36).

HV18 *Demand-Controlled Ventilation (DCV)* (Climate Zones: all)

DCV can reduce the energy required to condition outdoor air for ventilation. To maintain acceptable IAQ, the setpoints (limits) and control sequence must comply with ASHRAE Standard 62.1 (ASHRAE 2013b). Refer to Appendix A of *62.1-2010 User's Manual* for specific guidance (ASHRAE 2010a). For grocery stores, DCV likely is only effective for areas of the building that would experience occupant density varying from very dense to unoccupied.

DCV controls should vary the amount of outdoor air in response to the need in a zone. The amount of outdoor air could be controlled by 1) a time-of-day schedule in the BAS, 2) an occupancy sensor (such as a motion detector) that indicates when a zone is occupied or unoccupied, or 3) a CO_2 sensor as a proxy for ventilation airflow per person that measures the change in CO_2 levels in a zone relative to the levels in the outdoor air. A controller will then operate the outdoor air, return air, and relief air dampers to maintain proper ventilation. For options 1 and 2, ventilation rates for the occupied period should be based on full occupancy and should be calculated in accordance with Section 6 of Standard 62.1. For option 3, the full-load ventilation rate should be calculated according to Section 6 and ventilation rate reductions should be controlled according to Informative Appendix C of Standard 62.1 (ASHRAE 2013b).

CO_2 sensors should be used in zones that are densely occupied and have highly variable occupancy patterns during the occupied period, such as conference rooms or meeting areas. For the other zones, occupancy sensors should be used to reduce ventilation when a zone is temporarily unoccupied. For all zones, time-of-day schedules in the BAS should be used to introduce ventilation air only when a zone is expected to be occupied.

Employing DCV in a DOAS requires an automatic damper, a CO_2 sensor, and an airflow measurement device for each DCV zone. If the automatic damper selected is of the pressure-independent type, it has flow measurement capability by definition.

Control of a DCV system to match airflow volume to occupancy requires a continuous search algorithm for the controller. The controller continuously calculates the volume required in the space based on the measured CO_2 concentration differential and then updates the volume setpoint to the pressure-independent damper controller. The following equation gives the required volume in the space based on the measured CO_2 differential:

$$V'_{ot} = \frac{R_a \times A_z}{(E_z - (R_p \times (C_R - C_{OA}) / (8400 \times m)))}$$

where

V'_{ot} = required airflow volume at any point in time

R_a = zone area ventilation rate (0.06 cfm/ft^2 for sales areas in Standard 62.1-2013)

A_z = area of the zone

E_z = air distribution effectiveness of the zone

R_p = zone people ventilation rate (7.5 cfm/person for sales areas in Standard 62.1-2013)

C_R = measured CO_2 concentration at the room

C_{OA} = measured CO_2 concentration of the outdoor air

m = metabolic level for occupants of the space (m = 1.6 for shopping or light activity) (Standard 62.1-2013, Informative Appendix C)

The above equation was extracted from *62.1-2010 User's Manual* (ASHRAE 2010a). The control system for the DOAS must be able to calculate the required ventilation rate for each zone based on the above equation then reset the flow setpoint for the zone to the calculated value.

Inaccurate CO_2 sensors can cause excessive energy use or poor IAQ, so they need to be calibrated as recommended by the manufacturer (see HV19).

Finally, when DCV is used, the system controls should prevent negative building pressure. If the amount of air exhausted remains constant while the intake airflow decreases, the building may be under a negative pressure relative to the outdoors. When air is exhausted directly from the zone (e.g., from kitchen exhaust hoods, restrooms, or janitorial closets), the DCV control strategy must avoid reducing intake airflow below the amount required to replace the air being exhausted.

HV19 *Carbon Dioxide (CO_2) Sensors* (Climate Zones: all)

The number and locations of CO_2 sensors for population-based DCV can affect the ability of the system to accurately determine the building or zone occupancy. A minimum of one CO_2 sensor per zone is recommended for systems with greater than 500 cfm of outdoor air. Multiple sensors may be necessary if the ventilation system serves spaces with significantly different occupancy expectations. At a minimum, sensors should be located in any space that is expected to have the greatest occupant density at any point during the operating schedule. Where multiple sensors are used, the ventilation should be based on the sensor recording the highest concentration of CO_2.

The number and location of sensors should take into account the sensor manufacturers' recommendations for their particular products as well as the projected usages of the spaces. Sensors should be located such that they provide a representative sampling of the air within the occupied zone of the space. For example, locating a CO_2 sensor directly in the flow path from an air diffuser would provide a misleading reading concerning actual CO_2 levels (and corresponding ventilation rates) experienced by the occupants. To capture the entire impact of occupants on the room airflow, the best location for a CO_2 sensor is as near as possible to a return grille.

The outdoor air CO_2 concentration can fluctuate significantly in urban areas. Outdoor air CO_2 concentration should be monitored using a CO_2 sensor located near the position of the outdoor air intake. CO_2 sensors should be certified by the manufacturer to have an accuracy to within ±50 ppm (factory calibrated). CO_2 sensors should be calibrated on a regular basis per the manufacturer's recommendations or every six months (per ASHRAE Standard 62.1 [ASHRAE 2013b]).

HV20 *Exhaust Air Systems* (Climate Zones: all)

Zone exhaust airflows (for restrooms, janitorial closets, and break rooms) should be determined based on the current edition of ASHRAE Standard 62.1 but should not be less than the values required by local code unless approved by the authority having jurisdiction.

Central exhaust systems for restrooms, janitorial closets, and break rooms should cycle off during unoccupied periods. Such a system should have a motorized damper that opens and closes with the operation of the fan. The damper should be located as close as possible to the duct penetration of the building envelope to minimize conductive heat transfer through the duct wall and avoid having to insulate the entire duct. During unoccupied periods, the damper should remain closed and the exhaust fan turned off, even if the air-conditioning system is operating to maintain setback or setup temperatures. Consider designing exhaust ductwork to facilitate recovery of energy (see HV13) from exhaust taken from spaces with air quality classification of 1 or 2 (e.g., restrooms) per Table 6.1 of Standard 62.1 (ASHRAE 2013b).

Exhaust systems should be designed to minimize energy consumption because of their continuous operation. Ductwork should be designed for low pressure drop per HV18. Exhaust or relief fans with a brake horsepower rating greater than 5 bhp should be selected with a fan efficiency grade (FEG) of 67 or better and selected within 10 points of peak efficiency per AMCA Standard 205-12 (AMCA 2012).

HV21 *Ductwork Design and Construction* (Climate Zones: all)

Low-energy-use ductwork design involves short, direct, and low-pressure-drop runs. The number of fittings should be minimized and should be designed with the least amount of turbulence produced. (In general, the first cost of a duct fitting is approximately the same as 12 ft of straight duct that is the same size as the upstream segment.) Excessive noise from ductwork airflow is a direct result of air turbulence. Round duct is preferred over rectangular duct. However, space (height) restrictions may require flat oval ductwork to achieve the low-turbulence qualities of round ductwork. Alternatively, two parallel round ducts may be used to supply the required airflow.

Air should be ducted through low-pressure ductwork with a system pressure classification of less than 2 in. w.c. Rigid ductwork is necessary to maintain low pressure loss and reduce fan energy. When a unit is serving multiple zones, supply air should be ducted to diffusers in each individual space.

In general, the following sizing criteria should be used for duct system components:

- Diffusers and registers, including balancing dampers, should be sized with a static pressure drop not to exceed 0.08 in. w.c. Oversized ductwork increases installed cost but reduces energy use due to lower pressure drop.
- Supply ductwork should be sized with a pressure drop no greater than 0.08 in. w.c. per 100 lf.
- Return ductwork should be sized with a pressure drop no greater than 0.04 in. w.c. per 100 lf.
- Exhaust ductwork should be sized with a pressure drop no greater than 0.05 in. w.c. per 100 lf.
- Flexible ductwork should be of the insulated type and should be
    - limited to connections between duct branches and diffusers or between duct branches,
    - limited to 5 ft (fully stretched length) or less,
    - installed without any kinks,
    - installed with a durable elbow support when used as an elbow, and
    - installed with no more than 15% compression from fully stretched length.

Hanging straps, if used, need to use a saddle to avoid crimping the inside cross-sectional area. For ducts 12 in. or smaller in diameter, use a 3 in. saddle; those larger than 12 in. should use a 5 in. saddle.

Long-radius elbows and 45° lateral take-offs should be used wherever possible. The angle of a reduction transition should be no more than 45° (if one side is used) or 22.5° (if two sides are used). The angle of expansion transitions should be no more than 15° (laminar air expands approximately 7°).

Air diffusers should be selected for the prevalent operational mode they will perform. Diffusers serving areas with open refrigerated cases will likely be operating in neutral or heating mode most of the time, so they should be selected to throw downward to drive neutral or warmer air toward the floor. Diffusers in predominantly cooled areas should have a more lateral throw to enhance mixing and minimize cold downdrafts. Caution should be exercised when using directional diffusers near open refrigerated cases, as even terminal velocities can disrupt the air curtains within the cases, creating case performance issues. Diffusers for VAV systems should be selected to avoid "dumping" when the AHU or zone terminal is at its lowest volume setting.

If a ducted return system is used, a single appropriately sized return inlet in a central location will minimize first cost, but the system should be designed to minimize both noise and excessive local airflow. For larger systems, several return air inlets near the air-conditioning unit may be required. Open plenum return systems are less expensive but must be carefully designed and constructed to minimize infiltration of humid air from the outdoors (Harriman et al. 2001).

Return air for HVAC units should be taken from low in the conditioned space in areas that require heating, if possible, to reduce stratification and to improve comfort. For systems recovering heat from refrigeration for space heating, low return is particularly important, to enable heating to be performed at the lowest temperature possible. Lower temperatures for heat recovery reduce the resulting refrigeration efficiency penalty. (See RF26.)

Infiltration in the ceiling plenum can be reduced by using a relief fan to maintain plenum pressure at about 0.05 in. w.c. higher than atmospheric pressure (see HV32). Lowering indoor humidity levels as well can reduce the risk of condensation (see HV12). In addition, exhaust duct systems should be properly sealed to minimize infiltration.

Caution: Ductwork should not be installed outside the building envelope. Ductwork connected to RTUs should enter or leave the unit through an insulated roof curb around the perimeter of the unit's footprint. Flexible duct connectors should be used to prevent sound and vibration transmission between the RTU and the building ductwork.

Duct board should be airtight (Seal Class A, from ASHRAE/IES Standard 90.1 [ASHRAE 2013c]) and should be taped and sealed with products that maintain adhesion (such as mastic or foil-based tape). Duct static pressures should be designed and equipment and diffusers should be selected not to exceed noise criteria for the space (see HV34 for additional information on noise control).

HV22 *Duct Insulation* (Climate Zones: all)

The following ductwork should be insulated:

- All supply air ductwork
- All return air ductwork located in an unconditioned space, including any space outside the insulated building envelope
- All exhaust and relief air ductwork between the motor-operated damper and penetration of the building exterior

In addition, all airstream surfaces should be resistant to mold growth and resist erosion, according to the requirements of ASHRAE Standard 62.1 (ASHRAE 2013b).

Exception: In conditioned spaces without a finished ceiling, only the supply air main ducts and major branches should be insulated. Individual branches and run-outs to diffusers in the space being served do not need to be insulated, except where it may be necessary to prevent condensation.

HV23 *Duct Sealing and Leakage Testing* (Climate Zones: all)

The ductwork should be sealed for Seal Class A from ASHRAE/IES Standard 90.1 (ASHRAE 2013c). All duct joints should be inspected to ensure they are properly sealed and insulated, and the ductwork should be leak-tested at the rated pressure. The leakage should not exceed the allowable cubic feet per minute per 100 ft^2 of duct area for the seal and leakage class of the system's air quantity apportioned to each section tested. See HV18 for guidance on ensuring the air system's performance.

HV24 *Fan Motor Efficiencies* (Climate Zones: all)

Motors for fans should meet NEMA premium efficiency motor guidelines (NEMA 2011) when available. EC motors may be an appropriate choice for many small units to increase efficiency.

Fan systems should meet or exceed the efficiency levels listed in this chapter and in the recommendation tables in Chapter 4. Depending on the HVAC system type, the efficiency level is expressed in terms of either a maximum power (W) per cubic foot per minute of supply air (for systems where fan power is not included in the packaged HVAC unit efficiency calculation) or a maximum ESP loss (for packaged systems where fan power is included in the EER calculation).

HV25 *Thermal Zoning* (Climate Zones: all)

Grocery stores may have a limited number of zones, with the general shopping space arbitrarily divided into zones based on the HVAC distribution scheme. The various spaces of a store, even though not divided by walls, can have very different heat gains and losses, resulting in different temperature profiles. The temperature in aisles with open refrigerated cases will likely be much lower than that of dry-good aisles. Special-purpose areas, such as offices, work rooms, or special sales areas with significantly different lighting, equipment loads, or occupant density than the general sales area, should be controlled as separate zones.

Zoning is typically accomplished with multiple HVAC units. The temperature sensor for each zone should be installed in a location that is representative of the entire zone.

Avoid using a single air handler (or RTU) to serve zones that have significantly different occupancy patterns. Using multiple air handlers allows air handlers serving unused areas of the building to be shut off, even when another area of the building is still in use. An alternative approach is to use a BAS to define separate operating schedules for these areas of the building, shutting off airflow to the unused areas while continuing to provide comfort and ventilation to areas of the building that are still in use. Of particular concern are areas of the building, such as receiving spaces, that might be open to the outside for a significant period of time. If these areas are conditioned, they should have separate HVAC units from the sales area to avoid introduction of humid air into the returns of the sales area HVAC units.

The number of spaces in a zone and the locations of the temperature sensors (thermostats) will affect the control of temperature in the various spaces of a zone. Locating the thermostat in one room of a zone with multiple spaces provides feedback based on only the conditions in that room. Placing spaces with refrigerated cases on the same thermal control zone as spaces without cases likely will increase energy consumption and degrade comfort in both spaces.

To prevent misreading of the space temperature, zone thermostats should not be mounted on exterior walls. Where this is unavoidable, use an insulated sub-base for the thermostat. Avoid locating thermostats near and especially above internal heat sources, adjacent to refrigerated cases, in locations where they can be influenced by direct sunlight, or in any other location where the conditions in the local interior microclimate are not representative of conditions in the overall space.

Proper zoning not only aids in energy efficiency but also can help increase occupant comfort in the space. The six primary factors that must be addressed when defining conditions for thermal comfort are as follows:

- Metabolic rate
- Clothing insulation
- Air temperature
- Radiant temperature
- Air speed
- Humidity

HV26 *System-Level Control Strategies* (Climate Zones: all)

Control strategies can be designed to help reduce energy. Having a setback temperature for unoccupied periods during the heating season or a setup temperature during the cooling season can help save energy by avoiding the need to operate heating, cooling, and ventilation equipment. Determination of unoccupied period conditions in a grocery store, however, must address the issue of humidity control. Temperature setup during unoccupied periods likely is not applicable to spaces with refrigerated cases. While programmable thermostats allow each zone to vary the temperature setpoint based on time of day and day of the week, they also allow occupants to override these setpoints or ignore the schedule altogether (by using the “hold” feature), reducing the potential for energy savings. A more sustainable approach is to equip each zone with a zone temperature sensor and then use a system-level controller that coordinates the operation of all components of the system. This system-level controller contains time-of-day schedules that define when different areas of the building are expected to be unoccupied. During these times, the system is shut off and the temperature is allowed to drift away from the occupied setpoint.

A preoccupancy ventilation period can help purge the building of contaminants that build up overnight from the off-gassing of products and packaging materials, if the ventilation strategy has not previously been determined by the Indoor Air Quality Procedure of ASHRAE Standard 62.1 (ASHRAE 2013b). Cool temperatures at night also can help precool the building. In humid climates, however, care should be taken to avoid bringing in humid outdoor air during unoccupied periods that could adversely affect the refrigerated cases and walk-ins.

Buildings with multiple-zone recirculating ventilation systems can be designed to account for recirculated outdoor air as well as system population diversity using the equations found in Normative Appendix A of Standard 62.1. (See HV18.)

Optimal start uses a system-level controller to determine the length of time required to bring each zone from the current temperature to the occupied setpoint temperature. Then, the controller waits as long as possible before starting the system so that the temperature in each zone reaches the occupied setpoint just in time for occupancy. This strategy reduces the number of hours that the system needs to operate and saves energy by avoiding the need to maintain the indoor temperature at the occupied setpoint when the building is unoccupied.

SAT reset is of limited use in grocery stores because of the need to minimize the humidity level in the space to minimize refrigerated case defrost. In dry climates, SAT for both DOAS units and mixed-air air-conditioning units may be reset upward to minimize compressor energy when space sensible loads permit.

Control systems should include the following:

- Control sequences that easily can be understood and commissioned
- A user interface that facilitates understanding and editing of building operating parameters and schedules
- Sensors that are appropriately selected for range of sensitivity and ease of calibration
- Means to effectively convey the current status of systems operation and of exceptional conditions (faults)
- Means to record and convey history of operations, conditions, and efficiencies
- Means to facilitate diagnosis of equipment and systems failures
- Means to document preventive maintenance

HV27 *Testing, Adjusting, and Balancing* (Climate Zones: all)

After the system has been installed, cleaned, and placed in operation, it should be tested, adjusted, and balanced in accordance with ANSI/ASHRAE Standard 111 (ASHRAE 2008) or the Sheet Metal and Air Conditioning Contractor’s National Association (SMACNA) testing, adjusting, and balancing manual (SMACNA 2002).

Testing, adjusting, and balancing will help to ensure that correctly sized diffusers, registers, and grilles have been installed, that each space receives the required airflow, and that the fans meet the intended performance. The balancing subcontractor should certify that the instruments used in the measurement have been calibrated within 12 months before use. A written report should be submitted for inclusion in the operation and maintenance (O&M) manuals.

HV28 *Commissioning (Cx)* (Climate Zones: all)

After the system has been installed, cleaned, and placed in operation, it should be commissioned to ensure that the equipment meets the intended performance and that the controls operate as intended. See QA1–QA17 for more information on commissioning.

HV29 *Filters* (Climate Zones: all)

Particulate air filters are typically included as part of factory-assembled HVAC equipment and should have at least a Minimum Efficiency Reporting Value (MERV) of 6, based on testing procedures described in ASHRAE Standard 52.2 (ASHRAE 2012c).

As explained in *Indoor Air Quality Guide: Best Practices for Design, Construction and Commissioning* (ASHRAE 2009a), the U.S. Environmental Protection Agency (EPA) maps areas not in compliance (nonattainment) with the National Ambient Air Quality Standards (NAAQS) (EPA 2014, 2015). PM2.5 particles are those smaller than 2.5 µm in diameter. In PM2.5 nonattainment areas (virtually all major metropolitan areas), use MERV 11 filters for outdoor air. Use a filter differential pressure gauge to monitor the pressure drop across the filters and send an alarm if the predetermined pressure drop is exceeded.

If high-efficiency filters are to be used, consider using lower-efficiency filters during the construction period. When construction is complete, all filters should be replaced before the building is occupied.

Note that air filters, especially when loaded, account for a significant amount of pressure drop in the air distribution system. Increasing filter size to reduce face velocity will reduce pressure drop across the filter.

HV30 *Chilled-Water System* (Climate Zones: all)

Chilled-water systems efficiently transport cooling energy throughout the building. Often they are combined with thermal storage systems to achieve electrical demand charge savings through mitigation of the peak cooling loads in the building. Thermal storage systems are not covered in this Guide. Chilled-water systems should generally be designed for variable flow through the building.

Small systems (<100 tons) should be designed for variable flow if the chiller unit controls can tolerate expected flow rate changes. Chilled-water systems should use two-way valves with a pressure-controlling bypass set to maintain the minimum evaporator water flow required by the chiller. For chillers that do not tolerate variable chilled-water flow, three-way valves should be used.

Piping should be sized using the tables in ASHRAE/IES Standard 90.1 (ASHRAE 2013c).

Select cooling coils for a design chilled-water ΔT of at least 15°F to reduce pump energy. Select cooling coils to minimize air pressure drop. Chilled-water temperature setpoints should be selected based on a life-cycle analysis of pump energy, fan energy, and desired air conditions leaving the coil. Use the recommended temperatures listed in the climate-specific recommendation tables in Chapter 4.

HV31 *Hydronic Heating Systems* (Climate Zones: all)

Condensing boilers can operate at up to 97% efficiency and can operate efficiently at part load. To achieve these high efficiency levels, condensing boilers require that return water temperatures be maintained between 70°F and 120°F, where the boiler efficiency ranges from 97% to 91%. This fits well with hydronic systems that are designed with ΔTs greater than 20°F (the optimal ΔT is 30°F to 40°F). The higher ΔTs allow smaller piping and less pumping energy. Because condensing boilers work efficiently at part load, VFDs can be used on the pumps to further reduce energy use.

Condensing boiler capacity can be modulated to avoid losses caused by cycling at less than full load. This encourages the installation of a modular (or cascade) boiler system, which allows several small units to be installed for the design load but allows the units to match the load for maximum efficiency of the system.

HV32 *Relief versus Return Fans* (Climate Zones: all)

Most grocery stores are single level with either plenum or minimally ducted return pathways and do not require return air fans to avoid building overpressurization. If the designer fears that overpressurization may be a problem because of a more complicated return air pathway, in conjunction with an air-side economizer, relief fans should be used in preference to return fans. Relief fans reduce overall fan energy compared to return fans, as long as return dampers are sized correctly. Return air fans would only be required if return duct static pressure drop exceeds 0.5 in. w.c.

Cautions

HV33 *Heating Sources* (Climate Zones: all)

Many factors come into play in deciding whether to use gas or electricity for heating, including availability of service, utility costs, operator familiarity, and the impact of source energy use.

Forced-air electric resistance and gas-fired heaters require a minimum airflow rate to operate safely. These systems, whether stand-alone or incorporated into an air-conditioning or heat pump unit, should include factory-installed controls to shut down the heater when there is inadequate airflow that can result in high temperatures.

Ducts and supply air diffusers should be selected based on discharge air temperatures and airflow rates.

HV34 *Noise Control* (Climate Zones: all)

Acoustical requirements may necessitate attenuation of the supply and/or return air, but the impact on fan energy consumption also should be considered and, if possible, compensated for in other duct or fan components. Acoustical concerns may be particularly critical in short, direct runs of ductwork between the fan and supply or return outlet. When equipment is located near spaces with noise level criteria, special care must be taken to address all equipment noise sources and sound paths into the space. Typical paths of concern for rooftop equipment located above a space are shown in Figure 5-38; sources and paths for equipment located nearby or adjacent to a space are shown in Figure 5-39.

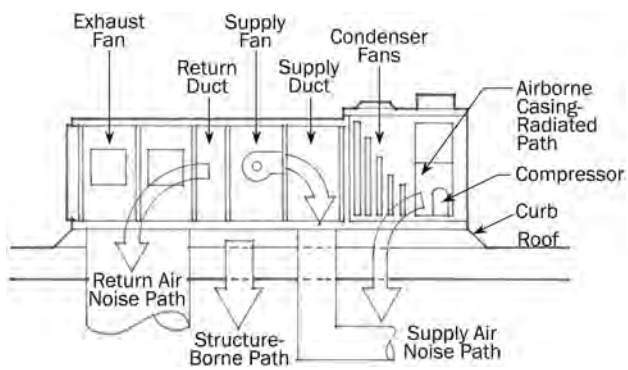

Figure 5-38 (HV34) Typical Noise Paths for Rooftop-Mounted HVAC Units

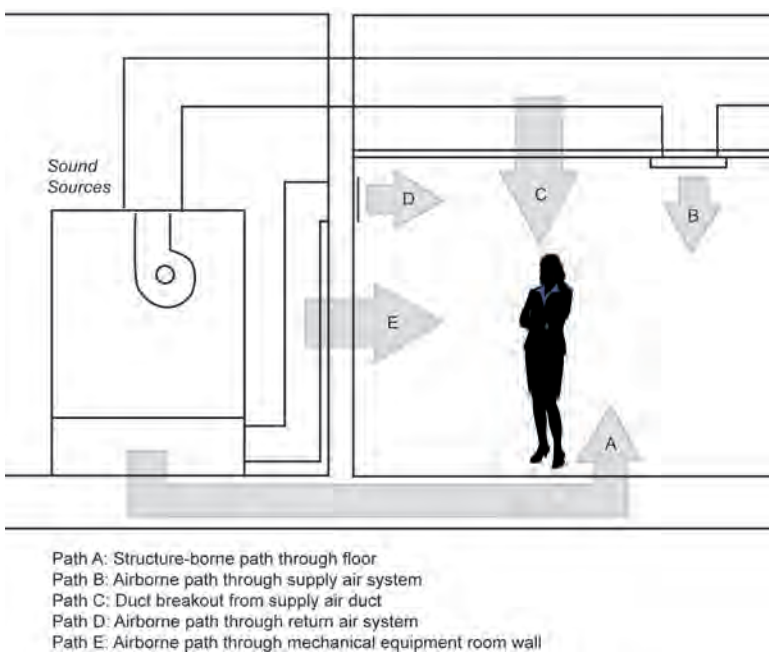

Figure 5-39 (HV34) Typical Noise Paths for Interior-Mounted HVAC Units

Location of air-conditioning or heat pump units and design of noise-attenuating separations should recognize acoustic requirements of adjacent spaces.

ASHRAE Handbook—HVAC Applications (ASHRAE 2011) is a potential source for recommended background sound levels in the various spaces that make up grocery stores.

HV35 *Proper Maintenance* (Climate Zones: all)

Regularly scheduled maintenance is an important part of keeping the HVAC system in optimum working condition. Neglecting preventive maintenance practices can quickly negate energy savings expected from the system design. ASHRAE/ACCA Standard 180 provides minimum requirements for maintenance and inspection and should be used to help develop a routine maintenance program for the building (ASHRAE 2012d).

All filters should be inspected monthly and should be cleaned or replaced when the pressure drop exceeds the filter manufacturer's recommendations for replacement or when visual inspection indicates the need for replacement. Energy recovery devices need to be inspected at least annually and cleaned periodically to maintain performance. Dampers, valves, louvers, and sensors must all be periodically inspected (at least semi-annually) and calibrated to ensure proper operation. This is especially important for outdoor air dampers and CO_2 sensors. Inaccurate CO_2 sensors can cause excessive energy use or poor IAQ, so they need to be calibrated as recommended by the manufacturer.

A BAS can be used to notify O&M staff when preventive maintenance procedures should be performed. This notification can be triggered by calendar dates, runtime hours, the number of times a piece of equipment has started, or sensors installed in the system (such as a pressure switch that indicates when an air filter is too dirty and needs to be replaced). Increase the frequency of inspections if indicators of unacceptable performance are found during two successive inspections.

Annual coil cleaning should be scheduled to maintain desired coil air approach temperature and to avoid additional air pressure drop.

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QUALITY ASSURANCE

Quality assurance (QA), including commissioning (Cx), will help ensure that a building and its systems function in accordance with the design intent and thus meet the performance goals established for it. QA should be an integral part of the design and construction processes and part of the continued operation of the facility. General information on QA and Cx is included in Chapter 2 of this Guide.

COMMISSIONING

QA1 *Design and Construction Team* (Climate Zones: all)

Selection of the design and construction team members is critical to a project's success. Owners and managers need to understand how team dynamics can play a role in the building's resulting performance. Owners should evaluate qualifications, past performance, costs of services, and the availability of the candidates in making their selection. Owners need to be clear in their expectations of how team members should interact. It should be clear that all members should work together to further team goals. The first step is to define members' roles and responsibilities. This includes defining deliverables at each Cx phase of the project.

QA2 *Selection of Quality Assurance Provider* (Climate Zones: all)

Quality assurance is a systematic process of verifying the Owner's Project Requirements (OPR), operational needs, and Basis of Design (BoD) and ensuring that the building's systems perform in accordance with the defined needs. The selection of a QA provider should include the same evaluation process the owner would use to select design and construction team members. Some of the parameters an owner should investigate and consider when reviewing the qualifications of the QA services provider are past performance of projects, costs of services, and the availability of the candidate. Owners may select a member of the design or construction team as the QA provider for the design and construction/project documents. While there are exceptions, most designers are not comfortable with the post-construction operating and testing of equipment and assemblies. Most contractors do not have the technical background necessary to evaluate system and equipment performance. Performing technical procedures in the Cx process requires in-depth technical knowledge of the building envelope and the refrigeration, mechanical, electrical, and plumbing systems, as well as operational and construction experience. This post-construction QA function is best performed by a third party responsible to the owner because conflicts of interest and political issues often inhibit members of the design and construction teams from fulfilling this responsibility.

QA3 *Owner's Project Requirements and Basis of Design* (Climate Zones: all)

The OPR details the functional requirements of a project and the expectations of how the facility will be used and operated. This includes strategies and recommendations selected from this Guide (see Chapter 4) that will be incorporated into the project, anticipated hours of operation provided by the owner, and BoD assumptions.

The OPR forms the foundation of the design team's tasks by defining project and design goals, measurable performance criteria, owner directives, budgets, and schedules and combining it with supporting information in a single, concise document. The QA process of the design and construction/project documents depends on a clear, concise, and comprehensive OPR. Development of the OPR requires input from all key facility users and operators. It is critical to align the complexity of the systems with the capacity and capability of the facility staff.

The next step is for the design team members to draft a narrative document of how their design responds to the OPR information. This document is the BoD. It records the standards and regulations, calculations, design criteria, decisions and assumptions, and system descriptions. The BoD must clearly articulate the specific operating parameters required for the systems to form the basis for later quality measurements. Essentially, this BoD is the engineering background information that is not provided in the CDs that map out how the architects and engineers end up with their designs. For example, the BoD should state key criteria such as future expansion and redundancy considerations. It should include important criteria such as what codes, standards, or guidelines are being followed for the various engineered systems, including ventilation and energy. The BoD provides a good place to document owner input needed for engineered systems, such as identifying what electrical loads are to be on emergency power.

QA4 *Design and Construction Schedule* (Climate Zones: all)

The inclusion of QA activities in the construction schedule fulfills a critical part of delivering a successful project. Identify the activities and time required for design review and performance verification to minimize time and effort needed to accomplish activities and correct deficiencies.

QA5 *Design Review* (Climate Zones: all)

The commissioning authority (CxA)/QA provider should provide a fresh perspective that allows identification of issues and opportunities to improve the quality of the CDs and verify that the OPR is being met. Issues identified in the design review can be more easily corrected early in the project, providing potential savings in construction costs and reducing risk to the team.

QA6 *Defining Quality Assurance at Prebid* (Climate Zones: all)

The building industry has traditionally delivered buildings without using a verification process. Changes in traditional design and construction procedures and practices require education of the construction team that explains how the QA process will affect the various trades bidding the project. It is extremely important that the QA process be reviewed with the bidding contractors to facilitate understanding of and to help minimize fear associated with new practices. Teams who have participated in the Cx process typically appreciate the process because they are able to resolve problems while their manpower and materials are still on the project, significantly reducing delays, callbacks, and associated costs while enhancing their delivery capacity.

These requirements can be reviewed by the Architect and Engineer of Record at the prebid meeting, as defined in the specifications.

QA7 *Verifying Building Envelope Construction* (Climate Zones: all)

The building envelope is a key element of an energy-efficient design. Compromises in assembly performance are common and are caused by a variety of factors that easily can be avoided. Improper placement of insulation, improper sealing or lack of sealing at air barriers, poorly selected or performing glazing and fenestration systems, incorrect placement of shading devices, misplacement of daylighting shelves, and misinterpretation of assembly details can significantly compromise the energy performance of the building (see the Cautions sections throughout this chapter). The value of the Cx process is that it is an extension of the QA processes of the designer and contractor as the team works together to produce quality, energy-efficient projects.

QA8 *Verifying Lighting Construction* (Climate Zones: all)

Lighting plays a significant role in the energy consumption of the building. Lighting for all of the space types should be reviewed against an anticipated schedule of use throughout the day.

QA9 *Verifying Electrical, Refrigeration, and HVAC Systems Construction* (Climate Zones: all)

The performance of electrical, refrigeration, and HVAC systems are key elements of this Guide. How systems are designed as well as installed affect how efficiently they will perform. Collaboration between the entire design team is needed to optimize the energy efficiency of the facility. Natural daylight and artificial lighting will impact the heating and cooling loads with respect to both capacity and operation mode. The design reviewers should pay close attention to the fact that proper installation is just as important as proper design. Making sure the installing contractor's foremen understand the owner's goals, the QA process, and the installation details is a key factor of system performance success. A significant part of this process is a careful and thorough review of product/equipment submittals to ensure compliance with the design plans and specifications. The timing of this review is critical to ensure that problems are identified at the beginning of each system installation, which minimizes the number of changes (and the associated added time and cost) and leaves time for corrections. It is also important to continue performing observations and inspections during and shortly after construction. For instance, the CxA will ideally have written checklists based on the equipment submittals reviewed during the construction administration phase. The CxA will turn over the checklists to the construction team to complete based on manufacturer start-up reports and other collected information (including warranty and wiring information). Once the checklists are complete, the CxA will do an on-site random sampling to check results and confirm that the reported findings are true and repeatable.

QA10 *Performance Testing* (Climate Zones: all)

Performance testing is essential to ensure that commissioned systems are properly implemented. Unlike most appliances these days, none of the mechanical/electrical systems in a new facility are "plug and play." Functional test procedures are often written in response to the contractor's detailed sequence of operations. The CxA will supervise the controls contractor running the equipment through its operations to prove adequate automatic reaction of the system to artificially applied inputs. The inputs simulate a variety of extreme, transition, emergency, and normal conditions.

If possible, it is useful to operate and monitor key aspects of the building for a one-month period just before contractor transfer to verify energy-related performance and the final setpoint configurations in the O&M documents. This allows the building operator to return the systems to their original commissioned states (assuming good maintenance) at a future point, with comparative results.

QA11 *Substantial Completion* (Climate Zones: all)

Substantial completion is achieved when life safety systems have been implemented and verified and the facility is ready to be occupied. All of the systems should be operating as intended. Expected performance can only be accomplished when all systems operate interactively to provide the desired results. As contractors finish their work, they will identify and resolve many performance problems. The CxA/QA provider verifies that the contractor maintained a QA process by directing and witnessing testing and then helps to resolve remaining issues.

QA12 *Final Acceptance* (Climate Zones: all)

Final acceptance generally occurs after the Cx/QA issues in the issues log have been resolved except for minor issues the owner is comfortable with resolving during the warranty period.

QA13 *Establish Building Operation and Maintenance (O&M) Program* (Climate Zones: all)

Continued performance and control of O&M costs require a maintenance program. The O&M manuals provide information that the O&M staff uses to develop this program. Detailed O&M system manual and training requirements are defined in the OPR and executed by the project team to ensure the O&M staff has the tools and skills necessary. The level of expertise typically associated with O&M staff for buildings covered by this Guide is generally much lower than that of a degreed or licensed engineer, and they typically need assistance with development of a preventive maintenance program. The CxA/QA provider can help bridge the knowledge gaps of the O&M staff and assist the owner with developing a program that will help ensure continued performance. The benefits associated with energy-efficient buildings are realized when systems perform as intended through proper design, construction, operation, and maintenance.

MEASUREMENT AND VERIFICATION

QA14 *Monitor Post-Occupancy Performance* (Climate Zones: all)

Establishing measurement and verification (M&V) procedures for actual building performance after a building has been commissioned can identify when corrective action and/or repair is required to maintain energy performance. Utility consumption and factors affecting utility consumption should be monitored and recorded to establish building performance during the first year of operation.

Variations in utility usage can be justified based on changes in conditions typically affecting energy use, such as weather, occupancy, operational schedule, maintenance procedures, and equipment operations required by these conditions. While most buildings covered in this Guide will not use a formal M&V process, tracking the specific parameters listed above does allow the owner to quickly review utility bills and changes in conditions. Poor building and equipment performance is generally obvious to the reviewer when comparing the various parameters. CxA/QA providers typically can help owners understand when operational tolerances are exceeded and can provide assistance in defining what actions may be required to return the building to peak performance.

Another purpose of the post-occupancy evaluation is to determine actual energy performance of low-energy buildings to verify design goals and document real-world energy savings. Additionally, the post-occupancy evaluation provides lessons learned in the design, technologies, and operation and analysis techniques to ensure these and future buildings operate at a high level of performance over time. For details and some case studies and lessons learned, refer to the published National Renewable Energy Laboratory (NREL) report (Torcellini et al. 2006).

QA15 *M&V Electrical Panel Guidance* (Climate Zones: all)

Designing the electrical distribution system to be submetered reduces complexity, minimizes the number of meters, shortens installation time, and minimizes rewiring. Disaggregate electrical panels (put lights together on one panel, HVAC on another, miscellaneous loads on a third, etc.), and repeat for emergency circuits. Meter as much as possible at the main distribution panel and repeat for emergency circuits to minimize installation and wiring costs. Consider using electrical panels with integral submeters to reduce capital costs. Integrate testing of the meters into the Cx plan to ensure that the submetering system is operating correctly.

QA16 *M&V Data Management and Access* (Climate Zones: all)

Detailed M&V systems can result in an overwhelming amount of data. The success of an M&V system depends on proper management of this data. Collect submetered data at resolutions appropriate for the intended use. For example, save 1 min data for one day to aid with equipment troubleshooting and identify failures; save data at 5 min intervals for one week to help analyze the building schedules; and save 15 min data for at least one year to help with benchmarking, to determine annual energy performance, to compare to the original energy model (weather variance removed), and to compare end-use benchmarks. In general, make sure you have sufficient data resolution to determine electricity demand information and equipment failures.

To ensure ease of interoperability and consistency with other submetering efforts for a portfolio of buildings, comply with the metering standard of the existing buildings. If one does not exist, consider developing a metering standard that documents interoperability and accessibility requirements. In addition, allow for external consultants and design team members to easily access the metered data remotely.

QA17 *M&V Benchmarking* (Climate Zones: all)

An owner should benchmark utility bills and submetered data to ensure energy performance targets are met and should be prepared to repeat this exercise monthly. CxA/QA providers typically can help owners understand when operational tolerances are exceeded and can help determine actions to return the building to peak performance.

Benchmarking a facility can identify poor performance in multiple ways. Submetered data can be benchmarked against previous trends, energy models, or other facilities with submetered data. Monthly energy performance should be benchmarked against historic performance and other facilities in a portfolio or other, similar facilities. Annual energy performance should be benchmarked using EPA's ENERGY STAR Portfolio Manager (EPA 2015) and the energy targets provided in this Guide (see Chapter 4).

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ADDITIONAL BONUS SAVINGS

ENVELOPE OPTIONS

EN30 *Vehicular Door—Opaque, High Speed* (Climate Zones: all)

Under specific usage conditions, high-speed vehicular doors can contribute additional energy savings by minimizing air exchange. These doors are characterized by minimum opening and closing speeds of 18 in./s, a maximum hold time of 10 s, and an automatic closing device. It is recommended that these doors be used only where there is a minimum average annualized 75 open/close cycles per day.

EN32 *Interspacial Airflows* (Climate Zones: all)

Interior partition walls that separate the back of the house (loading docks, receiving/storage areas, compressor rack spaces) from the rest of the store (sales area) need to be designed to minimize the airflow between those spaces to facilitate maintenance of humidity and thermal comfort. These walls need to be well sealed at the floor and roof intersections. Doors in these walls (typically these are double-acting, self-closing, impact doors) need to be designed with features (gaskets and/or brushes) that minimize the airflows while maintaining ease of use.

REFRIGERATION OPTIONS

RF32 *Improved Insulation and Vapor Retarder* (Climate Zones: all)

Improved insulation and vapor retarders for piping can be considered for additional savings. The standard for grocery store refrigeration piping, flexible elastomeric foam (FEF) insulation (typically 1/2 to 3/4 in. thickness), is designed primarily to avoid condensation. Additional insulation on conventional DX systems with individual piping circuits will probably not be cost-effective given the greater heat gain and cost and the practicality of increasing insulation on a large number of individual piping runs.

However, DX systems with loop piping design and piping for CO_2 and glycol indirect loops are a different situation, because of less total length and the larger diameter of pipes, so additional insulation is likely to be cost-effective in these situations. On DX loop systems, added insulation reduces subcooled liquid-line heating and suction-line superheating. Gains to the subcooled liquid line are equivalent to additional load on the associated suction group. Gains to the suction lines are not additional load per se, but they only produce a benefit to the extent that return gas temperature is reduced from what it would otherwise have been. Heat gains on indirect piping systems are fully realized as 100% cooling load; thus, insulation on indirect systems is more cost-effective than on DX suction lines. For additional information, see the discussion in the Mass-Flow-Based Design section in Chapter 3.

The most effective way to improve insulation is to increase the thickness of the FEF insulation currently in common use. For example, use 1 in. thickness on medium-temperature suction and secondary piping and all subcooled liquid lines, and use 1.5 to 2 in. thickness on low-temperature suction and indirect piping.

A key aspect with any pipe insulation system is the integrity of the vapor retarder. Historically, FEF insulation has exhibited a reasonable life, but with known variability. In conditioned spaces, in relatively dry climates, and on suction lines that are subject to gas defrost heating on a daily basis, moisture gain may not be evident. In harsh and humid locations, on continuous low-temperature piping, FEF insulation could have significant moisture gain and even fail within a matter of several years or months.

For longevity and ongoing maintained insulation value, consider a separate film or tape vapor retarder with a permeance rating of ≤0.02 perm, as recommended in *ASHRAE Handbook—Refrigeration* (ASHRAE 2014). This is a significant improvement as compared with the vapor retarder inherent with FEF insulation, where the permeance rating may also vary with insulation thickness. FEF insulation manufacturers may also have product options and offerings that improve the vapor retarder.

OTHER HVAC SYSTEM TYPES

HV37 *Ground-Source Heat Pump (GSHP) with DOAS* (Climate Zones: all)

This variation of the WSHP system takes advantage of the high thermal capacitance of the earth to store heat rejected into the ground during the cooling cycle as a resource for the heating cycle. In general, successful implementation of a ground-coupled heat pump system requires relative balance between the amount of heat extracted from the ground for the heating cycle and the amount of heat rejected into the ground for the cooling cycle. An appropriately sized ground-coupling system results in a lower heat rejection temperature during the cooling cycle compared with cooling tower heat rejection. While the lower heat extraction temperature of a ground-coupled system compared with fuel-fired makeup typically results in a lower COP, performance is improved by the fact that no energy is consumed for makeup heating.

Performance characteristics of selected GSHP units should be the same as the WSHP specifications listed in HV8. GSHP units used with a closed-loop well system should have a rated low-temperature heating capability at a minimum temperature not more than 30°F. The circulating fluid for a closed-loop ground-coupled heat pump system should incorporate an antifreeze additive to prevent icing of the loop.

External air pressure drop for these units, as with WSHPs, should be limited to 0.7 in. w.c.

Following are some considerations for incorporation of a ground-coupled heat pump:

- Balance of winter heating loads with summer cooling loads
- Accurate determination of heat diffusivity of earth in contact with the ground-coupled heat transfer system, at a minimum through use of a test well to determine the nature of ground strata and ground water levels
- Adequate sizing of the ground-coupling system, using accurate ground thermal diffusivity information to limit minimum supply water temperature during the winter and maximum supply water temperature during the summer

Ensure appropriate design and control of the hydronic circulation system to optimize pumping energy and maximize heat pump annual heating and cooling efficiency.

HV38 *Custom Air-Handling Units* (Climate Zones: all)

For especially humid climates, special dehumidification capabilities may be required to meet the 50% savings target. These may include desiccant dehumidification and various forms of heat recovery reheat.

Desiccant dehumidification is typically useful in locations for which the sensible cooling provided by conventional condensing dehumidification must be offset to maintain space comfort conditions. One approach to desiccant dehumidification is to use a sidestream unit downstream of the cooling coil. A portion of the supply air is routed through the desiccant device and is both heated and dehumidified before it is remixed with the primary airstream. The resulting mixed air is both warmer and dryer than the supply air upstream of the desiccant device.

Passive heat recovery reheat devices typically consist of heat transfer surfaces in the air upstream of the cooling coil connected with heat transfer surfaces downstream of the coil. The two heat transfer surfaces, typically fin-tube coils, may be connected with a pumped loop or a heat pipe refrigerant loop. Sensible heat recovery wheels between the return air path and the supply air path accomplish the same goal. This technology enables the air downstream of the cooling coil to be reheated with heat harvested upstream of the coil, resulting in little added energy for reheat other than pump energy for a pumped loop. This technique is typically more effective for recirculating or mixed-air systems than DOAS units. For DOAS units, the most challenging ambient condition is a rainy day between 60°F and 65°. In this condition, little heat can be harvested upstream of the coil, and space sensible loads, as influenced by the exterior ambient temperature, will be reduced by the lower exterior temperature, even though dehumidification requirements remain high.

For mixed-air AHUs, separate conditioning paths for outdoor air and recirculated air may be provided. A separate cooling coil for the outdoor stream may be controlled to maintain a maximum leaving-air dew-point temperature limit no matter the sensible heating or cooling requirements in the space. Coils in the mixed airstream can be used to adjust the dry-bulb temperature of the mixed supply air to the ideal setpoint for space sensible heating or cooling conditions.

The 50% savings goal has been achieved for the systems listed for typical climates for the eight climate zones. More extreme humid climates or specialized conditions of the candidate building may require strategies as described above to reach the 50% savings goal.

RENEWABLE ENERGY

RE1 *Photovoltaic (PV) Systems* (Climate Zones: all)

Photovoltaic (PV) systems have become an increasingly popular option for on-site electric energy production for energy cost savings in grocery stores. These systems require very little maintenance and have long lifetimes but are often difficult to cost-justify without alternative financing and leveraging available incentives. However, the average cost of PV systems has declined significantly in recent years. PV systems can be effectively used in grocery stores in almost all climate zones in the United States. Figure 5-40 shows the PV solar energy resources in the U.S.

Options for PV system installations include rooftop (including collectors integrated with the roofing membrane), ground mounted, or as the top of a covered parking system. Unshaded south-facing standing-seam metal roofs or flat membrane roofs offer the simplest and most cost-effective mounting surfaces for rooftop PV systems. Ensure roof structures are adequate for the added weight of the PV system and be aware of any rooftop warranty implications from adding PV systems. Use roof mounting systems that do not penetrate the continuous insulation of the roof deck, such as a self-ballasted racking system. To allow for easy installation of PV systems on the roofs of grocery stores, provide extra conduit from the roof to the inverters, typically located in either an electrical room or a secure outdoor location. Also ensure the main electrical distribution system, from the main distribution panel to the building transformer, has the capability of carrying additional PV wiring.

There are many unique funding opportunities for PV systems in grocery stores. In addition to the many rebate programs offered by state and local utility companies, there are often significant incentives, loans, grants, and buyback programs for PV systems in grocery stores. DSIRE™, the Database for State Incentives for Renewables & Efficiency (NC State 2015), shows some opportunities available to grocery retailers in various states.

There are numerous tools available for modeling energy production from PV systems. One such tool is NREL's PVWatts™ calculator, available on the NREL website as part of their Renewable Data Resource Center (NREL 2014). The tool determines the energy production and cost savings of grid-connected PV energy systems throughout the world. It allows grocery store designers, installers, and manufacturers to easily develop estimates of the performance of PV installations.

RE2 *Wind Turbine Power* (Climate Zones: all)

Wind energy is one of the lowest-priced renewable energy technologies available today, costing between $0.05 and $0.11 per kilowatt-hour, depending upon the wind resource and project financing of the particular project. For grocery stores, small to medium-sized wind turbines are typically considered. These turbines range from 4 to 200 kW and are typically mounted on towers from 50 to 100 ft tall and connected to the utility grid through the building's electrical distribution system.

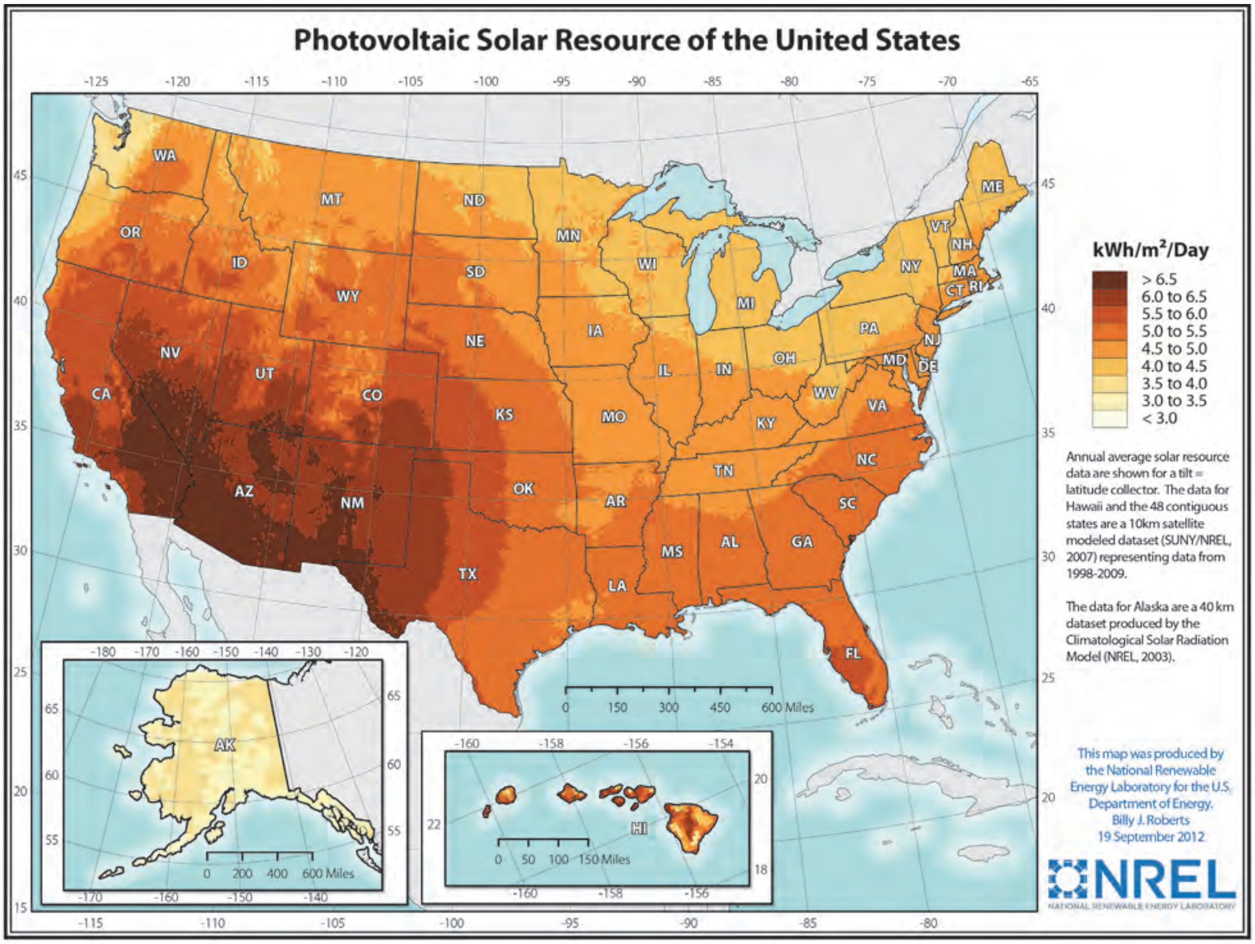

Figure 5-40 (RE1) Photovoltaic Solar Resources of the U.S.
Source: NREL (2015a)

One of the first steps to developing a wind energy project is to assess the area's wind resources and estimate the available energy. It can be determined from wind resource maps (NREL 2015b) whether an area of interest should be further explored. Note that the wind resource at a micro level can vary significantly; therefore, a professional evaluation of the specific area of interest should be performed.

The map in Figure 5-41 shows the annual average wind power estimates at 50 m above ground. It combines high- and low-resolution datasets that have been screened to eliminate land-based areas unlikely to be developed due to land use or environmental issues. In many states, the wind resource has been visually enhanced to better show the distribution on ridge crests and other features. Estimates of the wind resource are expressed in wind power classes ranging from Class 1 (lowest) to Class 7 (highest), with each class representing a range of mean wind power density or equivalent mean speed at specified heights above the ground. The map in FIgure 5-41 does not show Classes 1 and 2, as Class 1 areas are unsuitable and Class 2 areas are marginal for wind energy development. In general, at 50 m, wind power Class 4 or higher can be useful for generating wind power. More detailed state wind maps are available at the WINDExchange website (EERE 2014).

Although wind turbines themselves do not take up a significant amount of space, they need to be installed an adequate distance from the nearest building for several reasons, including turbulence reduction (which affects efficiency), noise control, and safety. It is essential that coordination occurs between the owner, design team, and site planner to establish the optimal wind turbine location relative to the other facilities on the site.

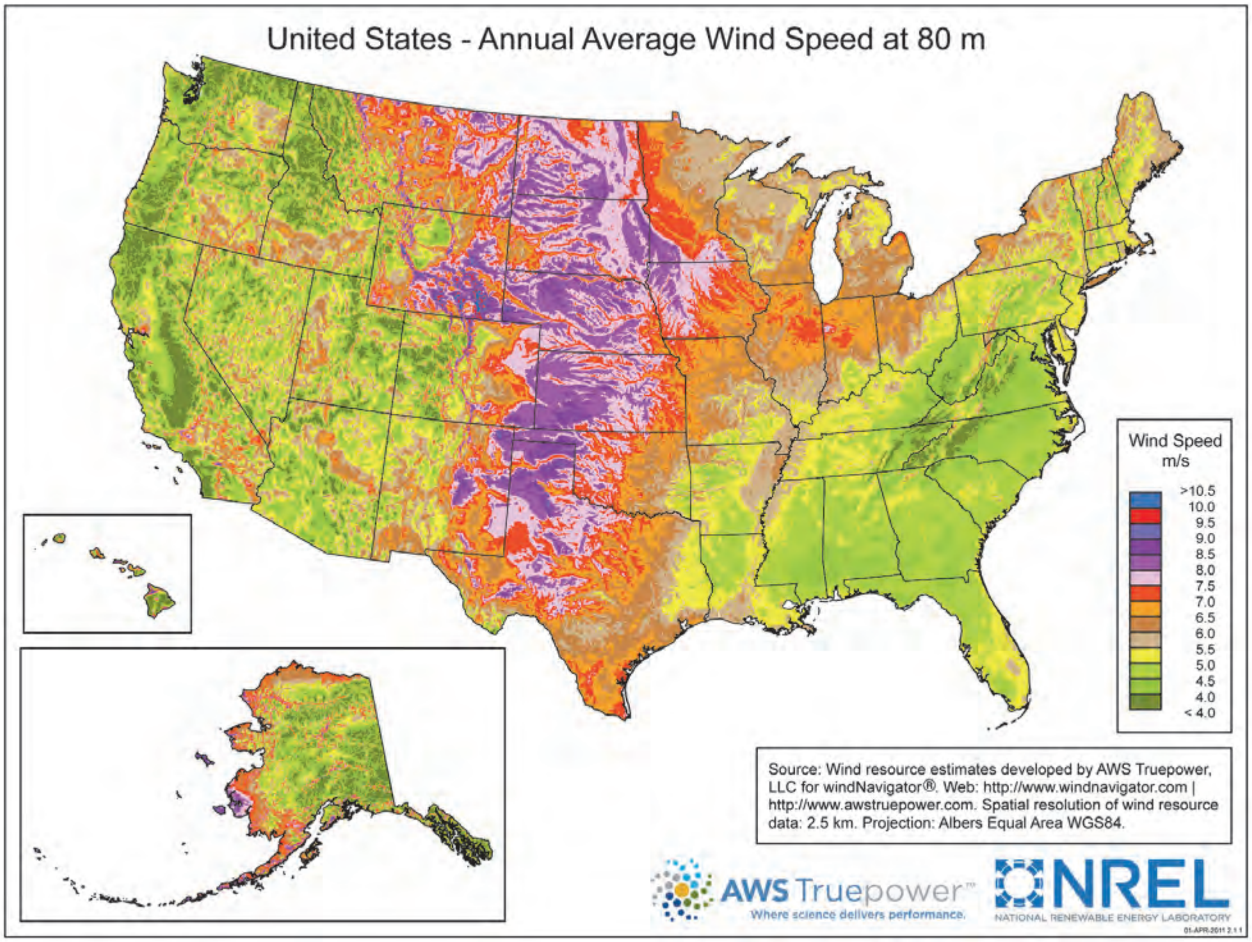

Figure 5-41 (RE2) Average Annual Wind Power Estimates
Source: EERE (2014)

The three largest complaints about wind turbines are noise, hazard to birds, and aesthetic appearance. Most of these problems have been resolved or greatly reduced through technological development or through proper siting of wind turbines. Most small wind turbines today have an excellent safety record. An important factor to consider is how the wind turbine controls itself and shuts itself down. Can operators shut it off and stop the turbine when they want or need to do so? This is extremely important, yet unfortunately there are very few small turbines that have reliable means to stop the rotor on command. The few that do may require you to do so from the base of the tower—not exactly where you want to be if the turbine is out of control in a wind storm. Look for a system that offers one or more means to shut down and preferably stop the rotor remotely.

While the market for large-scale wind turbines is dominated by horizontal axis machines, there are significant advantages to using vertical axis machines for grocery store on-site generation. While horizontal axis machines are generally more efficient at converting wind energy to electricity in ideal locations, vertical axis machines can take advantage of turbulent wind regimes that are generally common near buildings and produce less vibration and noise.

Using energy modeling, the electric energy consumption of a building can be estimated. Using this data in conjunction with the financial details of the project, including the available rebates, the owner and designer must choose the size of turbine that meets their needs. Note that the closer the match of the turbine energy output to the demand, the more cost-effective the system will be. Make sure that all costs are listed to give a total cost of ownership for the wind turbine. This includes the wind turbine, tower, electrical interconnection, controls, installation, maintenance, concrete footings, guy wires, and cabling.

In addition to evaluating the initial cost of the turbine, it is extremely important to consider the federal and state policies and incentive programs that are available. DSIRE™ (NC State 2015) provides a list of available incentives, grants, and rebates. Also critical to the financial success of a wind turbine project is a favorable net metering agreement with the utility.

RE3 *Transpired Solar Collector* (Climate Zones: 3 4 5 6 7)

A transpired solar collector is a renewable energy technology that, when coupled with a mechanical system, provides free heating of the air. As illustrated in Figure 5-42, the system is composed of a perforated metal panel with an air cavity between the panel and the exterior wall. As the panel absorbs solar radiation, air is drawn into the cavity. As the air passes over the surface of the wall, the air is warmed.

Warm air is drawn into the mechanical system during heating mode. The free heating of the air can significantly reduce the demand for electric or fossil-fuel heating. When heated air is not desired, a bypass damper allows the system to relieve the warm air out of the cavity. Equally important, a separate outdoor air location is required at these times to provide ventilation air using ambient-temperature air rather than the warm air in the transpired solar collector.

Since buildings often go through a morning warm-up, east- and south-facing walls are most suitable for transpired solar collector installations. In very cold climates, a west-facing wall may also be suitable.

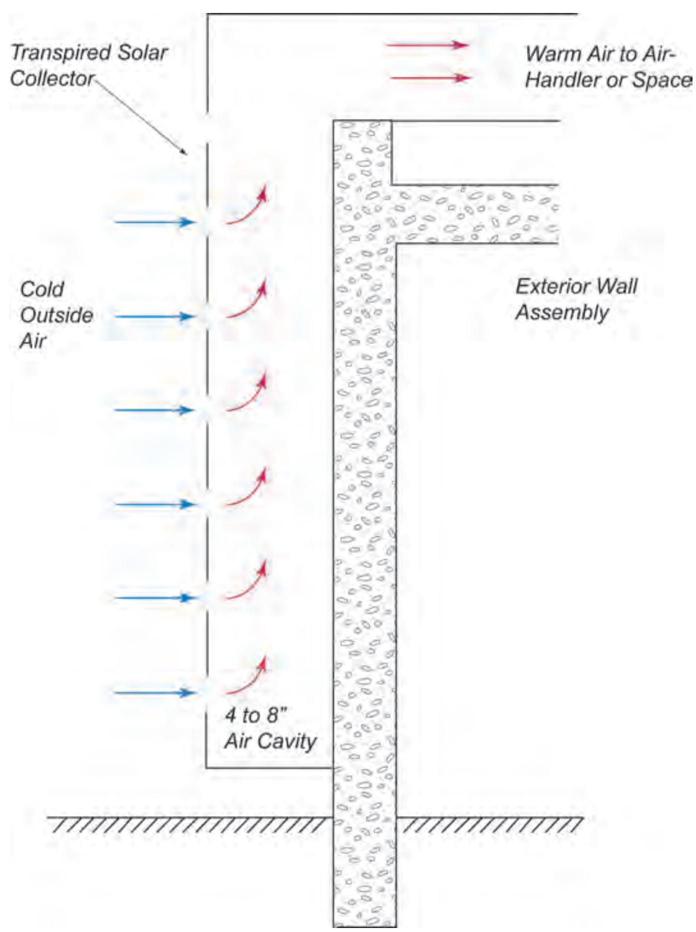

Figure 5-42 (RE3) Transpired Solar Collector

RE4 *Power Purchase Agreements* (Climate Zones: all)

A primary barrier to the use of various on-site renewable energy strategies is the high initial capital investment cost. One way to finance and thus implement such a strategy is through a power purchase agreement. Grocery stores have successfully implemented renewable energy systems, such as photovoltaic, wind, and solar hot water systems, using these financing programs.

Power purchase agreements involve a third party who will design, install, own, operate, and maintain the power generation asset. The retailer contracts to purchase the energy produced by the generation system, usually for a long period of time. This arrangement allows the retailer to avoid the high first cost and keeps the balance sheet clear of obligation. It also locks in an energy price, thus hedging the cost of energy over time from fluctuations in the prices of other energy sources. The agreements are complicated, with many considerations, and require negotiation by people familiar with the complexities, from an engineering perspective as well as from legal and financial perspectives.

RE5 *Renewable Energy Credits* (Climate Zones: all)

Renewable energy credits provide an opportunity for grocery store operators to purchase renewable energy without the need to install generating assets on site. Renewable energy credits are available as bundled, meaning the credits are associated with the actual energy purchase, or unbundled, meaning the credits are purchased separately from the energy.

RE6 *Electrochemical and Thermal Storage* (Climate Zones: all)

Various technologies exist to provide energy storage that can accompany renewable energy systems and provide added value to grocery store operators. Battery storage can help mitigate peak demand events and reduce demand and capacity charges from utilities. Also, when combined with renewable energy sources such as photovoltaic or wind turbines, battery storage can allow grocery store operators to better utilize the locally generated power by using it when grid power is most expensive. Additionally, addition of battery storage can provide grocery stores with a limited amount of backup power that can keep life safety systems up and running during a power outage.

Thermal energy storage is also an option for grocery stores. Several options are available to designers, including thermal storage of hot water from refrigeration heat reclaim coils and thermal storage of cooling, allowing shifting of compressor runtime to off-peak hours while maintaining cooling.

RE7 *Combined Heat and Power* (Climate Zones: all)

On-site combined heat and power offers several advantages to grocery store owners when economic conditions are right. On-site power offers backup power generation when grid power is unavailable, providing insurance against product loss. Likewise, on-site power generation can be a hedge against volatility in the electric market by providing the operator with flexibility in power sourcing. Finally, by using the waste heat from on-site thermal generators such as engines, microturbines, or fuel cells, domestic hot water and/or auxiliary cooling can be provided in addition to electric power. Such strategies increase the overall efficiency of the energy conversion and thus reduce operating expenses.

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Appendix A— Envelope Thermal Performance Factors

A

Each climate zone recommendation table in Chapter 4 presents a prescriptive construction option for each opaque envelope measure. Table A-1 presents U-factors for above-grade components and F-factors for slab-on-grade floors that correspond to each prescriptive construction option. Alternative constructions would be equivalent methods for meeting the recommendations of this Guide provided they are less than or equal to the thermal performance factors listed in Table A-1.

Table A-1 Opaque Construction Options

Roof Assemblies	
R	U
Insulation Above Deck	
20 c.i.	0.048
25 c.i.	0.039
30 c.i.	0.032
35 c.i.	0.028
Metal Building	
19 +10 FC	0.041
19 + 11 Ls	0.037
25 + 11 Ls	0.031
30 + 11 Ls	0.029
25+11+11 Ls	0.026

Slabs	
R-in.	F
Unheated	
15–24	0.52
20–24	0.51
20–48	0.434
Heated	
7.5–12	1.02
10–24	0.90
15–24	0.86
20–24	0.843
20–48	0.688
25–48	0.671
20–24	0.843

Walls, Above Grade	
R	U
Mass Walls	
5.7 c.i.	0.151
7.6 c.i.	0.123
9.5 c.i.	0.104
11.4 c.i.	0.090
13.3 c.i.	0.080
15.2 c.i.	0.071
19.0 c.i.	0.048
Steel Framed	
13 c.i.	0.124
13 + 3.8 c.i.	0.084
13 + 5.0 c.i.	0.077
13 + 7.5 c.i.	0.064
13 + 10.0 c.i.	0.055
13 + 12.5 c.i.	0.049
13 + 18.8 c.i.	0.037
Metal Building	
0 + 9.8 c.i.	0.094
0 + 15.8 c.i.	0.060
0 + 19.0 c.i.	0.050
0 + 22.1 c.i.	0.044
0 + 25.0 c.i.	0.039

Walls, Below Grade	
R	C
7.5 c.i.	0.119
10 c.i.	0.092
15 c.i.	0.063

Floors	
R	U
Mass	
6.3 c.i.	0.107
10.0 c.i.	0.074
14.6 c.i.	0.056
16.7 c.i.	0.051
20.9 c.i.	0.042
23.0 c.i.	0.038
Steel Framed	
30	0.038
38	0.032

C = thermal conductance, Btu/h·ft^2·°F
c.i. = continuous insulation
F = slab edge heat loss coefficient per foot of perimeter, Btu/h·ft·°F
FC = filled cavity
Ls = liner system
R = thermal resistance, h·ft^2·°F/Btu
R-in. = R-value followed by the depth of insulation in inches
U = thermal transmittance, Btu/h·ft^2·°F

Note: All information in this table is in Inch-Pound (I-P) units. For slabs, the “in.” refers to the depth of the vertical slab edge insulation. See ASHRAE/IES Standard 90.1-2013, p. 150, for additional explanation (ASHRAE 2013). All units used in this table are defined in the Abbreviations and Acronyms section at the beginning of this Guide.

REFERENCE

ASHRAE. 2013. ANSI/ASHRAE/IES Standard 90.1-2013, *Energy Standard for Buildings Except Low-Rise Residential Buildings*. Atlanta: ASHRAE.

Appendix B— International Climatic Zone Definitions

B

Table B-1 shows the climate zone definitions that are applicable to any location. The information is from ANSI/ASHRAE/IES Standard 90.1-2010, Normative Appendix B, Table B-4 (ASHRAE 2010). Weather data are needed in order to use the climate zone definitions for a particular city. Weather data for a number of cities in Canada and Mexico are available on the AEDG web page (www.ashrae.org/AEDG) under "Additional Information." Weather data by city are available for a large number of international cities in *ASHRAE Handbook—Fundamentals* (ASHRAE 2013).

Table B-1 International Climatic Zone Definitions

Climate Zone Number	Name	Thermal Criteria*
1A and 1B	Very Hot–Humid (1A) Dry (1B)	9000 < CDD50°F
2A and 2B	Hot–Humid (2A) Dry (2B)	6300 < CDD50°F ≤ 9000
3A and 3B	Warm–Humid (3A) Dry (3B)	4500 < CDD50°F ≤ 6300
3C	Warm–Marine (3C)	CDD50°F ≤ 4500 and HDD65°F ≤ 3600
4A and 4B	Mixed–Humid (4A) Dry (4B)	CDD50°F ≤ 4500 and 3600 < HDD65°F ≤ 5400
4C	Mixed–Marine (4C)	3600 < HDD65°F ≤ 5400
5A, 5B, and 5C	Cool–Humid (5A) Dry (5B) Marine (5C)	5400 < HDD65°F ≤ 7200
6A and 6B	Cold–Humid (6A) Dry (6B)	7200 < HDD65°F ≤ 9000
7	Very Cold	9000 < HDD65°F ≤ 12600
8	Subarctic	12600 < HDD65°F

*CDD = cooling degree-day, HDD = heating degree-day.

DEFINITIONS

Marine (C) Definition—Locations meeting all four of the following criteria:

- Mean temperature of coldest month between 27°F and 65°F
- Warmest month mean < 72°F
- At least four months with mean temperatures over 50°F
- Dry season in summer. The month with the heaviest precipitation in the cold season has at least three times as much precipitation as the month with the least precipitation in the rest of the year. The cold season is October through March in the Northern Hemisphere and April through September in the Southern Hemisphere.

Dry (B) Definition—Locations meeting the following criterion:

- Not marine and $P < 0.44 \times (T - 19.5)$
  where
  P = annual precipitation, in.
  T = annual mean temperature, °F

Moist (A) Definition**—**Locations that are not marine and not dry.

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ASHRAE. 2013. *ASHRAE Handbook—Fundamentals*. Atlanta: ASHRAE.
ASHRAE. 2010. ANSI/ASHRAE/IES Standard 90.1-2010, *Energy Standard for Buildings Except Low-Rise Residential Buildings*. Atlanta: ASHRAE.

Advanced Energy Design Guide for Grocery Stores

This Guide was prepared under ASHRAE Special Project 138.

Advanced Energy Design Guide for Grocery Stores is the fifth in a series designed to provide recommendations for achieving 50% energy savings over the minimum code requirements of ANSI/ASHRAE/IESNA Standard 90.1-2004. The energy savings target of 50% is the next step toward achieving a net zero energy building, which is defined as a building that, on an annual basis, draws from outside resources equal or less energy than it provides using on-site renewable energy sources. ANSI/ASHRAE/IESNA Standard 90.1-2004 provides the fixed reference point and serves as a consistent baseline and scale for all of the 50% Advanced Energy Design Guides.

This Guide focuses on grocery stores ranging in size from 25,000 to 65,000 ft^2 with medium- and low-temperature refrigerated cases and walk-ins. The information in this Guide can be combined with that in *Advanced Energy Design Guide for Medium to Big-Box Retail Buildings* and used for larger stores that consist of both grocery and general merchandise. This Guide does not cover parking garages, campus utilities such as chilled water and steam, water use, or sewage disposal.

The specific energy-saving recommendations are summarized in a single table for each climate zone and allow contractors, consulting engineers, architects, and designers to easily achieve advanced levels of energy savings without detailed energy modeling or analyses.

In addition, this Guide discusses principles of integrated design and how they can be used to implement energy-efficient strategies. A chapter addressing design philosophies for grocery stores is also included. This chapter is devoted primarily to refrigeration as well as the interaction between refrigeration and other building systems.

An expanded section of tips and approaches is included in the “How to Implement Recommendations” chapter. These tips are cross-referenced with the recommendation tables. This chapter also includes additional “bonus” recommendations that identify opportunities to incorporate greater energy savings into the design of the building.

Case studies and technical examples throughout the Guide illustrate the recommendations and demonstrate the technologies in real-world applications.

For more information on the entire *Advanced Energy Design Guide* series,
please visit www.ashrae.org/aedg.

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