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Building Technologies Office

Energy Savings Potential and Research & Development Opportunities for Commercial Refrigeration

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NOTICE

Energy Savings Potential and R&D Opportunities for Commercial Refrigeration

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List of Abbreviations

ADL	Arthur D. Little, Inc
AHRI	Air-conditioning and Refrigeration Institute
AMR	Active magnetic regenerator
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BF	Ballast factor
BT	U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Building Technologies Program
Btu	British thermal unit
CBECS	Commercial Buildings Energy Consumption Survey
CEC	California Energy Commission
CEE	Consortium for Energy Efficiency
CFM	Cubic feet per minute
COP	Coefficient of performance
CRE	Commercial refrigeration equipment
DOE	U.S. Department of Energy
ECM	Electronically commutated permanent magnet
EER	Energy efficiency ratio
EERE	U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy
EEV	Electronic expansion valves
EHD	Electro-hydrodynamic
EIA	U.S. Department of Energy, Energy Information Administration
EISA	Energy Independence and Security Act of 2007
EPACT	Energy Policy Act of 2005
FEMP	Federal Energy Management Program
GWP	Global warming potential
HFC	Hydrofluorocarbon
HID	High intensity discharge
HP	Horse power
HVAC	Heating, ventilation, and air-conditioning
IMH	Ice-making head
kWh	Kilowatt-hour
LED	Light-emitting diode
LPW	Lumens per watt
MCE	Magnetocaloric effect
MMBtu	Million British thermal units
MR	Magnetic refrigeration
NAFEM	North American Food Equipment Manufacturers
NCI	Navigant Consulting, Inc
OEM	Original equipment manufacturer
PBP	Payback period
PSC	Permanent split capacitor

Quad	Quadrillion (10^{15}) British thermal units
RCU	Remote condensing unit
R&D	Research and development
R, D & D	Research, Development, and Demonstration
RTTC	Southern California Edison Refrigeration and Thermal Test Center
SCU	Self-contained unit
SEER	Seasonal energy efficiency rating
SMMA	Motor and Motion Association
SPM	Shaded pole motor
TA	Thermoacoustic
TBtu	Trillion (10^{12}) British thermal units
TDA	Total display area
TE	Thermo-electric
TWh	Terawatt-hour (10^{19} kWh)
TXV	Thermostatic expansion valve
UEC	Unit energy consumption
VIP	Vacuum-insulated pane

Executive Summary

This study documents the energy consumption of commercial refrigeration equipment (CRE) in the U.S. and evaluates the energy savings potential of various technologies and energy efficiency measures that could be applied to such equipment. The equipment and systems considered in this analysis include all major commercial refrigeration equipment categories, specifically:

- Supermarket refrigeration systems, including:
    - Display cases and walk-in coolers/freezers
    - Machine rooms (with compressor racks),
    - Condensing units
    - Interconnecting piping
    - Controls
- Self-contained food service equipment (preparation tables, buffet tables, etc.),
- Self-contained beverage merchandisers,
- Self-contained reach-in refrigerators and freezers,
- Self-contained ice machines,
- Self-contained refrigerated vending machines,
- Walk-in coolers and freezers having dedicated refrigeration systems (i.e., those not cooled by supermarket refrigeration systems).

This study does not include equipment types that do not consume substantial amounts of energy on a national basis, such as:

- Refrigerated water coolers (drinking-water fountains and bubblers)
- Self-contained merchandisers other than beverage merchandisers (such as merchandisers for bagged ice and ice cream).

Further, while they may consume significant amounts of energy, refrigeration systems used in food distribution warehouses are normally considered industrial refrigeration and were therefore beyond the scope of this study.

Energy Consumption

Estimates of commercial refrigeration energy consumption have varied significantly in recent years, ranging from approximately 0.73 to 1.10 quadrillion Btu (quad)[1] of primary energy consumption[2], representing 4.1 to 6.3% of the total primary energy used in commercial buildings (DOE 2006, DOE 2008a). **Figure ES-1** shows the 2006 annual primary energy consumption in commercial buildings by end use according to the 2006 Buildings Energy Data Book (DOE 2008a), indicating that commercial refrigeration equipment accounts for about 4% of total commercial building energy consumption. The 2006 data is the most recent information published by the DOE.

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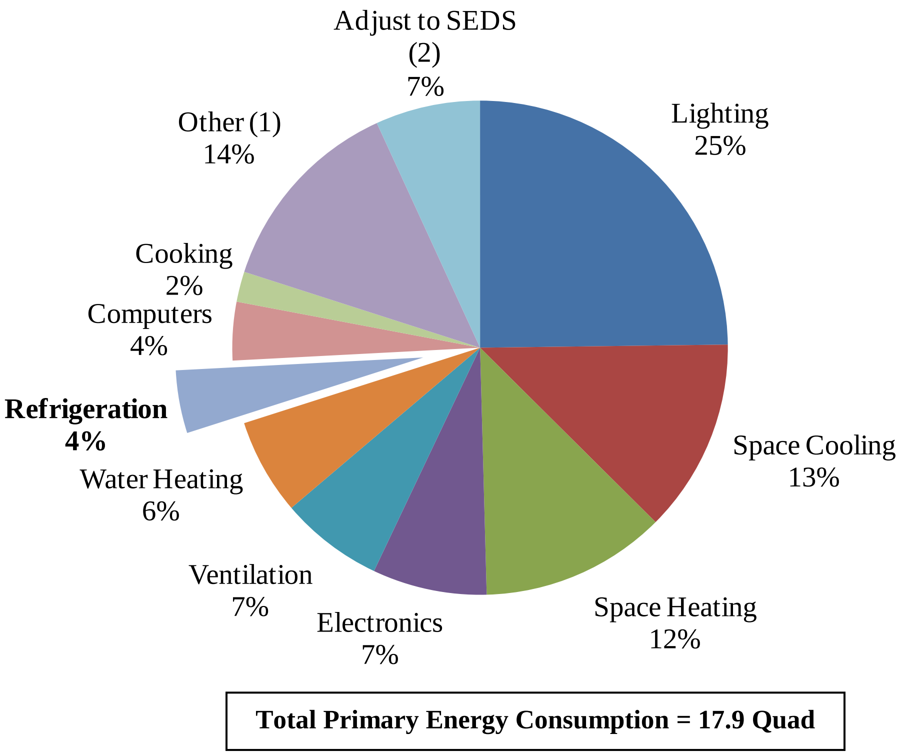
[1] A quad is a unit of energy equal to one quadrillion (10^{15}) British thermal units.
[2] Primary energy includes energy consumed in the generation, transmission, and distribution of electricity.



Source: DOE 2008a

Figure ES-1: Annual US Primary Energy Use of Commercial Buildings by End Use, 2006

Notes:
(1) "Other" includes service station equipment, ATMs, telecommunications equipment, medical equipment, pumps, emergency electric generators, combined heat and power in commercial buildings, and manufacturing performed in commercial buildings.
(2) "Adjust to SEDS" represents energy attributable to the commercial buildings sector, but not directly to specific end-uses, used by EIA to relieve discrepancies between data sources.

Based on the bottom-up analysis employed in this study, we estimate that primary energy consumption due to commercial refrigeration in 2008 was approximately 1.23 Quad, similar to the DOE estimate for 2004, with moderate market growth. **Table ES-1** compares the estimates of annual CRE energy consumption from this report, two recent DOE Buildings Energy Data Books, and a 1996 report by Arthur D. Little, Inc. (ADL, 1996).

Table ES-1: Comparison of Annual CRE Energy Consumption Estimates

Source	Year of Data	Annual CRE Energy Consumption (Quad)	% of Total Commercial Building Energy Use
This Report	2008	1.23	N/A [1]
2008 Buildings Energy Data Book (DOE 2008a)	2006	0.73	4.1%
2006 Buildings Energy Data Book (DOE 2006)	2004	1.10	6.3%

ADL 1996	1996	0.99	7.0%
[1] Data regarding commercial building energy usage in 2008 has not yet been published. 1.23 Quad would amount to 7.1 % of 2006 commercial building energy consumption.			

Figure ES-2 shows an estimated breakdown of total CRE energy consumption by equipment segment. Compressor racks, display cases, condensers, and supermarket walk-ins, which together represent supermarket refrigeration systems, use over 50 percent of the total energy used by commercial refrigeration. Walk-in coolers and freezers (other than supermarket walk-ins), which are typically used for food storage in food service applications, are the next largest energy-consuming category at 12 percent.

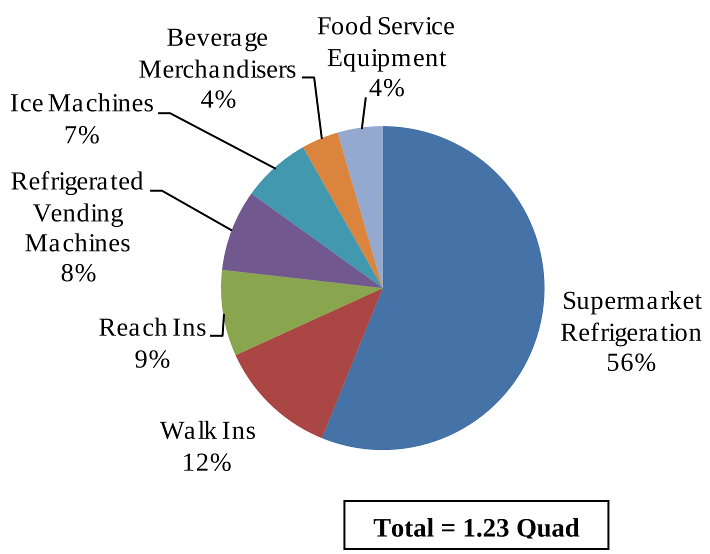


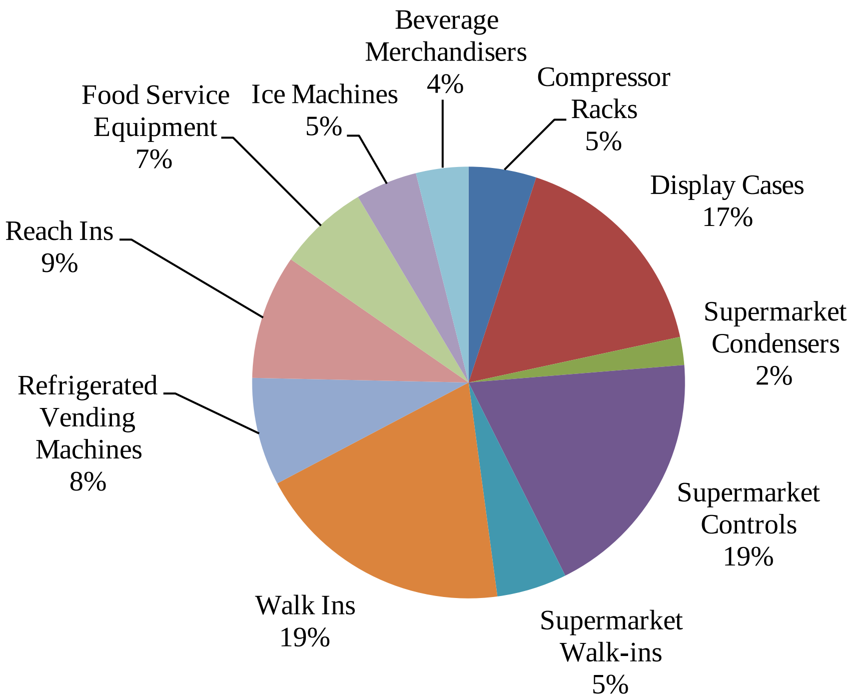
Figure ES-2: Annual Primary Energy Usage of Commercial Refrigeration by Equipment Type (2008)

Energy-Savings Opportunities using Commercially Available Technologies

Results of this study suggest that approximately 0.12 quad/yr could be saved if the installed base were simply replaced by typical new equipment. An additional 0.41 quad in annual CRE energy use could be saved by incorporating technologies and components that are currently commercially available but not widely implemented,[3] as shown in **Figure ES-3**. The technologies and components considered for this estimate could be implemented with simple payback periods of 7 years or less. High efficiency fan motors and compressors are applicable to many equipment types and are the most common improvements represented by this estimate. Supermarket refrigeration systems account for most of the potential energy savings in the CRE industry, followed by walk-ins and refrigerated vending machines.

[3] These commercially available technologies are not necessarily available in current equipment packages.

Nearly all refrigeration energy savings in supermarkets and up to 80 percent in CRE with dedicated refrigeration systems can be achieved using better controls, improved fan motors, high efficiency compressors, high efficiency lighting, and advanced door technologies.



Total Energy Savings = 0.41 Quad/year

Figure ES-3: Annual Potential Primary Energy Savings from Commercially Available Technologies

Energy Savings Opportunities Requiring Research, Development, and Demonstration
This study also identified technologies under development or investigation that are not yet commercially available, but that have the potential for significant additional energy savings. While technologies such as light-emitting diode (LED) lighting and high efficiency fan blades have already begun to enter the market, they are not yet cost-competitive for commercial refrigeration applications and require further cost-reduction to assure widespread adoption. A range of higher risk technologies continue to be investigated by companies and universities, and DOE could play a prominent role in supporting research, development and demonstration (RD&D) for these technologies. Alternative refrigeration systems, such as thermo-electric, thermo-acoustic, and magnetic refrigeration, theoretically have the potential to save up to 30 percent of system energy use, but significant breakthroughs in areas such as materials are needed before they can compete economically with conventional vapor-compression refrigeration. Vacuum panel insulation also has the potential to offer substantial system energy savings, once cost and performance issues have been resolved.

Barriers
Significant barriers in the commercial refrigeration industry make the widespread implementation of high efficiency options difficult. Equipment buyers are under pressure to keep upfront cost low, reliability high, and refrigerated space maximized. In some segments such as vending machines and beverage merchandisers, split incentives are a formidable barrier. The equipment is often specified and provided for free to the customer by a bottler or vending machine operator who does not pay the energy bill and thus has little incentive to specify high efficiency equipment. In addition, across many segments, there is limited awareness of energy savings potential, although it is growing with the rising cost of energy and the introduction of ENERGY STAR qualified products.

For supermarket chains, energy costs are a substantial portion of operating costs, and they also have access to engineering resources that enable them to evaluate new technologies. Consequently, they are more likely than other CRE customer segments to adopt advanced, high efficiency equipment.

Recommendations
The DOE Building Technologies Program has the opportunity to accelerate the development and adoption of energy saving technologies in a number of ways, including the establishment of new and revised energy efficiency standards, support of demonstration activities for emerging technologies, and support of research and development for advanced technologies. Detailed recommendations are included in section 8 of this report.

1 Introduction

The U.S. Department of Energy (DOE), Office of Energy Efficiency and Renewable Energy (EERE), Building Technologies Program (BT) has sponsored this study of commercial refrigeration equipment (CRE) to:

- Provide an overview of CRE applications
- Characterize equipment types
- Assess the energy savings potential for CRE in the U.S.
- Outline key barriers to widespread adoption of energy-savings technologies
- Summarize the status of regulatory and voluntary efficiency programs
- Recommend initiatives that might help increase energy savings based on currently available technologies
- Recommend opportunities for advanced energy saving technology research.

This report is modeled after the 1996 report, "Energy Savings Potential for Commercial Refrigeration Equipment", by Arthur D. Little, Inc. (ADL 1996). Relative to the 1996 report, this report:

- Updates information
- Examines more equipment types
- Outlines long-term research and development opportunities.

1.1 Report Organization

This report is organized as outlined in Table 2-1.

Table 1-1: Report Organization

Chapter	Content/Purpose
1	Executive Summary
2	Introduction – report objectives, organization, approach, and overview of commercial refrigeration industry
3	Overview of Commercial Refrigeration Equipment Types– CRE applications, equipment types, and market characteristics
4	Description of Baseline Equipment – definition of baseline characteristics for the 7 analyzed equipment types, including equipment description, unit energy use, purchase and installation costs, lifetime/reliability/maintenance, major manufacturers, major end-users
5	Energy Saving Potential Using Current Technologies– calculation of energy savings potential and payback period for the 7 analyzed equipment types; Discussion of technical and market barriers to implementation
6	Advanced Energy Saving Technologies and Tools– discussion of technologies that have not yet been commercialized

7	Impact of Regulatory and Voluntary Efficiency Programs – summary of federal and state energy conservation standards as well as ENERGY STAR and Consortium for Energy Efficiency criteria for the 7 analyzed equipment types.
8	Recommendations
References	---

1.1 Approach

To understand the amount of energy savings possible using more efficient technology in the commercial refrigeration equipment, we evaluated the seven most important types of commercial refrigeration equipment:

- Supermarket refrigeration systems
- Walk-in coolers and freezers
- Food preparation and service equipment
- Reach-in refrigerators and freezers
- Beverage Merchandisers
- Ice machines
- Refrigerated Vending Machines

For each of the seven equipment types, we defined the baseline equipment for making energy-use comparisons. The baseline equipment represents the typical new piece of equipment sold in 2008, which is often more efficient than typical equipment in the field. By using this new equipment baseline, energy-savings estimates exclude the savings that will be achieved with no DOE action, through normal equipment replacement cycles.

We gathered information for this study by:

- Conducting interviews with major CRE manufacturers
- Reviewing product information on manufacturer websites
- Researching the status of relevant high efficiency technologies

To estimate energy savings potentials for each equipment type, we:

1. Estimated the unit energy use for a typical installed unit in 2008
2. Estimated the unit energy use for a typical new unit in 2008
3. Identified all currently commercialized technology options that would increase efficiency
4. Estimated energy use reduction in terms of *system* energy use for each technology option
5. Estimated end-user cost premium, for each technology option[4]
6. Calculated the simple payback period (PBP) for each technology option[5]

[4] See 0 0for a discussion of inflation and markup factors used in order to update costs to $2008 and to reflect the appropriate stage of the distribution chain.
[5] Simple PBP (yrs) = Cost Premium ($) / [Energy Use Reduction (kWh/yr) x Electricity Price ($/kWh)]

7. Calculated the energy use reduction, cost premium, and simple PBP for a realistic combination of technology options deemed to be the maximum energy savings potential technologically feasible with currently commercialized technologies, referred to as “max tech”.

Using the savings percentage that we calculated for each equipment type, we estimate the national energy savings potential by applying it to the entire installed base. In our final estimate of national energy savings, we include all currently commercialized technologies that have a simple payback period of less than 7 years.

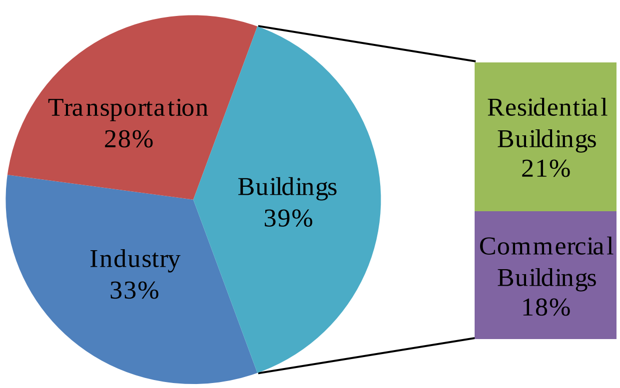
We characterized each advanced technology option in terms of three variables:

- a rough unit energy savings potential (percent)
- a classification of technical risk (high, medium, or low)
- an estimated time to commercialization

We did not estimate costs for advanced technologies given the high level of uncertainty involved.

1.2 Overview of Commercial Refrigeration Industry

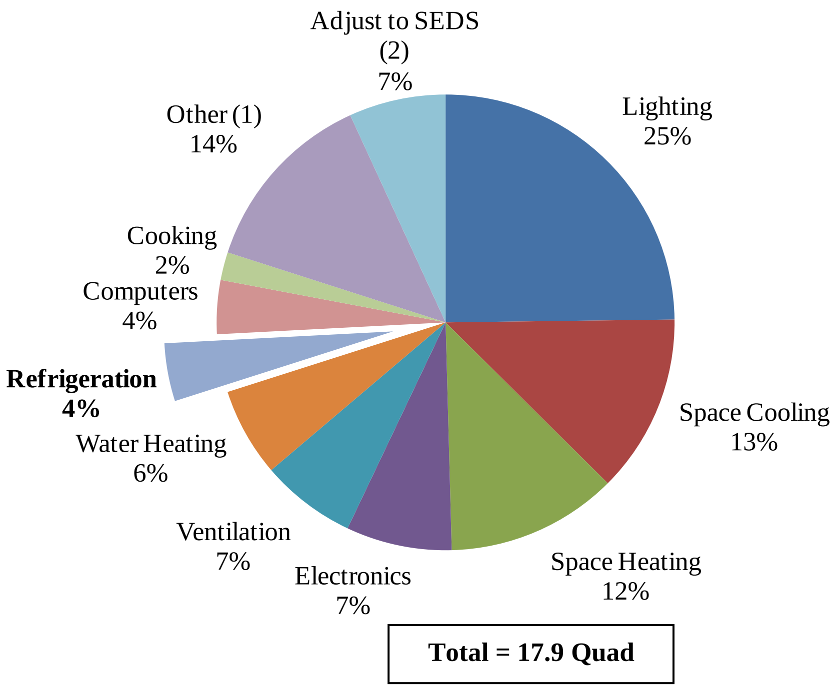
Commercial buildings consume approximately 18% of the total primary energy used in the U.S (**Figure 1-1**). **Figure 1-2** shows the annual primary energy consumption in commercial buildings by end use from the 2008 Buildings Energy Data Book (DOE 2008a). Commercial refrigeration, at 4 percent of the total, follows lighting, space cooling and heating, electronics, ventilation, and water heating in annual primary energy consumption. Estimates from 2 years earlier suggested that commercial refrigeration accounted for over 6% of primary energy usage, so considerable uncertainty exists.



Total Consumption = 99.5 Quad

Source: DOE 2008a

Figure 1-1: U.S. Primary Energy Consumption by Sector, 2006



Source: DOE 2008a

Figure 1-2: Annual US Primary Energy Use of Commercial Buildings by End Use, 2006

Notes:
(1) “Other” includes service station equipment, ATMs, telecommunications equipment, medical equipment, pumps, emergency electric generators, combined heat and power in commercial buildings, and manufacturing performed in commercial buildings.
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While the commercial refrigeration sector includes all types of refrigeration equipment used in commercial buildings, most of this equipment is associated with food preservation. Food preservation has become essential to our wellbeing as it gives us affordable year-round access to a variety of fresh food and drink. Food preservation is needed for both food sales, (primarily supermarkets, convenience stores), and food service (primarily restaurants and cafeterias). Remote vending of refrigerated beverages and food has also developed into an important application of commercial refrigeration. In addition, ice machines and drinking-water coolers are widely used.

A number of different types of equipment are used in the various applications of commercial refrigeration, depending on the intended purpose of the equipment. This report considers the following seven equipment types to be representative of the commercial refrigeration equipment:

- Supermarket refrigeration systems (consisting of display cases and walk-in refrigerators and freezers using remote compressor racks and condensers)
- Walk-in coolers and freezers
- Food preparation and service equipment

- Reach-in refrigerators and freezers
- Beverage Merchandisers
- Ice Machines (excluding ice dispensers)
- Refrigerated Vending Machines

The following equipment types, even though they fall within the category of CRE, were not evaluated in this report due to their small contribution to overall energy use:

- Water coolers
- Milk coolers and dispensers
- Ice Dispensers
- Cold-plate counter tops

Further, while they may consume significant amounts of energy, refrigeration systems used in distribution warehouses were beyond the scope of this study.

As detailed in this report, we estimate that primary energy consumption in the commercial refrigeration sector is approximately 1.23 Quadrillion Btu per year (Quad), slightly over the Buildings Energy Data Book estimate for 2006. **Table 1-2** compares the estimates of annual CRE energy consumption from this report, two recent DOE Buildings Energy Data Books, and the 1996 ADL report.

Table 1-2: Comparison of Annual CRE Energy Consumption Estimates

Source	Year of Data	Annual CRE Energy Consumption (Quad)	% of Total Commercial Building Energy Use
This Report	2008	1.23	N/A [1]
2008 Buildings Energy Data Book (DOE 2008a)	2006	0.73	4.1%
2006 Buildings Energy Data Book (DOE 2006)	2004	1.10	6.3%
ADL 1996	1996	0.99	7.0%
[1] Data regarding commercial building energy usage in 2008 has not yet been published. 1.23 quads would amount to 7.1 % of 2006 commercial building energy consumption			

The breakdown by the seven analyzed equipment types is shown in **Figure ES-2**. Supermarket refrigeration systems use 55 percent of the total energy used by commercial refrigeration. Walk-in coolers and freezers, which are typically used for food storage in the food service industry, are the next largest energy-consuming category with 13 percent.

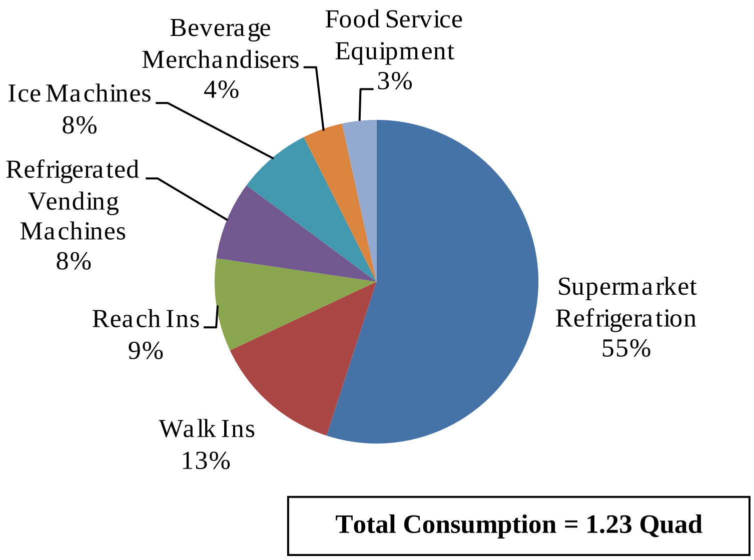


Figure 1-3: Annual Primary Energy Usage of Commercial Refrigeration by Equipment Type

2 Overview of Commercial Refrigeration Equipment Types

A description of the market for each equipment type, including the estimated installed base and overall energy consumption, are provided below.

Table 2-1 is a summary of the total installed base and annual energy consumption by equipment type for 2008. The details of these estimates are provided in the remainder of this report.

Table 2-1: 2008 Installed Base and Total Energy Consumption by Equipment Type

<table>
<tr><th colspan="2">Equipment</th><th>Installed Base (units)</th><th>Total Primary Energy Consumption (TWh/year)</th></tr>
<tr><td rowspan="4">Supermarket Refrigeration Systems</td><td>Display Cases</td><td>2,100,000</td><td>214</td></tr>
<tr><td>Compressor Racks</td><td>140,000</td><td>373</td></tr>
<tr><td>Condensers</td><td>140,000</td><td>50</td></tr>
<tr><td>Walk-ins</td><td>245,000</td><td>51</td></tr>
<tr><td colspan="2">Walk-in Coolers and Freezers (Non-Supermarket)</td><td>755,000</td><td>148</td></tr>
<tr><td colspan="2">Food Preparation and Service Equipment</td><td>1,516,000</td><td>55</td></tr>
<tr><td colspan="2">Reach-in Refrigerators and Freezers</td><td>2,712,000</td><td>106</td></tr>
<tr><td colspan="2">Beverage Merchandisers</td><td>920,000</td><td>45</td></tr>
<tr><td colspan="2">Ice Machines</td><td>1,491,000</td><td>84</td></tr>
<tr><td colspan="2">Refrigerated Vending Machines</td><td>3,816,000</td><td>100</td></tr>
<tr><td colspan="2">Total</td><td></td><td>1,225</td></tr>
<tr><td colspan="4">Note: Energy consumption values have been rounded to the nearest whole number, and therefore the total does not exactly equal the sum of the energy consumption values for each equipment type.</td></tr>
</table>

Commercial refrigeration equipment can be classified into two categories: split-system refrigeration systems and self-contained refrigeration systems. Split-system configurations have a condenser unit that is located remotely, usually on the rooftop, which allows it to exchange heat with the outside air. Split-systems are discussed in the supermarket refrigeration system section. Self-contained units have all of the components, including the condenser, contained in a single package. The remaining six equipment types in this report are self-contained. **Figure 2-1** shows the components in a self-contained refrigeration system. Refer to **Figure 3-2** for a diagram of a split-system refrigeration system.

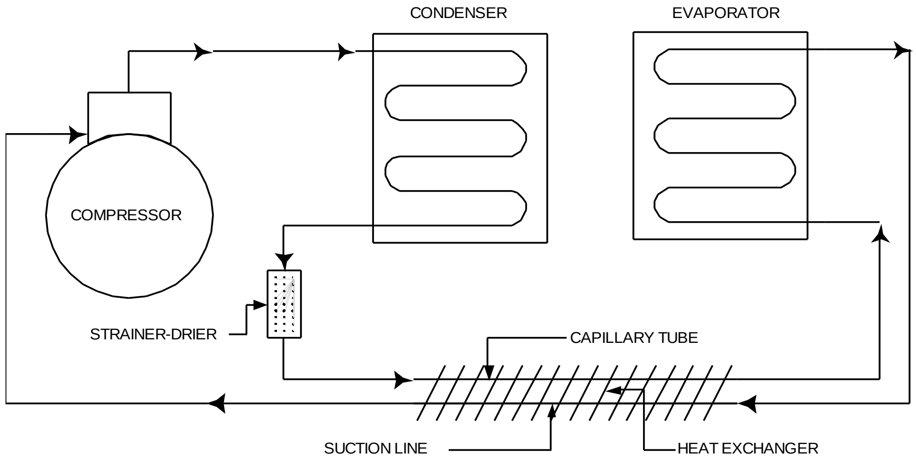


Figure 2-1: Self-Contained Refrigeration Circuit

2.1 Supermarket Refrigeration Systems

There are approximately 35,000 supermarkets in the United States (Progressive Grocer 2008). Food sales from supermarkets represent roughly $535 billion annually, or 56% of the overall food sales market (Progressive Grocer 2008). Supermarkets are distinguished from smaller grocery stores by having revenues exceeding two million dollars, according to Progressive Grocer Magazine.

Table 2-2 shows the allocation of food sales and the number of stores for various store types. In recent years, the trend for supermarkets has been towards smaller numbers of larger stores. Convenience stores are full-line, self-service grocery stores which are open long hours and offer a limited line of high-convenience items. Military convenience stores tend to be larger than civilian stores, and are listed separately in the table.

Table 2-2: Number of Stores and Average Sales in the Grocery Industry as of 2007

Store Type	Number of Stores (1,000's)	US Annual Sales ($ Billions)
Supermarket	35.0	535.4
Convenience	145.9	306.6
Grocery (<$2 million)	13.7	18.2
Wholesale Clubs	1.2	101.5
Military Convenience Stores	0.4	2.2
Total	**196.2**	**963.9**

Source: Progressive Grocer 2008

Supermarket complexity has increased in recent years, as supermarkets have increased their sales in a variety of specialty areas such as deli and bakery. Many supermarkets sell a variety of non-food products, such as personal hygiene, paper products, cleaning products, flowers, etc. Supermarkets are also shifting into food service (i.e., prepared meals), which represents one of the largest growing food service sectors.

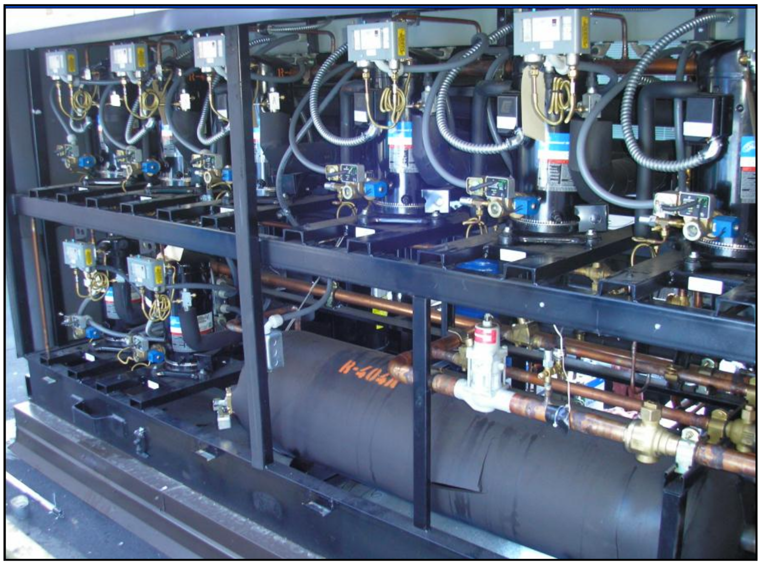
To quantify supermarket refrigeration energy use, we consider the energy use of display cases (which contain the refrigerated merchandise), compressor racks (which are the sets of compressors that run the refrigeration system), condensers, and walk-ins.

A typical supermarket has about 60 display cases to display the fresh and frozen food products throughout the store. **Table 2-3** shows the shipment data for display cases between 1999 and 2008. We use this shipment data to estimate the installed base of display cases to be approximately 2,100,000 cases, assuming an average lifetime of 10 years.

Table 2-3: Display Case Shipments (1999-2008)

Year	Shipments
1999	340,453
2000	347,262
2001	175,000
2002	183,300
2003	191,549
2004	185,000
2005	170,000
2006	175,500**
2007*	181,000
2008*	185,000
Total	**2,134,064**
* Statistical Forecast from Appliance Magazine "54rd Appliance Industry Forecasts," (CRE 2009a) **Data not available, estimated to be the average of 2005 of 2007 data.	

Compressor racks are configurations of paralleled-connected compressors located in machinery rooms, predominately found in supermarkets (**Figure 2-2**). Typically, a supermarket will have 10 to 20 compressors mounted in racks (3 to 5 compressors per rack).



Source: Zero Zone 2009

Figure 2-2: Supermarket Refrigeration System Compressor Rack

Annual shipments of compressor racks are estimated to be 15,400 racks, assuming an installed base of 140,000 racks (four per supermarket), a replacement rate of 10 percent per year, and a supermarket growth rate of 1 percent per year.

Condensers used in supermarkets are usually rooftop air-cooled condensers. We estimate that the typical supermarket configuration includes one condenser per refrigeration circuit, which would total four condensers per supermarket.

Walk-in coolers and freezers are used for storage of fresh and frozen food. A supermarket has multiple walk-ins located in various locations outside of the sales area. The walk-in cases are kept at different temperatures depending on the contents. Walk-in coolers with merchandising doors are used where applicable (typically for milk and juice). The number and size of walk-ins varies depending on store layout, but we estimate that the typical supermarket has seven separate walk-ins incorporated into its central refrigeration system. **Table 2-4** shows the breakdown of walk-in units in a supermarket by floor area, number of units, and items stored.

Table 2-4: Supermarket Walk-In Unit Types

Walk-In Type	Total Area (ft2)[1]	Estimated # of Units / store [2]	Items Stored
Meat Coolers	400	1	Meat
Other Coolers	2600	4	Produce, Dairy, Deli, Other
Freezers	1000	2	Frozen Food
Total	4000	7	
Sources:			

[1] ADL 1996
[2] NCI Estimate

The following section discusses walk-ins with dedicated refrigeration systems, which is a configuration generally found in non-supermarket applications.

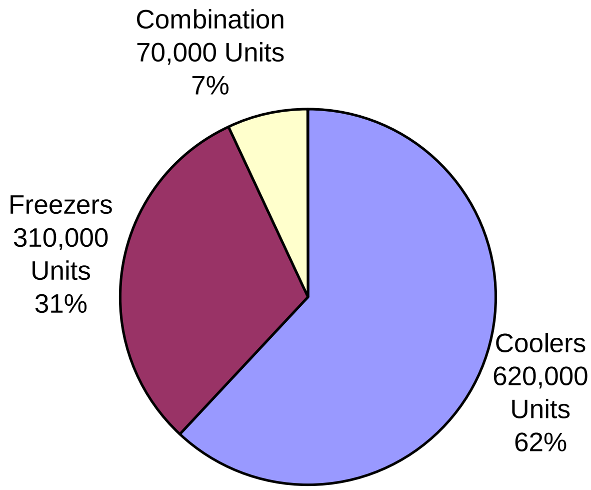
2.2 Walk-in Coolers and Freezers

Walk-in coolers and freezers, also known as walk-ins, are large, insulated refrigerated spaces with access doors large enough for people to enter. Walk-ins are used for food storage and merchandising in the food service and food sales applications. Walk-ins found in supermarkets are generally cooled by central supermarket refrigeration systems and are discussed within the supermarket refrigeration system section. In the following section, we discuss walk-ins with dedicated refrigeration systems.

There are two major classes of walk-ins: low refrigerated space temperature (-20 to -10°F) and medium refrigerated space temperature (10 to 35°F). Although walk-ins can be used in a wide variety of areas, they are primarily used in food service and food sales (Freedonia 2004). Examples of non-food applications are blood and flower storage, but these sectors account for a small proportion of walk-ins. A list of establishments that use walk-ins includes:

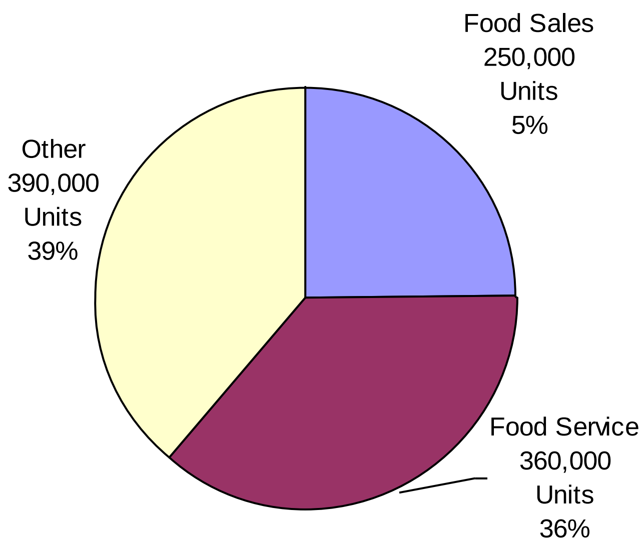
- Restaurants and Bars
- Convenience Stores
- Cafeterias
- Florists
- Research Laboratories

Assuming all walk-ins are used in commercial applications, there is an installed base of 1,000,000 walk-ins based on inventory estimates from major walk-in manufacturers. Of this total installed base, roughly 245,000 are assumed to be part of supermarket refrigeration systems (seven per supermarket), and the remaining 755,000 are assumed to be installed in non-supermarket applications with dedicated refrigeration systems. Annual walk-in sales are estimated to be 40,000 units per year (Caroll Coolers 2008, Freedonia 2004). The market value for sales in 2008 is estimated to be $800 million (Freedonia 2004). Figure 2-3 and Figure 2-4 show the estimated installed base broken out by temperature level and end user category, respectively.



Sources: Major Manufacturers, ADL 1996

Figure 2-3: Walk-ins by Temperature Level (2008) – 1,000,000 units total



Sources: CBECS 2003, Major Manufacturers

Figure 2-4: Walk-ins by End Use Applications (2008) – 1,000,000 units total

Based on discussions with manufacturers, walk-ins are estimated to consume approximately 19.1 TWh annually, which represents approximately 199 trillion Btu primary energy (see Table 3-3).

Table 2-5: Commercial Sector Overview - Walk-in Coolers and Freezers

Application	Unit Type	Estimated Inventory	Unit Energy Consumption (kWh/yr)[1]	Total Energy Consumption (TWh/yr)	Total Energy Consumption (%)	Primary Energy Consumption (TBtu/yr)[3]
Non-Supermarket	Cooler	468,100	16,200	7.6	40%	78.9
	Freezer	234,050	21,400	5.0	26%	52.1
	Combination	52,850	30,200	1.6	8%	16.6
Supermarket		245,000	varies [2]	4.9	26%	51.0
Total		1,000,000	-	19.1	100%	198.6

[1] ADL 1996. Unit energy consumption (UEC) that is typical of the current installed base; Includes compressors, evaporator and condenser fans, lighting, defrost, and anti-sweat for Non-Supermarket units; Includes evaporator fan, lighting, defrost, and anti-sweat for supermarket applications.
[2] Energy use values are based on the baseline supermarket walk-in cooler and walk-in freezer presented in section 3.1.2.
[3] 10,405 Btu primary energy/kWh site energy (DOE 2008a: Table 6.2.4)

2.3 Refrigerated Food Service Equipment

Refrigerated food service equipment is used to provide refrigeration and storage to maintain food products prior to and while serving them in foodservice settings. Such equipment is predominantly found in restaurants, hotels, convenience stores, supermarkets, schools, and other facilities where food is served. Major types of equipment, which are illustrated in Figure 3-5, include:

- Preparation Table – a commercial refrigerator with a countertop refrigerated compartment with or without cabinets below
- Worktop Table – a counter-height commercial refrigerator or freezer with a worktop surface.
- Buffet Table - a commercial refrigerator, such as a salad bar, that is intended to receive refrigerated food, to maintain food product temperatures, and to provide customer service.

a) Preparation Table **b) Worktop Table** **c) Buffet Table**

Sources: McCall Refrigeration (a, b), Electrolux Refrigeration (c)

Figure 2-5: Refrigerated Food Service Table Types

As shown in Table 3-6, the installed base of refrigerated food service equipment is estimated to be approximately 1,516,000 units. The unit energy consumption is estimated using the models available in the CEC appliance database (**Table 2-7**). The national energy consumption of food service equipment is 5.27 TWh/yr, shown in **Table 2-8**.

Table 2-6: Food Service Equipment Installed Base Data

Building Type	FSE / Building [1]	# of Food Service Buildings, 2003[2]	Installed Base, 2003	Annual FSE Shipments [1]	Installed Base, 2008
Foodservice	3	297,000	891,000	125,000	1,516,000
Sources: [1] NCI Estimate [2] CBECS 2003					

Table 2-7: Food Service Equipment Unit Energy Consumption

Unit Type	Number of Units Listed in CEC Database	% of Units Listed in CEC Database	Average Volume (ft^3)	Average Unit Energy Consumption (kWh/yr)[1]
Prep Table	219	62%	16	3,927
Buffet Table	44	13%	3.9	1,994
Worktop Table	89	25%	13.5	1,857
Weighted Average UEC (Typical New):				**3,162**
Weighted Average UEC (Typical Installed):				**3,478**
Source: CEC 2008. [1] We assume that the CEC database represents the models available on the market, and therefore the average value represents the energy use of a typical				

new model. We assume that the typical installed UEC is 10% greater than the typical new.

Table 2-8: Food Service Equipment Energy Consumption Summary

Unit Type	Estimated Installed Base[1]	Typical Unit Energy Consumption (kWh/yr) [1]	Total Energy Consumption (TWh/yr)	Primary Energy Consumption (TBtu/yr)[2]
Food Service equipment	1,516,000	3,478	5.27	55

Sources:
[1] Unit energy consumption (UEC) that is typical of the current installed base; average of prep table, worktop table, and buffet table average UECs currently available on the market (CEC 2008).
[2] 10,405 Btu primary energy/kWh site energy (DOE 2008a: Table 6.2.4)

2.4 Reach-ins

Commercial reach-in cabinets are upright, self-contained refrigerated cases with solid or glass doors whose purpose is to hold frozen and/or refrigerated food products. These cases are commonly used in commercial and institutional food-service establishments. These are self-contained units, i.e., the entire refrigeration system is built into the reach-in unit and heat is rejected to the surrounding interior air.

In this study, "reach-in" collectively refers to three types of commercial refrigeration equipment, namely reach-in freezers, reach-in refrigerators, and reach-in refrigerator-freezers. Each equipment category is described and analyzed separately below.

This study does not include descriptions of pass-through, roll-in, and roll-through cabinets, since these equipment categories account for less than 20% of the overall reach-in refrigerator market and less than 15% of the overall reach-in freezer market, based on the number of models of each type listed in the California Energy Commission Appliances Database (CEC 2008). However, the estimates for reach-in installed base includes reach-in cabinets of all types, and the energy consumption differences between the cabinet configurations are deemed to be minor enough to approximate all cabinet types as basic reach-in cabinets (as opposed to pass-through, roll-in, and roll-throughs).

The reach-in refrigerator-freezer, sometimes referred to as a "dual-temp", combines one or more refrigerator compartments and one or more freezer compartments into a single unit. A description of the typical dual-temp unit is provided below. However, because dual-temps make up a relatively small segment with limited shipment data and all of the relevant technologies are covered by the refrigerator and freezer categories, dual-temps do not have separate energy-savings and economic analyses in this report.

Table 2-9 and **Table 2-10** provide an overview of the reach-in market size for freezers and refrigerators, respectively, based on commercial refrigeration by building type as reported by the Commercial Building Energy Consumption Survey since 1995 (CBECS 1995, 1999, and 2003). The three most recent editions of CBECS were used to estimate the 2008 installed bases for reach-in freezers and refrigerators, using a linear trend line, as shown in **Figure 2-6**. Table 2-11 summarizes the annual energy consumption of the total reach-in installed base.

Table 2-9: Reach-in Freezer Installed Base Data

Building Type	RIF / Building[1]	2003		1999		1995	
		Buildings w/ CRE (1,000)[2]	Installed Base (1,000s)	Buildings w/ CRE (1,000)[3]	Installed Base (1,000s)	Buildings w/ CRE (1,000)[4]	Installed Base (1,000s)
Education	2	93	186	84	168	97	194
Food Sales	1	205	205	159	159	136	136
Food Service	1	283	283	334	334	272	272
Health Care	2	23	46	12	24	15	30
Lodging	1	42	42	30	30	36	36
Retail	3	98	294	97	291	101	304
Office	1	53	53	26	26	26	26
Total – 2003		**797**	**1,109**	**742**	**1,032**	**683**	**998**

Sources:
[1] Estimated by NCI confirmed through communication with a major reach-in manufacturer in May 2008.
[2] CBECS 2003
[3] CBECS 1999
[4] CBECS 1995

Table 2-10: Reach-in Refrigerator Installed Base Data

Building Type	RIR / Building[1]	2003		1999		1995	
		Building s w/ CRE (1,000)[2]	Installed Base (1,000s)	Building s w/ CRE (1,000)[3]	Building s w/ CRE (1,000)[2]	Installed Base (1,000s)	Building s w/ CRE (1,000)[3]
Education	2	93	186	84	168	97	194
Food Sales	1	205	205	159	159	136	136
Food Service	2	283	566	334	668	272	544
Health Care	3	23	69	12	36	15	45
Lodging	1	42	42	30	30	36	36
Retail	3	98	294	97	291	101	304
Office	2	53	106	26	52	26	52
Total – 2003		**797**	**1,468**	**742**	**1,404**	**683**	**1,311**

Sources:
[1] Estimated by NCI confirmed through communication with a major reach-in manufacturer in May 2008.
[2] CBECS 2003
[3] CBECS 1999
[4] CBECS 1995

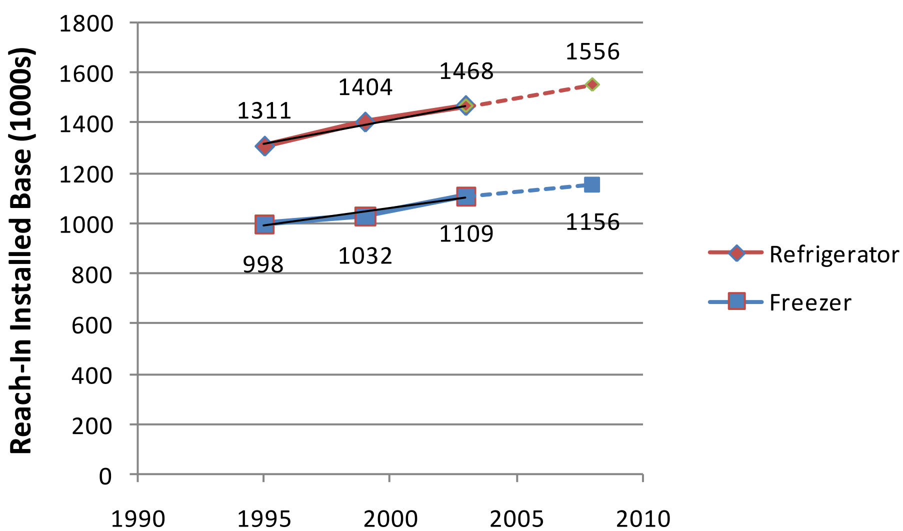


Figure 2-6: Reach-In Installed Base Trends (1995-2008)

Table 2-11: Reach-In Energy Consumption Summary

	Annual Shipments 2006 (units)[1]	2008 Installed Base (units)	Typical Unit Energy Consumption (kWh/yr)[2]	Total Energy Consumption (TWh/yr)[3]	Primary Energy Consumption (TBtu/yr)[4]
Freezers	52,000	1,156,000	4,158	4.8	56
Refrigerators	263,000	1,556,000	3,455	5.4	50

Sources:
[1]Appliance Magazine 2008; NCI estimates that 85% of the listed 309,375 "commercial refrigerators" refer to reach-in refrigerators.
[2] Unit energy consumption (UEC) that is typical of the current installed base; 20 percent reduction in energy use from ADL 1996.
[3]Adapted from CEC Appliances Database using an average of unit energy consumption values for each manufacturer, weighted by market share. See section 3.4.2 for a more detailed explanation.
[4] 10,405 Btu primary energy/kWh site energy (DOE 2008a: Table 6.2.4)

Reach-in Freezers
Reach-in freezers are upright, refrigerated cases whose purpose is to hold frozen food products. As shown in Table 2-11, the 2006 annual sales are estimated to be about 52,000 units according to Appliance Magazine (2008). Based on the trend in recent CBECS (CBECS 1995, 1999, and 2003), there is an estimated installed base of 1,156,000 reach-in freezers. Approximately 55% are one-door units (ADL 1996).

The one-door unit was chosen as the representative reach-in freezer since it is the most common unit currently used. Based on an estimated inventory of 1,156,000 units, reach-in freezers consume roughly 4.8 TWh annually.

Reach-in Refrigerators
Reach-in refrigerators are upright, refrigerated cases whose purpose is to hold refrigerated food products. Annual shipments are estimated to be about 263,000 units (Table 2-11) (Appliance Magazine 2008). Based on the trend in recent CBECS (CBECS 1995, 1999, and 2003), there is an estimated installed base of 1,556,000 reach-in refrigerators. Approximately 65% are two-door units. (ADL 1996)

The two-door unit was chosen as the prototypical reach-in refrigerator since it is the most common unit currently used. Based on the estimated inventory of 1,556,000 units, reach-in refrigerators consume roughly 5.4 TWh annually.

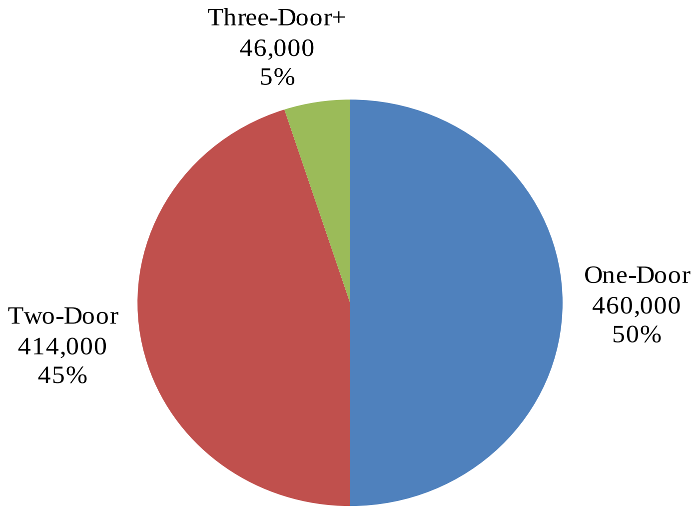
2.5 Beverage Merchandisers

Beverage merchandisers are self-contained, upright, refrigerated cabinets that are designed to hold and/or display refrigerated beverage items for purchase without an automatic vending feature. Typically they have glass doors and bright lighting. These cases are commonly used in convenience stores, aisle locations in supermarkets, and some retail stores and small foodservice establishments. Because the refrigeration system is self-contained, the heat is rejected to the building interior, and their energy use is not included in the supermarket refrigeration section above.

The majority of beverage merchandisers are owned by bottling and vending companies, such as Coca-Cola and Pepsi. Beverage merchandisers are also sold directly to retailers. Bottling and vending companies place the merchandisers in retail locations such as convenience stores and supermarkets and are responsible for delivering the beverages to the site, filling the merchandisers, and maintaining and servicing the merchandiser. The retail operator is responsible for paying energy costs in both cases.

Supermarkets and convenience stores use centralized systems to refrigerate most of their display cases. However, beverage merchandisers use a self-contained refrigeration system and have the ability to be located in areas to maximize sales. They are easy to install in locations away from central refrigeration equipment and easy to relocate. For example, in supermarkets beverage merchandisers are put near locations such as the checkout lane to lure customers into an “impulse buy”.

Beverage merchandiser annual sales are estimated to be approximately 80,000, or 10 percent of the installed base (NCI estimate). According to a leading manufacturer’s estimate[6], there is an installed base of between 750,000 and 1,000,000 beverage merchandisers. Approximately 50% are one-door units. Figure 2-7 shows the beverage merchandiser installed-base breakdown for the commercial sector.



Source: Adapted from ADL 1996

Figure 2-7: 2008 Estimated Installed Base of Beverage Merchandisers.

Note: Total estimated 2008 inventory - 920,000 units; Assumes 15% growth since 1996.

[6] From conversation between Daniel Pinault of NCI and leading beverage merchandiser manufacturer in May 2008.

Table 2-12: Beverage Merchandiser – 2008 Equipment Installed Base

Unit Type	Estimated Installed Base[1]	Unit Energy Consumption (kWh/yr) [2]	Total Energy Consumption (TWh/yr)	Total Energy Consumption (%)	Primary Energy Consumption (TBtu/yr)[3]
One-Door	460,000	3,076	1.41	33%	14.7
Two-Door	414,000	6,080	2.52	58%	26.2
Three-Door+	46,000	8,960	0.41	9%	4.3
Totals	**920,000**		**4.34**	**100%**	**45.2**

Sources:
[1] ADL 1996, adjusted for 15% total growth between 1996 and 2008. NCI considers 15% to be a reasonable estimate based on the growth rates of other sectors: supermarket stores grew 16% from 30,000 to 35,000 stores (Progressive Grocer 1995, 2008) and reach-in refrigerators are estimated to have grown by 19% (Figure 2-6, CBECS 1995, 2008).
[2] Unit energy consumption (UEC) that is typical of the current installed base; 20 percent reduction in energy use from ADL 1996.
[3] 10,405 Btu primary energy/kWh site energy (DOE 2008a: Table 6.2.4)

2.6 Ice Machines

Ice machines are used to produce a variety of ice types used in the food service, food preservation, hotel, and healthcare industries. Ice machines are classified into three primary equipment types depending on whether the ice-making mechanism and the condensing unit are contained in a single package and whether the unit has an integrated storage bin. The three types are summarized in Table 2-13 below. See Figure 3-8 for illustrations. Self-contained units make up one third of the shipments, and the remaining two thirds are comprised of ice-making heads and remote condensing units.

Table 2-13: Ice Machine Equipment Types

	Configuration of Condensing Unit	Configuration of Storage Bin
Ice Making Head (IMH)	Integrated	Separate
Self-Contained Unit (SCU)	Integrated	Integrated
Remote-Condensing Unit (RCU)[7]	Separate	Separate

[7] Regulatory programs further divide this category into remote-condensing units *with* remote compressors and remote-condensing units *without* remote compressors.

a) Ice Making Head on an Insulated ice storage bin **b) Self-Contained Unit** **c) Remote Condenser**

Source: Ice-O-Matic

Figure 2-8: Ice Machine Equipment Types

The types of ice produced, as shown in **Figure 2-9**, include:

- Cube ice -distinct portions of fairly uniform, hard, solid, usually clear ice, generally weighing less than two ounces (60 grams) per piece
- Flake ice - chips or flakes of ice containing up to 20 percent liquid water by weight used primarily for temporary food preservation (e.g., supermarket display cases, fishing boats) and occasionally for soft drinks
- Nugget ice - small chewable, nugget-shaped portions of ice created by compressing the slushy ice/water mixture of flake ice into a nugget; used primarily for keeping drinks cool

a. Cube Ice **b. Flake Ice** **c. Nugget Ice**

Sources: Ice-O-Matic (a), Hoshizaki America (b), Food Service Warehouse (c)

Figure 2-9: Ice Types

Ice cube machines are predominantly found in restaurants, hotels, convenience stores, schools, and other facilities where food is served, such as stadiums, convention centers, and office buildings. They are typically located indoors (e.g., kitchen area, hotel vending room) and occasionally located outdoors (e.g., walkway of resort hotels). Cube weights range from about 1/6 - 1/2 oz., with about 70 percent of sales in the 1/6 - 1/4 oz. range.

Cube shapes include cubic, rectangular, crescent, lenticular, and pillow. The cube shape is usually unique to a particular manufacturer, and thus is used to distinguish one manufacturer from another. The maximum cube dimension is about 1 1/4", depending on the cube shape. Desirable ice cube characteristics include minimal liquid content, smooth ice to minimize carbonation loss in soft drinks, high displacement to minimize the drink serving amount, slow melting to reduce drink dilution, and clarity.

Flake ice machines are used primarily for food preservation in the food sales industry. They are often made for high ice-production capacity needs. Fish displays and salad bars are common grocery applications for flake ice.

Nugget ice has become increasingly popular as drink ice, because it is more chewable than traditional cubes. Each major ice machine manufacturer has now developed its own brand of nugget ice, including Chewblet, Cubelet, Chunklet, and Pearl Ice (see Table 3-47). Between 2003 and 2006, sales of nugget ice machines rose 23% to 16,673 units (WSJ 2008).

There are two distinct ice-making processes governing the ice-types produced:

- The *batch* process, which involves alternate freezing and harvesting periods, is used to make cube ice. Water flows over an evaporator where it freezes until cubes are fully formed. The ice cubes are then harvested and moved to storage. The ice may be in cube shape, or in a variation of a solid shape.
- The *continuous* process is used to make flake and nugget ice, usually in a barrel-shaped evaporator. Ice flakes are either a) formed on the inside of a stationary evaporator and scraped off by a rotating auger, or b) formed on the outside of the rotating evaporator and scraped off by a stationary scraper. Nugget machines compress the ice flakes to form nuggets. (ARI 2007, ENERGY STAR 2008)

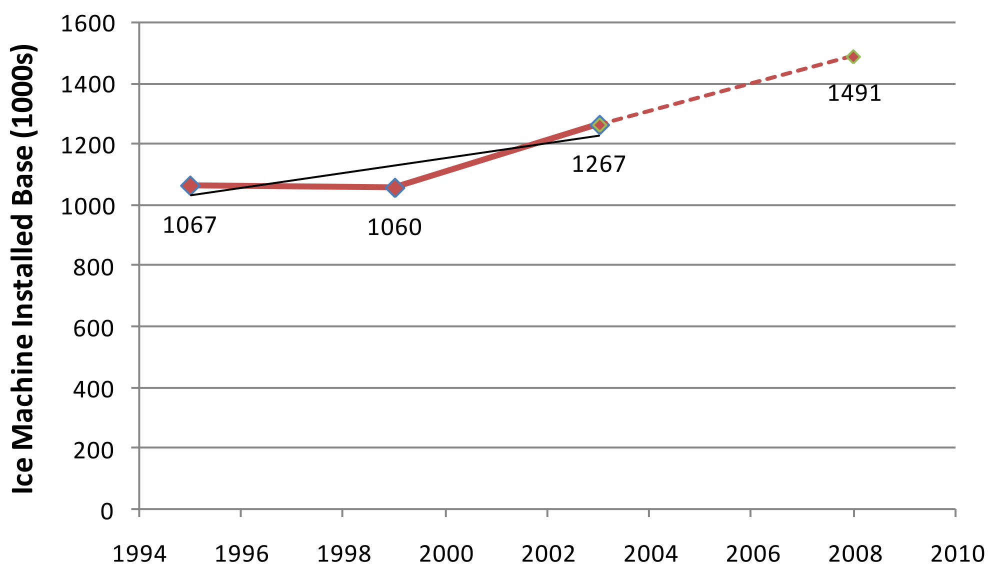
Machines are referred to by their nominal capacity, or harvest rate, defined as the weight of ice produced per 24-hour period. Nominal capacities refer to operation in an ambient temperature of 90°F, inlet water temperature of 70°F, and inlet water pressure of 30 ± 3 psig (ARI 2007). Commercial ice machine capacities range from 50 lbs/24 hrs to 2500 lbs/24 hrs.

Table 2-14 provides an overview of the ice machine market size for freezers and refrigerators, respectively, based on commercial refrigeration by building type as reported by the Commercial Building Energy Consumption Survey since 1995 (CBECS 1995, 1999, and 2003). The three most recent editions of CBECS were used to estimate the 2008 installed bases for ice machines, using a linear trend line, as shown in **Figure 2-10**. Table 2-11 summarizes the annual energy consumption of the total ice machine installed base.

Table 2-14: Ice Machine Installed Base Data

Building Type	Ice Machine per Building[1]	2003		1999		1995	
		Buildings w/ CRE (1,000)[2]	Installed Base (1,000s)	Buildings w/ CRE (1,000)[3]	Buildings w/ CRE (1,000)[2]	Installed Base (1,000s)	Buildings w/ CRE (1,000)[3]
Education	2	93	93	84	84	97	97
Food Sales	1	205	205	159	159	136	136
Food Service	1	283	283	334	334	272	272
Health Care	2	23	115	12	60	15	75
Lodging	1	42	420	30	300	36	360
Retail	3	98	98	97	97	101	101
Office	1	53	53	26	26	26	26
Total – 2003		**797**	**1,267**	**742**	**1,060**	**683**	**1,067**

Sources:
[1] Estimated by NCI confirmed through communication with a major ice machine manufacturer in May 2008.
[2] CBECS 2003
[3] CBECS 1999
[4] CBECS 1995



Source: CBECS 1995, 1999, and 2003

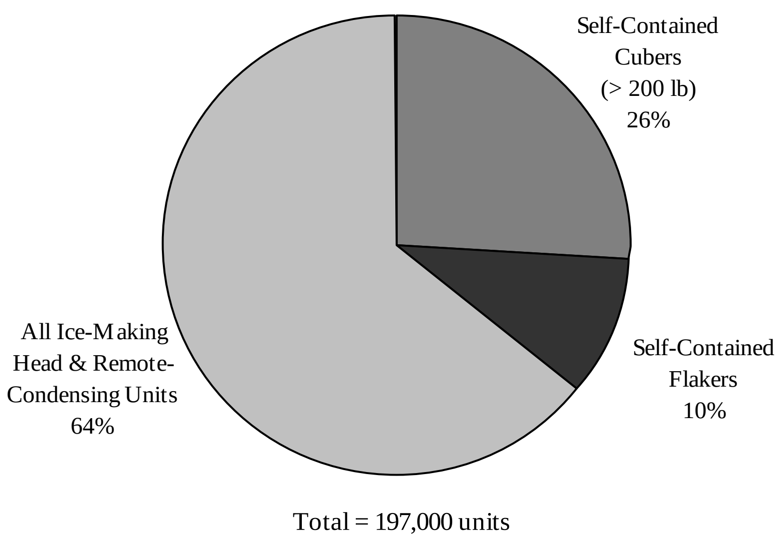
Figure 2-10: Ice Machine Installed Base Trend (1995-2008)

Table 2-15: Ice Machine Energy Consumption Summary

	Annual Shipments 2006 (units)[1]	2008 Installed Base (units)	Unit Energy Consumption (kWh/yr)[2]	Total Energy Consumption (TWh/yr)[3]	Primary Energy Consumption (TBtu/yr)[4]
Ice Machines	197,000	1,491,000	5,429	8.1	84.2

Sources:
[1] Appliance Magazine 2007
[2] Unit energy consumption (UEC) that is typical of the current installed base; Assumed to have 20 percent higher energy use than the typical new ice machine on the market in 2008 (5250 kWh/yr). See section 3.6.2 for a more detailed explanation.
[3]Assumes 50% system duty cycle per year, which calculates to 5270 kWh/year per unit.
[4] 10,405 Btu primary energy/kWh site energy (DOE 2008a: Table 6.2.4)

Total shipments of commercial ice machines in 2006 were approximately 197,000 units. Annual ice machine shipments have averaged 191,000 units over the period between 1998 and 2006, according to Appliance Magazine (2007). Figure 2-11 summarizes the shipments by equipment type.



Source: DoC 2007

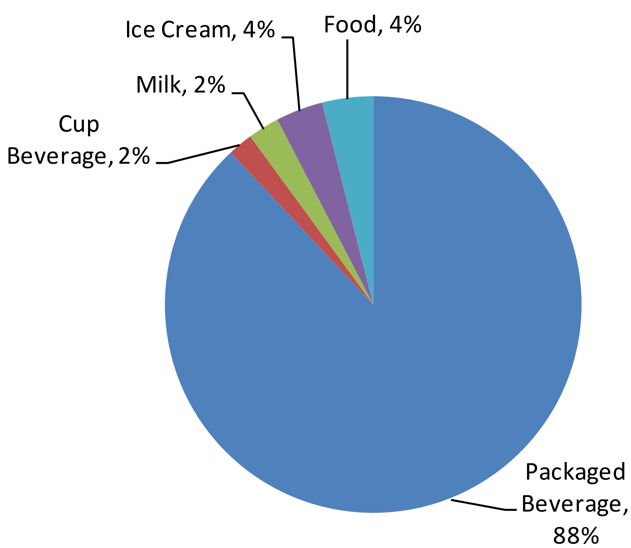
Figure 2-11: Commercial Ice Machine Shipments in 2006 by Product Type

Note: Self-contained cube machines with a capacity less than 200 lbs/24 hours have been excluded from the graph above, because they are predominantly sold in the residential market. Total Shipments amounted to 194,637 units, according to the Department of Commerce. Appliance Magazine reports total shipments to be 197,880 units. Assume nugget machines are included in the flaker segment and the IMH/RC category includes all three types.

2.7 Refrigerated Vending Machines

Refrigerated vending machines are upright, refrigerated cases whose purpose is to hold cold beverages and/or food products and vend them in exchange for currency or tokens. The most common locations are: offices and office complexes; public locations; plants and factories; hospitals and nursing homes; colleges and universities; primary and secondary schools; government and military buildings (Vending Times 2008a). The refrigerated vending machine is self-contained, i.e., the entire refrigeration system is built into the machine and heat is rejected to the surrounding air.

There is an estimated installed base of about 3.8 million refrigerated vending machines. Refrigerated packaged beverage vending machines account for almost 90% of the installed base. The packaged beverage vending machine was chosen for analysis in this report since it is the most common unit. There are approximately 342,000 packaged beverage vending machines shipped per year (DOE 2008b). It is expected that its energy consumption characteristics will be similar to those of other types of refrigerated vending machines. Figure 2-12 shows the refrigerated vending machine inventory breakdown for the commercial sector.

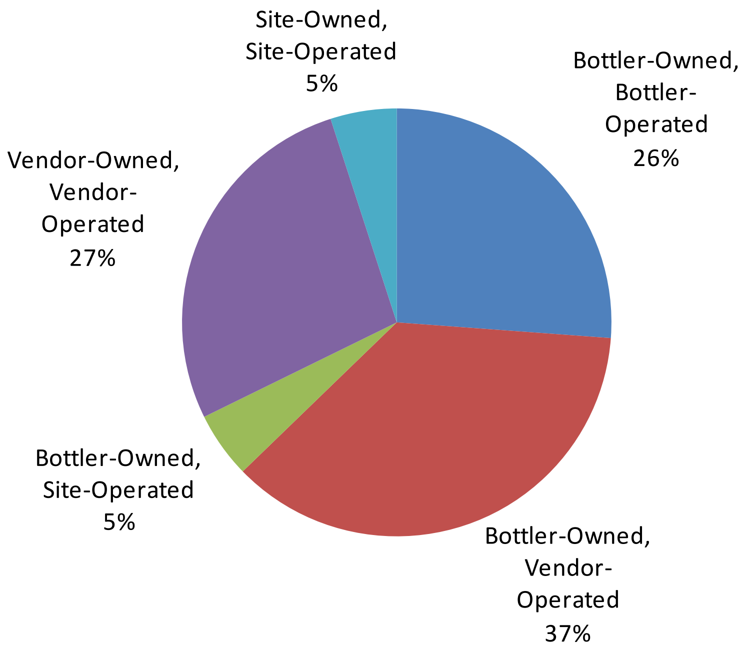


Total Estimated 2007 Installed Base: 3,816,000 Units

Source: Vending Times 2008a

Figure 2-12: Refrigerated Vending Machines – 2007 Equipment Inventory

About 70% of all beverage vending machines are purchased directly from the manufacturer by bottling companies (i.e., Coca-Cola, Pepsi, etc.) (Automatic Merchandiser 2007). Some of these units are supplied to independent vending operators on consignment while the remaining units are owned and operated by the bottlers themselves (Figure 2-13). The other 30% of beverage vending machines are purchased by owner/operators. These include "Mom & Pop" stores, canteens, foodservice operators and vending operators (i.e. American Vending).



Total Estimated 2007 Inventory
3,816,000 Units

Source: Vending Times 2008a; Automatic Merchandiser 2007; NCI estimates

Figure 2-13: Refrigerated Packaged Beverage Vending Machines – Owner/Operator Inventory Share

There are about 10,000 packaged beverage vending machine operators in the United States. Approximately 6,600 are independent operators and 3,400 are bottling company operators (Vending Times 2008b). The operators are responsible for delivering beverages to the vending site, filling the machines about once per week, and maintaining and servicing the machines. The vending site is responsible for paying energy costs.

The average packaged beverage vending machine dispenses about 173 beverages per week (Vending Times 2008a). About half of vending machine revenue goes directly to the bottling company and the remainder is divided between the operator and the vending site.

There are two basic types of packaged beverage vending machines:

- Fully-cooled
- Zone-cooled

In a fully cooled beverage vending machine, all beverages enclosed within the machine are visible to the customer and, therefore, the entire internal volume is refrigerated. The zone-cooled packaged beverage vending machine only cools the beverages that are soon-to-be-vended, meaning only a small portion, or zone, of the internal volume is refrigerated. These vending machines typically have a solid, opaque front. Zone-cooled vending machines account for about 87 percent of the packaged beverage vending machine market (Vending Times 2007). However, the aesthetic appeal and large product variety options of fully-cooled, glass front vending machines have increased their popularity in recent years. We assume fully-cooled machines represent the remaining 13 percent of the installed base.

Refrigerated vending machines consume about 9.6 TWh annually (Table 2-16).

Table 2-16: Refrigerated Packaged Beverage Vending Machine Energy Consumption (2008)

Unit Type	Estimated Inventory[1]	Unit Energy Consumption (kWh/yr)[2]	Total Energy Consumption (TWh/yr)	Total Energy Consumption (%)	Primary Energy Consumption[3] (TBtu/yr)
Fully-cooled	496,080	2,743	1.4	14%	14.2
Zone-cooled	3,319,920	2,483	8.2	86%	85.8
Total	**3,816,000**	**-**	**9.6**	**100%**	**100.0**

Sources:
[1] Vending Times 2008a
[2] NCI estimates that the typical installed refrigerated vending machine uses approximately 10 percent more than the typical new vending machine on the market today, which we assume to be the baseline efficiency used in the ongoing beverage vending machine DOE Rulemaking (DOE 2008b). See Table 3-52 and Table 3-53 for these "typical new" energy consumptions.
[3] Based on 10,405 Btu primary energy/kWh site energy (DOE 2008a, Table 6.2.4)

3 Description of Baseline Equipment

This section describes the baseline equipment characteristics (unit energy consumptions, costs, manufacturers, and end-users) for each of the seven commercial refrigeration equipment types addressed in this report:

- Supermarket refrigeration systems (display cases, compressor racks, and condensers)
- Walk-in coolers and freezers
- Food preparation, worktop, and buffet tables
- Reach-in refrigerators and freezers
- Beverage Merchandisers
- Ice machines
- Refrigerated Vending Machines

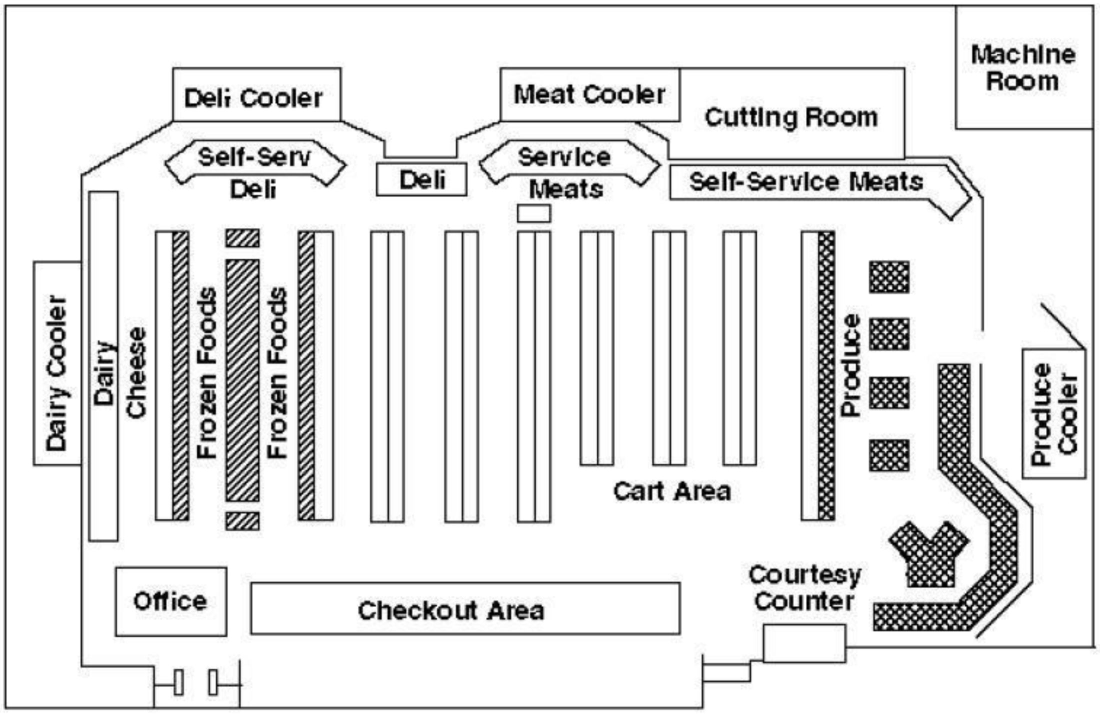
Throughout the report, "baseline" equipment is used to refer to a typical new piece of equipment sold in 2009. The baseline equipment definitions will serve as a basis for comparison in our energy savings analysis in section 4.

3.1 Supermarket Refrigeration Systems

3.1.1 Equipment Description and Illustrations

General System Description

The purpose of supermarket refrigeration systems is to preserve and merchandise food products. Refrigerated food products are generally stored in walk-in coolers and freezers prior to transfer to the refrigerated display cases on the retail floor, from which customers can access them for purchase. The layout of a typical supermarket is depicted in **Figure 3-1**. The refrigerated display cases are generally located at the periphery of the store near their associated walk-in coolers/freezers used for food storage.



Source: IEA 2003

Figure 3-1: Typical Supermarket Layout

Supermarkets range in size from less than 10,000 ft^2 to greater than 80,000 ft^2 total selling area. The average supermarket selling area is approximately 33,000 ft^2. Including non-selling area used for food storage, food preparation, and offices, the typical store size is 47,000 ft^2 (FMI 2008b).

The typical supermarket uses a direct expansion refrigeration system, as shown in **Figure 3-2**. In such a system, the refrigerant travels from the compressors in the machine room to the condensing unit (often on the roof) where heat is transferred from the refrigerant to the outside air, through a refrigerant piping network to the expansion valves and evaporators in the various display cases on the sales floor (where heat is drawn from the cabinet interiors to the refrigerant), and then back to the compressors. Supermarket piping runs are long, resulting in substantial refrigerant charges, typically 1300 to 2500 lbs. (IEA 2003). Therefore, leakage must be carefully controlled to avoid excessive refrigerant charge losses.

Alternatives to the conventional system configuration have gained attention in recent years due primarily to their potential to reduce refrigerant charge, thus reducing maintenance costs and direct global warming impact of the refrigerant. These alternative configurations include secondary loop systems and distributed compressor systems. Section 6.1 of this report (Advanced Energy Saving Technologies) includes further discussion of these technologies. The components of the direct expansion system, including display cases, compressor racks, and condensers, are described in detail below.

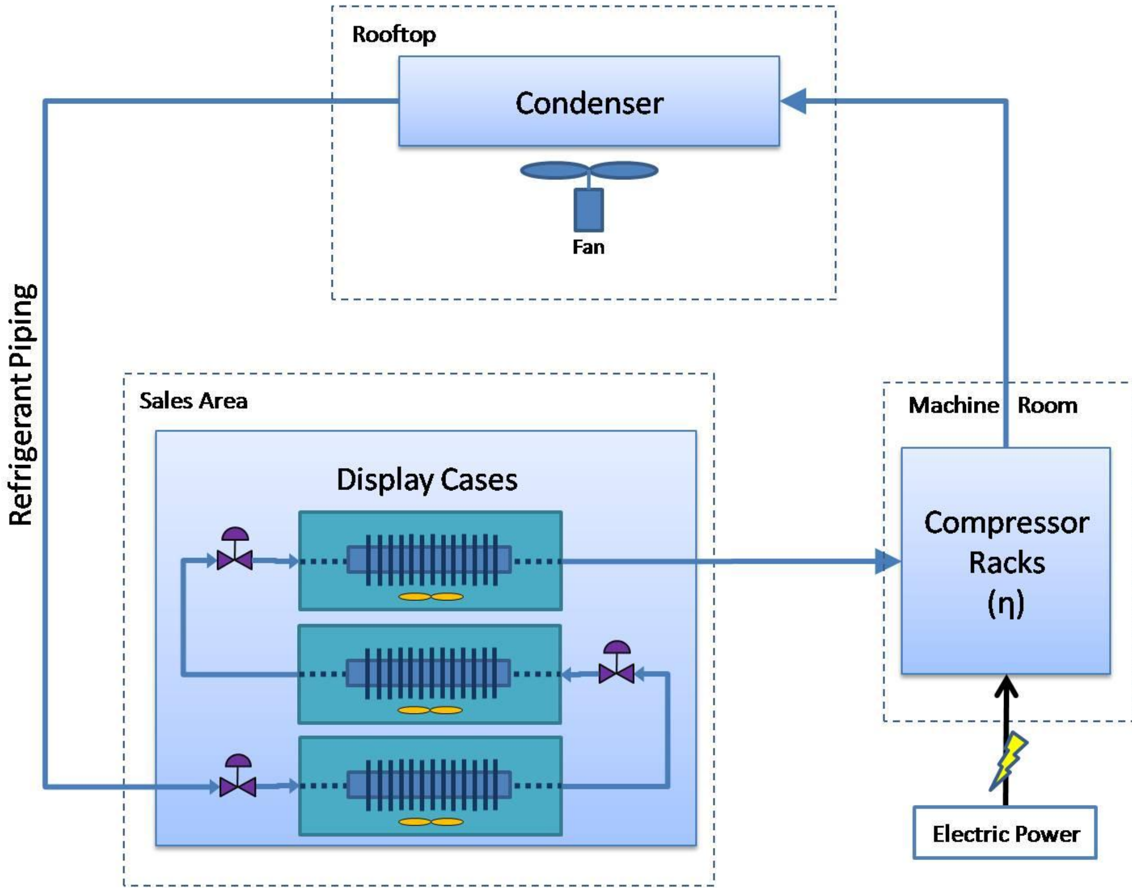


Figure 3-2: Typical Supermarket Refrigeration Circuit

Display Cases

The purpose of supermarket cases is to display food attractively for purchase by customers. Display cases vary by temperature and orientation in order to adequately display the range of fresh products sold in a supermarket. The evaporator coil temperature ranges and their associated applications are shown in **Table 3-1**. About half of the 60 display cases in a typical supermarket will be low or very low temperature cases.

Table 3-1: Evaporator Coil Temperature Ranges by Application

	Temperature Range	Applications
High Temperature	35°F and above	Produce, Flowers
Medium Temperature	10°F to 15°F	Meats, Seafood
	15°F to 25°F	Dairy, Produce, Beer/Juice, Walk-in Coolers (meat)
	25°F to 35°F	Walk-in coolers (dairy, produce), Prep Rooms
Low Temperature	-25°F to -15°F	Frozen Foods
Very Low Temperature	-35°F to -25°F	Ice Cream, Frozen Bakery

The goal for display cases is to maximize sales without sacrificing food preservation. Numerous display case configurations are used to meet the needs of the wide variety of refrigerated products sold in supermarkets. Open display cases are used when possible in order to make the products most accessible to customers. Otherwise, display cases use glass doors, which prevent heat from entering the cabinet more effectively, but require the customer to open the door to access the product. The most common case types are:

- Open Multi-Deck (Figure 3-3)
- Glass-Door Reach-Ins (**Figure 3-4**)
- Meat/Seafood/Deli Display Case (**Figure 3-5**)
- Coffin/Open-Island Freezers (Single-Level) (**Figure 3-6**)

Source: Hussmann 2008

Figure 3-3: Vertical Open Multi-Deck Dairy Display Case

Source: Hussmann 2008

Figure 3-4: Vertical Glass Door Reach-In Display Case for Frozen Food

Source: Hussmann 2008

Figure 3-5: Semi-Vertical Open Multi-Deck Meat Display Case

Source: Hussmann 2008

Figure 3-6: Open Island Display Case for Frozen Food

Every display case contains an expansion valve and one or more evaporators for case cooling. Evaporator fans circulate case air. In open display cases, in addition to cold air being circulated within the case, air is also blown over the open section of the case. This creates an air curtain that forms a boundary between the cold air in the case and the warmer store air, so that the cold air will not freely spill out of the case. Multiple fans are required for most cases.

Low temperature evaporators and some medium temperature evaporators require periodic defrosting to remove frost that condenses and freezes on the evaporator surface. The frost reduces cooling performance by increasing the thermal resistance to heat transfer from the coil to the air and by obstructing airflow.[8] Defrosting is generally accomplished with electric defrost or hot-gas defrost. Electric defrost involves electric resistive heating with a defrost coil that is integrated into the evaporator coil. Hot-gas defrost involves piping and valves that direct hot gas from the compressor discharge into the evaporator. Some medium-temperature cases can defrost during the off cycle simply by allowing the coil to rise to the display-case temperature (i.e., with no active heating of the coil).

The case insulation is typically 1½ to 2 inches thick, with insulating values from R-12 to R-18. Some very low temperature display cases can have insulation thicknesses up to 2 ½ inches. Glass doors are typically fitted with at least double-pane glass. Anti-sweat heaters and glass heaters are used to prevent condensation of water vapor on the outside of the glass. Today's more efficient doors use triple-pane glass having insulating gases encased between the panes, and don't require glass heaters. Cases are generally fitted with lighting to illuminate the products. Refrigerant piping (high-pressure-liquid and suction lines) must be connected to the case. Additional connections are electrical power and condensate drain lines.

[8] A thin layer of frost can actually increase heat transfer due to the added surface area provided by the crystalline structure of frost; however, this benefit is soon lost as the frost layer thickens.

The display case is typically equipped with T8 fluorescent lamps and electronic ballasts inside and outside the refrigerated volume to illuminate the product being sold. Cold air is typically circulated throughout the unit using shaded pole motor fans.

For the energy use analysis in Chapter 4, the baseline unit is defined as the vertical, open, medium temperature display case, which is the most common supermarket display case, similar to the one in Figure 3-3. This unit typically is 12 ft. long with a display area of approximately 53 ft^2. The case is typically insulated to R-13 with 1.5 in. of blown polyurethane foam.

Table 3-2 summarizes the physical characteristics of the vertical, open, medium temperature display case.

Table 3-2: Vertical, Open, Medium Temperature – Baseline Display Case Description

External		Internal	Insulation		Lighting	Evaporator Fans
Length (ft)	Display Area (ft^2)	Volume (ft^3)	Thickness (in)	R-Value per inch (ft^2-°F-h/Btu)	Bulb Type & Location	Motor Power Input (W)
12	53	130	1.5	13	• Inside cabinet: 3 rows of 4 4-ft T8 fluorescent w/ 4 electronic ballasts • Outside cabinet: 3 rows of 3 4-ft T8 fluorescent w/ 3 electronic ballasts	6 x 9W Shaded Pole Motors

Source: DOE 2009a (Baseline specifications in 0 of the DOE technical support document)

Compressor Racks

The compressor racks are located in a machine room separate from the sales area, as shown in **Figure 3-2**. A typical rack system consists of several compressors connected in parallel, piping, electronic controls, and insulation. Integrating the compressor rack into the supermarket refrigeration system requires an extensive piping network.

Most supermarkets use multiple compressor racks, two medium temperature racks and two low temperature racks, with about 200 total hp in connected compressor power. Using a rack of multiple compressors has several advantages over a single unit. Built-in controls integrated into the system allow specific manipulation of capacity loading and duty cycles. Partial loading distributes the necessary load throughout the rack system, preventing overloading a single unit. By modulating compressor capacity, both the longevity of materials and overall efficiency increase. Duty cycles are adjusted accordingly to necessary load requirements, reducing any extraneous energy consumption. A rack of compressors can contain compressors of different cooling capacities in a configuration called “uneven parallel”, which the most widely used approach. By incorporating compressors having different cooling capacities, a larger range of loads can be served efficiently than if all compressors were of equal capacities.

There are several types of compressors available:

- Reciprocating – Reliable and efficient, most widely used compressor.
- Scroll – Compression process enables high efficiency, light weight, minimal refrigerant loss; market share is modest, but growing (Emerson 2009).
- Screw – Reliable and ideal compressor for variable speed and efficient partial loads; very small market share.

This study assumes the baseline compressor rack configuration for a supermarket is two medium temperature racks and two low temperature racks of scroll compressors.

Condensers

A typical supermarket uses a set of rooftop condensers to exchange the heat of the refrigerated cabinets with the outside air. The typical capacity of a set of condensers is 1,520 MBtu, which is comprised of two low temperature condensers (THR_L = 240 MBtu/hr each, suction temperature = -25°F, condensing temperature 110°F) and two medium temperature (THR_M = 520 MBtu/hr each, suction temperature = 15°F, condensing temperature = 115°F) condensers (EIA 2008).[9]

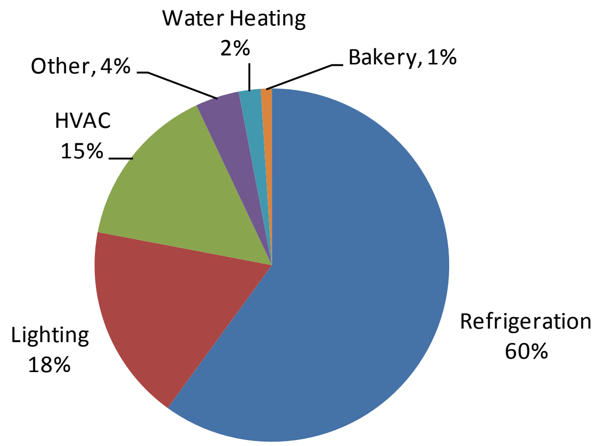
Walk-ins

Walk-ins are located around the periphery of the sales area for temporary food storage. To enabling more efficient restocking of display cases, each walk-in tends to store the food for the nearest section of display cases. Typical categories include produce, dairy, deli, meat, and frozen food (see **Figure 3-1**). See Section 3.2 for a more detailed equipment description for walk-in coolers and freezers.

3.1.2 Energy Consumption

Supermarkets typically use on the order of 3,000,000 kWh of electricity per year according to the EPA Supermarket Energy Use Profile (EPA 2007). The typical breakdown of this usage among building systems is shown below in Figure 3-7. Refrigeration represents 60 percent of supermarket energy use according to the EPA. However, other sources state that the percentage is as low as 30 to 50 percent (Lazzarin 2008).

[9] THR = Total Heat Rejected



Total Annual Electricity Consumption = ~ 3,000,000 kWh (Site Energy) [1]
or ~ 31,215 Million Btu (Primary Energy) [2]

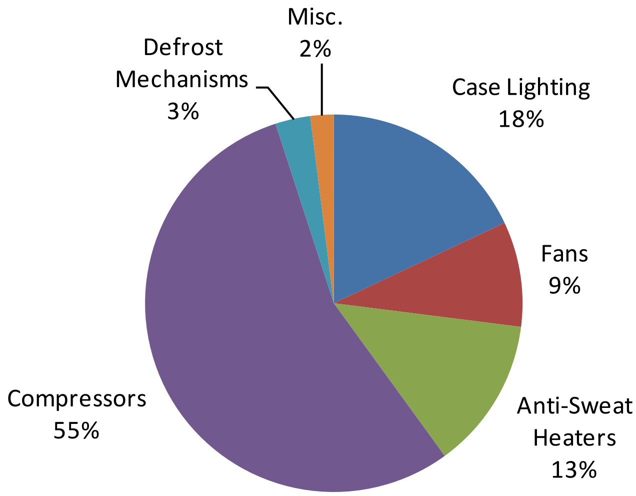
Source: EPA 2007

Figure 3-7: Annual Electricity Consumption Breakdown for Typical Supermarket (33,000 ft^2 selling area)

[1] Based on EPA analysis of data from the Energy Information Administration's 2003 Commercial Building Energy Consumption Survey.
[2] 10,405 Btu primary energy/kWh site energy (DOE 2008a, Table 6.2.4)

A supermarket with 47,500 ft^2 total area and 24 hour-per-day operation is chosen as the baseline for the energy consumption and savings. This store area is the median size of existing supermarkets, which is just above the average size of new stores at 46,000 ft^2, but well below the 55,000 ft^2 range of the 1990s (FMI 2008a). Most large supermarkets are open between 18 and 24 hours per day. For this analysis, we decided to analyze the upper limit of the range of operating hours. Since the refrigeration equipment operates 24 hours/day regardless of store hours, the differences are largely associated with additional product pull-down loads and more door openings (assuming higher sales volumes as store hours increase).

The typical breakdown of the refrigeration electricity consumed by refrigeration systems is shown below in Figure 3-8.



Sources: Manufacturer Spec Sheets, Manufacturer Interviews, and NCI Estimates

Figure 3-8: Typical Supermarket Refrigeration System Electricity Consumption Breakdown

We estimate the typical supermarket energy consumption to be 1.9 million kWh/yr, as shown in **Table 3-3**. Following the table are details on the energy consumption of each component, including display cases, compressors, condensers, and walk-ins.

Table 3-3: Supermarket Refrigeration Energy Consumption

	Estimated Inventory per Store[1]	Unit Energy Use (kWh/yr)	Store Energy Use (kWh/yr)	Total Energy Use [2] (TWh/yr)	Total Energy Use (%)	Primary Energy Use (TBtu/yr) [3]
Display Cases	60	9,783	586,980	20.5	31%	214
Compressors	1	1,023,333	1,023,333	35.8	54%	373
Condensers	1	138,000	138,000	4.8	7%	50
Walk-ins	7	varies	139,905	4.9	8%	51
Total			1,888,218	66.1	100%	688

Source: DOE 2009a
[1] NCI estimates
[2] Assume 35,000 supermarkets installed in the U.S.
[3] 10,405 Btu primary energy/kWh site energy (DOE 2008a: Table 6.2.4)
Note: Energy use values have been rounded to the nearest whole number, and therefore the total may not exactly equal the sum of the energy use values for each equipment type.

Display Cases

Table 3-4 shows the steady state thermal load breakdowns for a typical vertical medium temperature supermarket display case without doors.

Table 3-4: Supermarket Display Case Thermal Load Breakdown (Btu/hr)

Component	Vertical Glass Door Medium Temp	Vertical Glass Door Low Temp	Vertical Open Medium Temp
Evaporator Fan	512	491	920
Lighting	1,188	1,188	1,700
Defrost	n/a	677	n/a
Anti-sweat	1,195	2,391	n/a
Infiltration	524	865	16,070
Conduction	631	915	590
Radiation	1,456	2,756	1,450
Total	**5,507**	**9,283**	**20,730**
Note: Thermal load values have been rounded to the nearest whole number, and therefore the total may not exactly equal the sum of the thermal load values for each component.			

Source: DOE 2009a

Table 3-5 shows the energy consumption for each of the seven most common display case types. We generate a weighted average energy consumption of 8,894 kWh/yr for display cases using the estimated installed base by type.

Table 3-5: Average Unit Energy Consumption of Supermarket Display Cases

Display Case Type[1]	Installed Base (units)	Weighting Factor	Unit Energy Consumption (kWh/yr)
VOP.RC.M	387,430	28%	7,986
SVO.RC.M	295,520	22%	6,030
HZO.RC.M	45,410	3%	2,314
VCT.RC.M	27,670	2%	8,688
SOC.RC.M	93,120	7%	7,965
VCT.RC.L	384,830	28%	14,893
HZO.RC.L	142,780	10%	3,854
Weighted Average UEC (Typical New):			**8,894**
Weighted Average UEC (Typical Installed)[2]:			**9,783**

Source: DOE 2009a
[1] VOP = vertical open, SVO = semi-vertical open, HZO = horizontal open,
VCT = vertical transparent doors, SOC = service over counter, RC =

remote-condensing, M = medium temperature, L = low temperature
[2] We assume that the typical display case installed today uses 10 percent more energy than a typical new display case.

Table 3-6 shows the energy consumption breakdown for three typical new display case types. These three types are used in the energy-savings and economic analysis to estimate energy savings potential (see section 4).

Table 3-6: Energy Consumption Breakdown for Baseline Supermarket Display Cases (Typical New)

Display Case Type[1]	Component	Power Consumption/ Unit (W)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)
Medium Temperature with Glass Doors	Lighting [2]	348	100	3,049	35%
	Evaporator Fans [3]	150	96	1,259	15%
	Anti-sweat Heating	500	100	4,380	50%
	Total	**-**	**-**	**8,688**	**100%**
Low Temperature with Glass Doors	Lighting [2]	348	100%	3,048	21%
	Evaporator Fan [3]	150	96%	1,259	8%
	Defrost	5000	4%	1,825	12%
	Anti-sweat Heating	1000	100%	8,760	59%
	Total	**-**	**-**	**14,893**	**100%**
Medium Temperature with No Doors	Lighting[4]	641.6	100%	5,620	70%
	Evaporator Fan[5]	270	100%	2,365	30%
	Total	**-**	**-**	**7,986**	**100**

Source: DOE 2009a
[1] All three case types are vertically oriented
[2] 6 Lamps at 58-Watts/each, 6 ballasts
[3] Five 6-Watt output evaporator fans
[4] 12 bulbs inside refrigerated cabinet, 9 bulbs outside refrigerated cabinet, 7 ballasts (30W/lamp, 1.4W/ballast)
[5] Six 9-Watt output evaporator fans
Note: Energy consumption values have been rounded to the nearest whole number, and therefore the total may not exactly equal the sum of the energy use values for each component.

Compressor Racks

Approximately one third of the total annual electricity consumption for a large supermarket is attributable to compressors. Many options are available for incorporating different compressors into integrated compressor rack systems. Selecting a different type directly affects power and energy consumption. **Table 3-7** lists the energy consumption for a typical reciprocating compressor system, which is currently the most common compressor type. Scroll and screw compressors are also used in some supermarket systems and are included in our estimates of the energy use of the installed base. The capacity and energy consumption values represent the requirement for the entire refrigeration system of a typical store at the given temperature. In other words, the low temperature line items represent two low-temperature racks and the medium temperature line items represent two medium-temperature racks.

Table 3-7: Supermarket Compressor Energy Consumption[1]

Compressor Type	Temperature Range	Capacity (Btu/h)	Power (kW)	Energy Consumption (MWh/yr)
Reciprocating (Typical)	Low	308,000	63	350
	Medium	769,000	99	550

Source: Emerson 2009, NCI estimates

[1] Assume that "low temperature" rows represent the low temperature requirement for a typical supermarket and that the "medium temperature" rows represent the medium temperature requirement for a typical supermarket.

The total national energy use from supermarket compressor racks is estimated to be 373 Trillion Btu per year. **Table 3-8** shows the assumptions behind this energy consumption estimate. We estimate that the typical installed supermarket refrigeration system uses approximately 1,023 MWh per year.

Table 3-8: Supermarket Compressor Rack Primary Energy Use

	Annual Refrigeration Energy Use per Store (MWh/yr)[1]			# of Stores (units)[2]	Total Annual Electricity Use (TWh/yr)	Total Annual Primary Energy Use (Trillion Btu/yr)[3]
	Medium Temperature Racks	Low Temperature Racks	All Racks			
Typical Installed	400	623	1,023	35,000	35.8	373
Typical New	550	350	900	35,000	31.5	328

Sources:

[1] "Typical installed" energy consumption values are the average of the three compressor types shown in Table 3-7. "Typical new" energy consumption values are for the reciprocating compressor only.

[2]From Progressive Grocer (2008)
[3]Primary energy includes the energy associated with generation (for electricity only), transmission, and distribution to the end user. This report uses the national average efficiency for 2006 (10,405 Btu primary energy/kWh site energy) (DOE 2008a, Table 6.2.4).

Condensers

Supermarket condensers use approximately eight percent of supermarket refrigeration energy consumption to power the fan motors, which blow air across the condenser coil to assist in heat transfer. Supermarkets most commonly use one remote air-cooled condenser for each refrigeration circuit, all with 3-phase induction fan motors. We assume the typical supermarket uses four condenser coils (one per compressor rack). **Table 3-9** shows the condenser energy consumption for a typical supermarket installed today.

Table 3-9: Supermarket Condenser Energy Consumption

	Power (kW)	Unit Energy Consumption (kWh/yr)	Total Annual Electricity Use (TWh/yr)	Total Annual Primary Energy Use (Trillion Btu/yr)[1]
Typical Installed	25	138,000	4.8	50
Typical New	22	120,000	4.2	44

Source: EIA 2008.
[1]Primary energy includes the energy associated with generation (for electricity only), transmission, and distribution to the end user. This report uses the national average efficiency for 2006 (10,405 Btu primary energy/kWh site energy) (DOE 2008a, Table 6.2.4).

Walk-ins

Supermarket walk-ins also use approximately eight percent of supermarket refrigeration energy consumption to power the evaporator fan motors, lighting, and electric defrost. **Table 3-10** shows the walk-in energy consumption for a typical supermarket installed today.

Table 3-10: Supermarket Walk-In Energy Consumption

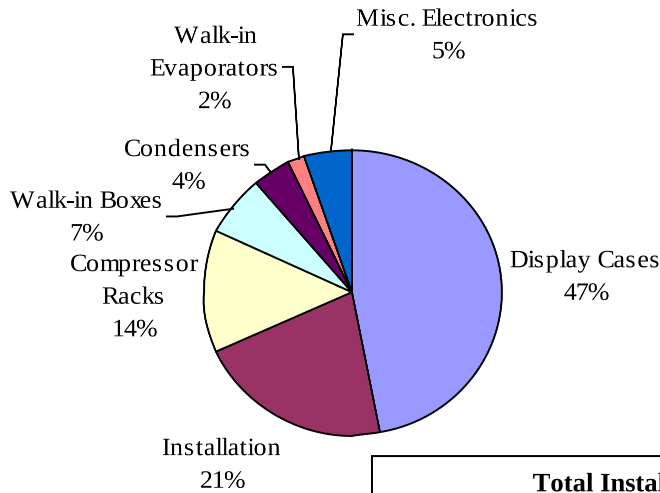
	Power Consumption per ft² (W/ft2) [1]	Total Power Consumption (kW) [2]	Duty Cycle (%)	Energy Consumption (kWh/yr)
Meat Coolers (400 ft²)				
Evaporator Fans	3.3	1.3	100%	11,680
Electric Defrost	25.0	10.0	4%	3,650
Lights	1.2	0.5	50%	2,102
				17,432
Other Coolers (2600 ft²)				
Evaporator Fans	3.3	8.7	100%	75,920
Lights	1.2	3.1	50%	13,666
				89,586
Freezers (1,000 ft²)				
Evaporator Fans	2.3	2.3	100%	19,710
Electric Defrost	25.0	25.0	4%	9,125
Lights	0.9	0.9	50%	4,052
				32,887

Source:
[1] Power consumption values are from walk-in section and normalized by floor area (Section 3.2.2).
[2] The floor areas for each walk-in type are from ADL 1996.

3.1.3 Purchase and Installation Costs

The total installed cost of a typical supermarket refrigeration system (approximately 100 tons) is approximately $1.3 million (ADL 1996, adjusted to $2008)[10]. Figure 3-9 shows the breakdown of the total installed cost of such a system. The breakdown of costs for two most significant equipment categories, display cases and compressor racks, are shown in **Table 3-11** and **Table 3-12**, respectively.

[10] See 0 for more discussion on inflation assumptions.



Total Installed Cost = $1.3 million
(increased from $1.05 million in ADL 1996; see 0 for discussion on cost escalation assumptions.)

Source: ADL 1996

Figure 3-9: Installed Cost Breakdown for a 100 Ton Supermarket Refrigeration System ($2008)

Table 3-11: Refrigerated Display Case Total Installed Cost ($2008)

	Cost per Case	Cost per Store[1]
Equipment Purchase Price	$ 9,700	$ 773,000
Installation	$ 2,000	$ 160,000
Total Installed	$ 11,700	$ 933,000

Source: DOE 2009a
[1]Assumes 80 cases per supermarket.

Table 3-12: Compressor Rack Total Installed Cost ($2008)

	Cost per Rack	Cost per Store[1]
Equipment Purchase Price	$200,000[2]	$798,000
Installation[2]	$20,000	$80,000
Total Installed	$220,000	$878,000

Source: Adapted from ADL 1996
[1] Assumes 4 compressor racks for a typical supermarket.
[2] Equipment purchase price ranges from $70-$115,000 per case.
[3] Installation includes machine delivery, placement, electric connections, and piping configuration and ranges between $15,000 and $25,000.

A new compressor rack is typically delivered to the supermarket as an integrated rack. The rack includes manifold piping, valves, the oil separator, the liquid receiver, controls, and the compressor motor starters along with the compressors. The complete assembly is mounted on a frame for easy shipping and installation. Complete rooftop mechanical rooms can also be delivered.

3.1.4 Life, Reliability, and Maintenance Characteristics

According to industry experts, the typical lifetime of refrigeration equipment used in large grocery and multiline retail businesses is 10 years, while a 15-year lifetime is more appropriate for refrigeration equipment used in small grocery or convenience stores (DOE 2009a). Supermarkets system compressors have a 10-year expected lifetime. Compressors within a rack are usually replaced individually. The typical replacement rate for compressors is one to two replacements per year, and the typical compressor lifetime is approximately 12 -15 years. (Emerson 2009) The typical lifetime of air-cooled condensers is at most 10 years. Refrigerated display cases have an estimated functional life of up to 15 years, but are usually replaced for cosmetic reasons prior to the end of their useful life. Display cases are often replaced only when stores are renovated. Renovations typically take place every 6 to 10 years, though not all equipment is replaced during each renovation (FMI 2008a). The systems are expected to operate reliably if properly installed and maintained. The potential for costly food loss due to failure has resulted in a high reliability level for all refrigeration system components. Refrigerated goods are estimated to represent approximately 45% of supermarket sales, and at a given point, the value of refrigerated inventory in a supermarket generally ranges from $200,000 to $300,000 depending on store size (TIAX 2005).

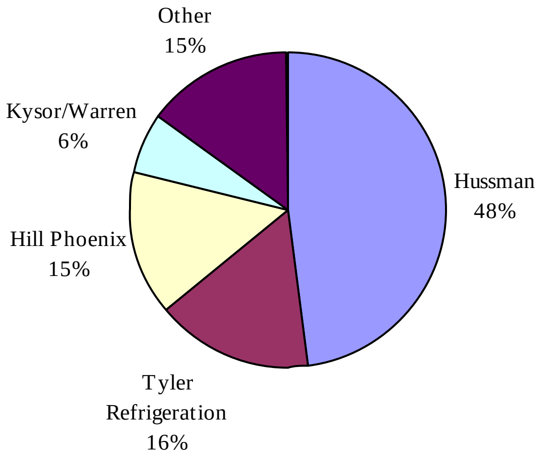
The average supermarket spends $40,000 to $50,000 on maintenance and repairs annually for HVAC, refrigeration, and lighting, excluding labor (Emerson 2009). Labor costs bring the total to about $100,000. Display cases require approximately $160 each in annual maintenance on average (DOE 2009a).

Refrigeration system maintenance activities include:

- cleaning evaporator coils, drain pans, fans and intake screens,
- lubricating motors,
- inspecting door gaskets and seals,
- lubricating hinges,
- cleaning condenser coils,
- checking refrigerant charge as necessary,
- checking compressor bearing lubrication, oil, and filter,
- checking starter panels and controls,
- checking defrost system operation.

3.1.5 Major Manufacturers and Distribution Channels

Four companies represent approximately 85 percent of the U.S. refrigerated display case market, with approximately 185,000 units shipped in 2004 (DOE 2009a). However, *Appliance Magazine* provides no precise definition of a refrigerated display case and it is therefore unclear what specific types of equipment the data covers – equipment that is self-contained versus remote condensing, and equipment with doors versus without doors. As of 2004, Hussmann Corporation, a division of Ingersoll Rand, was the largest domestic manufacturer of refrigerated display cases, holding approximately 48 percent of the U.S. market. Other manufacturers make up the remaining 52 percent of U.S. market share. **Figure 3-10** shows the breakdown of the U.S. refrigeration equipment market among major manufacturers.



Source: Appliance Magazine 2005

Figure 3-10: U.S. Market Share of Major Refrigerated Display Case Manufacturers

Note: Appliance Magazine has not published updated shipment and market share data for display cases since 2005.

Larger supermarket chains typically involve central engineering staff in design, selection, and installation of equipment. The manufacturer and/or consultant or contractor have a larger role in equipment selection for independent operators.

Major compressor rack manufacturers include:

- Hussman
- Kysor-Warren
- Heatcraft
- Tyler Refrigeration
- Hill Phoenix
- Zero Zone

Major compressor manufacturers include:

- Copeland

- Carlyle
- Bitzer

Major condenser fan motor manufacturers include:
- Baldor
- Emerson

3.1.6 Major End-Users

The major end-users of supermarket refrigeration systems are the large supermarket chains, super-center chains (grocery and mass merchandise), independent operators, and convenience stores.

Total food sales for the supermarket industry were $535.4 billion in 2007 (FMI 2008b). Table 3-13 provides a breakdown of the major chains.

Table 3-13: Major Food Retail Chains

Rank	Chain	No. of Stores	% Market Share of Supermarket Industry (by 2007 Grocery Sales)
1	Wal-Mart Supercenters	2,447	21
2	The Kroger Co.	3,269	12
3	Safeway, Inc.	1,738	8
4	Costco Wholesale Group	520	7
5	SUPERVALU, Inc.	2,512	6
6	Sam's Club	587	5
7	Publix Super Markets, Inc.	928	4
8	Ahold USA, Inc.	721	4
9	Delhaize America	1,551	3
10	H-E-B	337	3
Totals		**14,610**	**73**

Source: FMI 2007

Independent operators are subdivided into voluntaries and cooperatives. Voluntaries are groups of retailers that voluntarily do business with a particular wholesaler. These groups are distinguished by close teamwork between wholesaler and retailers, while maintaining independent status. Cooperatives are groups of retailers who have jointly established a wholesaling operation to maintain low costs.

See Appendix A for information on the distribution chain between the manufacturer and the end-user.

3.2 Walk-In Coolers and Freezers

Walk-ins have the basic components of a refrigeration system (evaporator, condenser and compressor) and use either a self-contained or split-system arrangement. Generally, packaged units are used in systems less than 3 hp while split-system units are used in systems greater than 3 hp. This section focuses on walk-ins with dedicated refrigeration systems. However, as discussed in section 3.1, walk-ins are also found as part of supermarket refrigeration systems, and we assume that the supermarket refrigeration load includes the necessary walk-in loads. Therefore, the compressor and condenser fan energy use is represented in the supermarket refrigeration energy use. However, we assume that the characteristics of the evaporator fans, lighting, defrost, and anti-sweat heaters discussed in this chapter still apply to supermarket walk-ins.

A self-contained walk-in system consists of a manufactured package that contains all the basic components. The refrigeration system is mounted either on the roof or wall of the insulated room and is configured such that the condenser has access to the outside of the room and the evaporator has access to the inside of the room. These units are less expensive due to the simplification of the construction and installation (ADL 1996). However, since they are shipped as a single package, they are limited to smaller sizes.

Split systems have a separate condensing unit (condenser and compressor) and unit cooler (evaporator, fans and expansion device). The condensing unit is usually located on the building rooftop or outside at ground level. Generally, refrigeration components and controls are ordered separately and assembled on site by local refrigeration contractors. This configuration is advantageous for walk-ins that are inside buildings because heat and noise can be kept outdoors. Also, it is generally more energy efficient to reject heat directly outdoors, rather than rely on the space-conditioning system to reject the heat.

3.2.1 Equipment Description and Illustrations

The purpose of walk-ins is to temporarily store refrigerated or frozen food or non-food products. Walk-ins range from 80 to 250 square feet of floor area and are approximately 8 feet in height. Figure 3-11 and Figure 3-12 are packaged walk-in units with the condensing unit attached to the side wall and roof, respectively, which are typical of full-service restaurants. Figure 3-13 shows a walk-in typically used in a convenience store.

Source: Nor-Lake 2008

Figure 3-11: Packaged Walk-In with Wall-Mounted Refrigeration System (Typical of Restaurants)

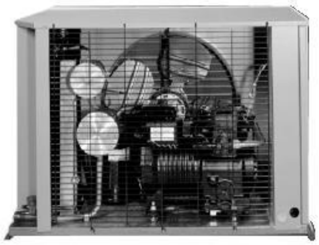
Source: Nor-Lake 2008

Figure 3-12: Packaged Walk-In with Top-Mounted Refrigeration System (Typical of Restaurants)

Source: Hussmann 2008

Figure 3-13: Walk-In Cooler with Merchandising Doors (Typical of Convenience Stores)

Figure 3-14 and **Figure 3-15** are pictures of the condensing unit and cooling unit respectively used in a split system. Figure 3-16 shows a schematic of a split system walk-in refrigeration system.

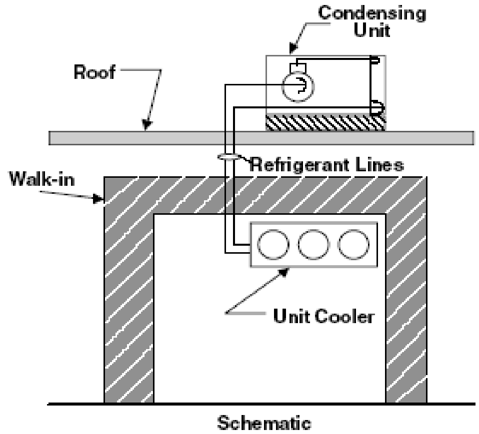


Source: AWRCO 2008

Figure 3-14: Typical Walk-In Condensing Unit

Source: AWRCO 2008

Figure 3-15: Typical Walk-In Unit Cooler



Source: ADL 1996

Figure 3-16: Walk-In Cooler Schematic

The basic components of a walk-in are those typical of a refrigeration system: semi-hermetic or hermetic compressors, evaporator fans, evaporator coils, condenser fans, and condenser coils. Refrigerant flow is controlled by a thermostatic expansion valve; however some smaller units use capillary flow restrictors. Most units today use HFC-404A refrigerant. Compressor sizes range from 1.5 hp to 5 hp depending on the application. Restaurants typically use 2 hp coolers and 3 hp freezers. Convenience stores, which are expected to cool or freeze many products delivered at room temperature, often use 5 hp motors. Most walk-ins use fans with shaded-pole motors while fans of high-efficiency units use PSC motors.

Walk-ins can be insulated using blown polyurethane, expanded polystyrene and extruded polystyrene (Craig Industries 2008). Walk-ins are typically insulated to R-28 with a minimum of 4 inches of blown polyurethane foam. Most manufacturers now provide the option of using 5 or 6 inches of insulation for lower temperatures (Bally 2008). According to the Energy Independence and Security Act of 2007 (EISA), *Public Law 110-140*, wall, ceiling and door insulation must be at least R-25 for coolers and R-32 for freezers (effective January 1, 2009). Freezer floors also need to have insulation of at least R-28. In most cases the walls are constructed of galvanized steel. Other wall structures used are stainless steel, aluminum and fiberglass-reinforced plastic.

There is always at least one door for walk-ins. Doors are generally well insulated and have durable gaskets. Freezers utilize anti-sweat heaters for their access doors. Convenience stores generally use merchandising doors on one side of the walk-in because they allow easy restocking of display shelves. They generally consist of multilayered insulated glass and have anti-sweat heaters.

Interiors of walk-ins are generally lit with one or more incandescent lights yielding an average load of one watt per square foot of floor space. Display walk-ins may also use fluorescent lighting for product illumination. Typically, T12 lamps are used, but T8 lamps and LEDs are becoming more common (Bally 2008).

3.2.2 Energy Consumption

We developed two baseline descriptions for determining energy consumption characteristics and energy savings potential. The first is a split-system convenience-store cooler with merchandising doors. The second is an indoor packaged walk-in freezer with no merchandiser. Several manufacturers confirmed our baseline equipment selections, including Kysor Panel Systems and Craig Industries. Table 4-9 lists the characteristics of each baseline walk-in configuration.

Table 3-14: Characteristics of the Baseline Cooler and Freezer

	Walk-in Cooler	Walk-in Freezer
Floor Size (ft^2)	240	80
Width (ft)	24	8
Depth (ft)	10	10
Height (ft)	8’6”	7’7”
Wall Thickness (in.)	4	4
Wall R-Value	28.6	30
Merchandising Doors (ft)	(10) 2’ x 6’ 1 5/8”	-
Number of Panes	2[1]	-
Access Doors	(1) 3’ x 6’6”	(1) 3’ x 6’6”

Refrigerant Type	R-404A	R-404A
Compressor HP	5	1 ½
Compressor Type	Semi-Hermetic Reciprocating	Semi-Hermetic Reciprocating
Ambient Temperature (°F)	95	90
Walk-in Temperature (°F)	35	-10
Condensing Temperature (°F)	105	113
Evaporating Temperature (°F)	25	-26
Compressor Capacity (kBtuh)	45.0	4.9
Compressor Power (W)	3850	1445
EER (Btu/W)	11.7	3.41
Liquid Suction Heat Exchanger	Yes	Yes
Anti-sweat Wattage (W)	300[4]	230[3]
Anti-sweat Control	None	None
Defrost Wattage (W)	-	1500
Defrost Control	-	Time Initiated / Temperature Terminated
Pan Heater Wattage (W)	-	500
Pan Heater Control	-	Time Initiated / Temperature Terminated
Source: Adapted from ADL 1996, updated with interview with major manufacturer in May 2008 [1] Double pane insulated (inert gas) door [2] Actual ambient temperature varies – reported temperature is compressor design point [3] Access door anti-sweat [4] Merchandising doors only		

The steady-state loads for the walk-ins are presented in Table 3-15. The over sizing of the compressor is greater for the cooler as it is sized for pulling product temperature down and due to the frequent opening of the merchandising doors.

Table 3-15: Walk-in Refrigeration Load Breakdown (Btu/h)

	Walk-in Cooler (240 ft^2)	Walk-in Freezer (80 ft^2)
Evaporator Fans	2,730	614
Coil Defrost	-	215
Pan Heater	-	71
Lighting	556	117
Wall Losses (wall)	1,270	1,103
Wall Losses (merch. doors)	4,146	0
Infiltration	420	150
Total	9,226	2,290
Compressor Capacity	44,970	4,929
Source: Adapted from ADL 1996, updated with interview with major manufacturer in May 2008		

Heat exchanger information is located in Table 3-16 below. The heat exchangers of both systems use standard fin-tube construction, however the cooler's evaporator is able to provide significantly more cooling than the freezer. Low-temperature evaporators have restrictions on fin spacing to ensure air flow is acceptable when frost layers are thick. The condenser fan motors are more efficient than the evaporator fan motors because they are larger. The fan blades are usually stamped aluminum (ADL 1996).

Table 3-16: Baseline Walk-in Heat Exchangers

		Walk-in Display Cooler	Walk-in Freezer
Evaporator	Face Area (in^2)	*	288
	Air Flow (CFM)	3,200	1,680
	Number of Fans	8	2
	Fan Type	Propeller: 12", steel hub, pressed aluminum blades	Propeller: 7", steel hub, pressed aluminum blades
	Fan Wattage	110 each	90 each
	Fan Motor Type	Shaded Pole (1/20 hp)	Shaded Pole (1/40 hp)
Condenser	Face Area (in^2)	*	270
	Air Flow (CFM)	*	1,625
	Number of Fans	2	1
	Fan Type	Propeller: steel hub, pressed aluminum blades	Propeller: 18", steel hub, pressed aluminum blades
	Fan Wattage	530 each	329
	Fan Motor Type	PSC (1/2 hp)	Capacitor Start Induction Run (1/6

			hp)
Source: Adapted from ADL 1996, updated with interview with major manufacturer in May 2008 * Data not available			

Table 3-17 and **Table 3-18** show the estimated energy consumption for the baseline freezer and cooler, respectively.

Table 3-17: Merchandising Walk-in Cooler - Energy Consumption Breakdown

	Power Consumption (W)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)	Energy Consumption per Area ($kWh/yr/ft^2$)[3]
Compressor	3,850	66	22,259	53	n/a
Evaporator Fans (8)	880	100	7,008	16	29.2
Condenser Fans (2)	1,508	66	8,719	21	n/a
Anti-sweat Heater	300	100	2,628	6	n/a
Display Lighting	219	66	1,266	3	5.3
Box Lighting	69	50	302	1	1.3
Total			**42,182**	**100**	

Source: Adapted from ADL 1996, updated with interview with major manufacturer in May 2008
[1] Sized to operate no more than 16 hours per day, yet duty cycle is typically less
[2] Display lighting is provided by 4 60" T8 31.5 W fluorescent lamp (w/ 2 lamps per ballast)
[3] Used for supermarket walk-in energy consumption (compressor, condenser fan, and anti-sweat heater energy is not included in supermarket walk-ins, because that energy is either accounted for separately as compressor rack and condenser energy or not include in the case of anti-sweat heaters).
Note: Energy consumption values have been rounded to the nearest whole number, and therefore the total may not exactly equal the sum of the energy use values for each component.

Table 3-18: Storage-only Walk-in Freezer - Energy Consumption Breakdown

	Power Consumption (W)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)	Energy Consumption per Area (kWh/yr/ft^2)[3]
Compressor	1,445	70	8,861	57	n/a
Evaporator Fans (2)	180	100	1,577	10	19.7
Condenser Fan	329	70	2,017	13	n/a
Coil Defrost [1]	1,500	4	548	4	6.8
Drip Pan Heater [2]	500	4	183	1	2.3
Anti-sweat Heater	230	100	2,015	13	n/a
Lighting	74	50	324	2	4.1
Total			**15,524**	100	

Source: Adapted from ADL 1996, updated with interview with major manufacturer in May 2008
[1, 2] Coil defrost and drip pan heater operated for 60 minutes every 24 hours
[3] Used for supermarket walk-in energy consumption (compressor, condenser fan, and anti-sweat heater energy is not included in supermarket walk-ins, because that energy is either accounted for separately as compressor rack and condenser energy or not include in the case of anti-sweat heaters).
Note: Energy consumption values have been rounded to the nearest whole number, and therefore the total may not exactly equal the sum of the energy use values for each component.

3.2.3 Purchase and Installation Costs

The purchase prices for a 10’ x 24’ cooler with a floor and merchandising doors and an 8’ x 10’ packaged freezer are shown in **Table 3-19**. These two units represent the baseline units described above.

Table 3-19: Walk-in Total Installed Cost ($2008)

	Walk-in Cooler (10 ft x 24 ft, with merchandising drs)	Walk-in Freezer (8 ft x 10 ft)
Equipment Purchase Price	$32,500[1]	$12,500
Installation	$4,700[2]	$1,000[3]
Total Installed	$37,200	$13,500

Source: ADL 1996, adjusted for inflation

[1] Breakdown of purchase price: refrigeration system (25)%, floor (10%), doors/lighting/shelving (35%), other (30%).

[2] Installation of larger split systems ranges from $3,100 to $6,300

[3] Installation of a packaged unit can be completed by two people in one day and ranges from $600 to $1200.

3.2.4 Life, Reliability, and Maintenance Characteristics

Walk-ins have an expected lifetime of 12 - 25 years. The compressor has an expected life of 8 to 12 years, thus will likely be replaced once or twice during the life of the walk-in. If the condensing unit is otherwise in good condition, only the compressor needs to be replaced. Environmental issues may drive the replacement of the entire condensing unit (Craig Industries 2008).

Maintenance for walk-ins includes monitoring the refrigerant pressures to confirm they are in normal ranges and cleaning heat exchanger surfaces of debris. Often maintenance is only done after the system fails or is cooling inadequately.

3.2.5 Major Manufacturers

The market for walk-ins is consolidating – several major manufacturers have emerged: Kysor Panel Systems, Standex (Nor-Lake, Master-Bilt), Manitowoc (Kolpak, Harford, Shannon Group), CrownTonka, Bally Refrigerated Boxes, and National Cooler. Manitowoc recently announced the acquisition of Enodis (Kysor Panel Systems is part of Enodis). However, the walk-in market for food services remains fragmented – there are many manufacturers and none have a dominant market share (CrownTonka 2008).

3.2.6 Major End-Users

The major end-users of walk-ins are fast-food restaurants, sit-down restaurants, institutional food service, convenience stores, and other mercantile applications (e.g. florists). The largest end users are major fast food restaurant chains (i.e., McDonalds, Kentucky Fried Chicken/Taco Bell, Burger King and Pizza Hut), large sit down restaurant chains such as Dennys, and large convenience store chains (e.g. 7-Eleven and Circle K). The larger end-users generally purchase walk-ins directly from the manufacturer. Other end-users purchase walk-ins through food sales and food service equipment dealers and through refrigeration equipment wholesalers (CSG 2008).

3.3 Refrigerated Food Service Equipment

3.3.1 Equipment Description and Illustrations

For the energy savings analysis, prep tables have been chosen as the baseline unit to represent the refrigerated food service equipment category, because they make up the majority of shipments within the category. Work-top tables and buffet tables, depicted in section 0, have far fewer shipments compared to prep tables and did not warrant a separate category in this report.

A food preparation table is a cabinet with a table top, sliding drawers, and a refrigerated storage compartment underneath the table top, and easily accessible food compartments with lids on the table top, also referred to as a “refrigerated rail”. Typically, it is about 2 to 3 ft deep, 3 ft high, and ranges in width from about 3 to 10 ft. A prep table is designed to provide convenient access and storage of food. Cabinet capacities range from about 10 to 40 ft^3; capacities and dimensions are standard from most manufacturers. **Figure 3-17** shows a typical prep table.

Source: McCall Refrigeration

Figure 3-17: Typical Preparation Table

Materials used on exteriors and interiors are stainless steel, painted steel, aluminum-coated steel, aluminum, and vinyl-clad steel with wood grain or other patterns. Materials must (1) be easy to keep clean; (2) not be discolored or etched by common cleaning materials; (3) be strong enough to resist denting, scratching, and abrasion; and (4) provide necessary frame strength.

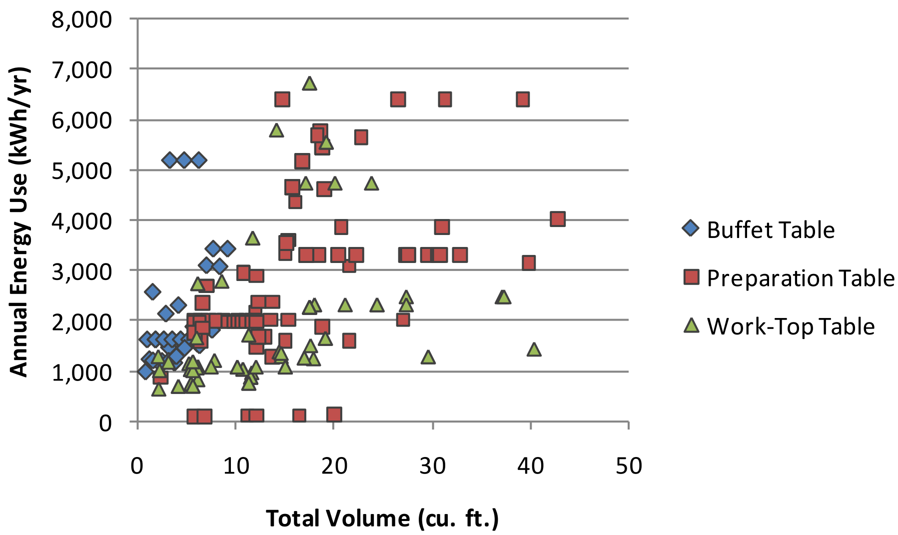
Refrigerated prep tables are available for medium-temperature ranges: a maximum of 41°F (to maintain freshness) and a minimum of 33°F core product temperature, with the most desirable average temperature close to 38°F. The rail is required to be refrigerated to maintain food product between 33°F and 41°F. Maintaining uniform bin temperatures in prep tables can be challenging. Cooling is provided by circulation of cold air under bins. Cooled air may not reach each bin evenly, which can cause non-uniform bin temperatures.

Prep tables typically have self-contained refrigeration systems. Primary refrigeration system components include:

- Compressor - Typically, conventional reciprocating refrigeration compressors are used, with capacities ranging from 1/5 to 3/4 hp, depending on prep table size
- Condenser - Conventional air-cooled fin-tube
- Expansion device - Capillary tubes are used in prep tables.
- Evaporator - Typically the design consists of copper tubing with aluminum fins.
- Refrigerant piping
- Refrigerant - HFC-134a and HFC-404A are the primary refrigerants used in most prep tables today

3.3.2 Energy Consumption

Since there are currently no government efficiency standards for preparation tables, unit energy consumptions vary significantly, as shown in **Figure 3-18**. According to the California Energy Commission appliances database (CEC 2008), buffet tables generally include less than 10 ft^3 in storage capacity with roughly 1,800 kWh in annual energy consumption, while preparation and worktop tables can vary from less than 5 ft^3 to greater than 40 ft^3 in capacity and up to 6,600 kWh in annual energy consumption.



Source: CEC 2008

Figure 3-18: **Refrigerated Food Service Equip. Energy Consumption vs. Capacity (all equipment types)**

In our energy savings and economic analysis, we analyze a prototypical prep table, described below in **Table 3-20** and **Table 3-21**, and consider the results to be applicable to all three types of food service equipment. **Table 3-20** lists the compressor description and the typical temperatures of our baseline prep table. A summary of the electrical energy consumption of a prep table is shown in **Table 3-21**. The total unit energy consumption of a typical new prep table is 2,341 kWh/yr. This estimate is just over 10 percent below the average tested energy consumption of eight prep tables tested by food service technology center in 2003 shown in **Table 3-22**, which averaged 2,660 kWh/yr (Fisher Nickel 2003).

Table 3-20: Compressor Description and Operating Temperatures (Baseline Prep Table)

Typical Compressor in 11 ft^3 Prep Table	
Type	Hermetic
Horsepower	1/3
Capacity (Btu/h)[1]	1,200
Power Draw (W)[2]	216

Typical Temperatures (°F)	
Cabinet	0
Evaporator	-20
Ambient	90
Condensing	110

Source: Adapted from Reach-in specifications (section 3.4.2)
[1] 40 percent of the capacity for a 48 ft^3 reach-in refrigerator
[2] Based on a COP of 1.63 (20% reduction from reach-in refrigerator COP of 2.04)

Table 3-21: Electrical Energy Consumption Breakdown (Baseline Prep Table)

Component	Power Consumption (W)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)
Compressor	216	66%	1,246	53%
Evaporator Fans	45	100%	394	17%
Condenser Fan	45	66%	260	11%
Anti-Sweat Heaters	49	100%	427	18%
Lighting	50	3%	14	1%
Total	-	-	**2,341**	100%

Source: Adapted from the reach-in refrigerator ADL 1996
[1] 1/3 hp compressor nominal power draw; actual compressor power draw varies.
[2] Duty cycle at 70°F ambient temperature is reduced by 10% from ADL 1996
[3] Condenser fan cycles with the compressor.
[4] There are 3 W of anti-sweat heaters per linear foot of door perimeter (27" x 58").
[5] One incandescent 25 W light bulb operates when door is open (0.5 to 1 hour per day).

Table 3-22: Prep Table Characteristics

Test Unit	Dimensions (in) (W x D x H)	Refrigerated Storage Volume (ft^3)	Power Draw (W) (Lid Up/Lid Down[1])	Unit Energy Consumption (kWh/yr)
1	44 x 20 x 17.5	8.9	208 / 151	1,717
2	26 x 25.5 x 17.5	6.7	376 / 298	2,912
3	45 x 27 x 26	18.3	268 / 167	1,882
4	41.5 x 17 x 22	9.0	463 / 232	3,004
5	44.5 x 24.5 x 24.5	15.5	304 / 255	2,416
6	44 x 24 x 20	12.2	423 / 413	3,611
7	41.5 x 17 x 22	9.0	334 / 324	2,842
8	26 x 25.5 x 17.5	6.7	386 / 281	2,880
Average		**10.8**		**2,660**

Source: Fisher-Nickel 2003
[1] Lid refers to the cover over the bins on the refrigerated rail.

3.3.3 Purchase and Installation Costs

The refrigerated food service equipment market is very competitive with many manufacturers, resulting in little price increase over the past few years. **Table 3-23** lists typical 2008 equipment and installation costs for these refrigerators. This report designates the preparation table to be the baseline unit for the energy savings analysis.

Table 3-23: Refrigerated Food Service Equipment Total Installed Costs ($2008)

	Preparation Table	Worktop Table	Buffet Table
Equipment Purchase Price	$1,200-$3,800	$1,600-$4,300	$1,700-$2,900
Installation	$150	$150	$150
Total Installed	$1,350-$3,950	$1,750-$4,450	$1,850-$3,050

Source: List prices from refrigerated food service equipment manufacturer websites were converted to equipment purchase prices using the markups described in 0. Installation cost estimated by NCI.

3.3.4 Life, Reliability, and Maintenance Characteristics

According to the North American Food Equipment Manufacturers (NAFEM), the typical life of a reach-in is 8 to 10 years (see discussion in Section 4.4 below). Refrigerated food service equipment is estimated to have the comparable life expectancy to reach-in refrigerators because of their similar construction.

Recommended refrigeration system maintenance includes cleaning the condenser coil to keep it clear of debris and dust.

3.3.5 Major Manufacturers

Major refrigerated food service equipment manufacturers include:

- True Manufacturing Company
- Delfield Refrigeration
- Continental Refrigerator
- McCall Refrigeration
- Victory Refrigeration
- Beverage-Air

3.3.6 Major End-Users

Refrigerated food service equipment is used in full-service restaurants, fast-food restaurants, and institutional foodservice establishments in buildings such as hospitals, schools, and office buildings. The largest end-users are large fast-food chains. The market for these equipment types is very fragmented due to the large variety and number of restaurants and other users.

3.4 Reach-Ins

3.4.1 Equipment Description and Illustrations

Reach-In Freezers

Reach-in freezers store frozen food products for commercial and institutional foodservice establishments. This section describes the characteristics of a baseline reach-in freezer, which are assumed to be typical of a new unit today.

Figure 3-19 shows a typical one-door reach-in freezer. Its storage capacity is about 24 ft^3. The case is typically insulated to an R-value between 15 and 20 with 2 to 2.5 inches of blown polyurethane foam. The unit typically stands on four casters (as shown in Figure 3-19) or adjustable legs. The cabinet and the doors are usually stainless steel. Anti-sweat heaters located along the door perimeter prevent condensation and frosting on the gasket and cabinet frame regions adjacent to the gasket. There is one incandescent light (typically 25W) inside the freezer that operates when the freezer door is open. The evaporator coil has an electric defrost heater with about 600W capacity that utilizes timed initiation and temperature termination control. This control scheme involves initiation of defrost based on a time schedule, and termination of defrost when the evaporator has reached a temperature indicating that frost has melted.

Source: Beverage-Air
Figure 3-19: One-Door Solid-door Reach-In Freezer

Table 3-24 summarizes the physical characteristics of the prototypical baseline reach-in freezer. The refrigeration system is located at the top of the unit. This configuration keeps refrigeration components away from spills and other debris unique to foodservice establishments, and reduces accumulation of dust on the condenser.

Table 3-24: Baseline Refrigerated cabinet description – Reach-In Freezers

Exterior Dimensions			Interior Dimensions			Insulation		Shelves		Door
W (in)	D (in)	H (in)	W (in)	D (in)	H (in)	Thickness (in)	R-Value/in (ft^2°F/Btu h)	#	Tot. Shelf Space(ft^2)	Type
30	32	83	25	27	58	2-2.5	6.5-7	3	15	Stainless Steel
Source: ADL 1996, confirmed with a survey of product literature found on major manufacturer websites										

The refrigeration system components of the baseline reach-in freezer consist of a ½ hp hermetic compressor, one 9-Watt shaded pole evaporator fan and one 1/20 hp PSC condenser fan. Refrigerant flow is governed by a thermostatic expansion valve. Most units manufactured today use either shaded pole or permanent split capacitor fan motors. Reach-in freezers typically use R-404A, as do most low temperature self-contained equipment.

Some models are available with hot gas defrost systems as an alternative to electric defrost. There is some concern in the industry, however, regarding the possibility of leaks due to thermal stresses caused by hot gas defrost. A few units are available with high-pressure gas or liquid anti-sweat heaters.

The reach-in freezer refrigeration circuit is shown in the typical self-contained system refrigeration circuit shown in **Figure 2-1**.

Reach-In Refrigerators

Reach-in refrigerators hold refrigerated food products for commercial and institutional foodservice establishments. This section describes the characteristics of a baseline reach-in refrigerator, which are assumed to be typical of a new unit today.

Figure 3-20 (a) shows a baseline two-door reach-in refrigerator. Its capacity is about 48 ft^3. The case is typically insulated to an R-value between 15 and 20 with 2 to 2.5 inches of blown polyurethane foam. The unit typically stands on four casters or adjustable legs. Cabinets and doors are usually stainless steel. Anti-sweat heaters are installed along the door perimeter to prevent condensation on and near the door gasket. There are two incandescent lights (usually 25W each) inside the refrigerator that operate when either refrigerator door is open. The refrigeration system is located at the top of the unit. This keeps refrigeration components away from spills and other debris, and reduces accumulation of dust on the condenser, while also keeping those components readily accessible for maintenance and servicing.

a. 2-Door Solid-door b. 1-Door Glass-door c. 3-Door Solid-door

Sources: Delfield (a, b), Beverage-Air (c)

Figure 3-20: Reach-in Refrigerators

Table 3-25 summarizes the physical characteristics of the baseline two-door reach-in refrigerator.

Table 3-25: Refrigerated Cabinet Description – Reach-in refrigerators

Exterior Dimensions			Interior Dimensions			Insulation		Shelves		Door
W (in)	D (in)	H (in)	W (in)	D (in)	H (in)	Thickness (in)	R-Value/in (ft^2°F/Btu h)	#	Tot. Shelf Space(ft^2)	Type
52	32	83	48	27	58	2-2.5	6.5-7	6	30	Stainless Steel

Source: ADL 1996

The baseline refrigeration system components consist of a 1/3 hp hermetic compressor, two 9-Watt PSC evaporator fans and one 1/20 hp PSC condenser fan. Refrigerant flow is governed by a capillary tube expansion device. Most units manufactured today use either shaded pole or permanent split capacitor fan motors. Like most medium temperature self-contained equipment, reach-in refrigerators typically use HFC-134a refrigerant.

Although reach-in refrigerators typically do not use defrost heaters, they do make use of off-cycle defrost to prevent build-up of frost on the evaporator. During an off-cycle, the evaporator fan continues to run to assure adequate evaporation of moisture to keep internal humidity levels appropriate for such foods as produce.

The reach-in refrigerator refrigeration circuit is well represented by the typical self-contained system refrigeration circuit shown in **Figure 2-1**.

Reach-In Refrigerator-Freezers
Reach-in refrigerator-freezers hold refrigerated and frozen food products for commercial and institutional foodservice establishments.

Figure 3-21 shows a typical two-door reach-in refrigerator-freezer. Its capacity is about 48 ft^3. The case is typically insulated to an R-value between 15 and 20 with 2 to 2.5 inches of blown polyurethane foam. Cabinets and doors are usually stainless steel. Anti-sweat heaters are installed along the door perimeter to prevent condensation on and near the door gasket. There are two incandescent lights (usually 25W each) inside the refrigerator that operate when either refrigerator door is open.

Source: Traulsen

Figure 3-21: Two-door, Solid-door Reach-in Refrigerator-Freezer

Refrigerator-Freezers typically have separate refrigeration systems serving the freezer and refrigerator compartments. They use R-404A in for the freezer compartment and R-134a for the refrigerator compartment. Most units manufactured today use either shaded pole or permanent split capacitor fan motors. Standard operating temperatures are 34 to 38°F for the refrigerator compartment and -5 to 0°F for the freezer compartment.

3.4.2 Energy Consumption

Reach-in Freezers

Characterization of the energy consumption breakdown for the baseline reach-in freezer is based on the one-door unit because it is the most common unit.

The annual unit energy consumption (UEC) of a typical reach-in freezer was estimated using the data found in the California Energy Commission Appliances Database (CEC 2008). The CEC Appliances Database contains all appliances, including reach-in freezers, currently certified to the CEC by their manufacturers as meeting currently-applicable efficiency standards (federal efficiency standards, in the case of reach-ins). The database lists the physical characteristics and the daily unit energy consumption as tested by the manufacturer. To estimate the typical UEC, we considered the UEC of all freezer models between 20 and 28 ft^3, the most common size for reach-in freezers. From that subset, we then found the average UEC by manufacturer and took an average of those values weighted by manufacturer market share. **Table 3-26** shows the data used for these calculations.

Table 3-26: Derivation of Typical Reach-in Freezer Unit Energy Consumption

Manufacturer	# of Units with volume 20 to 28 ft^3	2007 Market Share (%)[1]	Average Unit Energy Consumption (kWh/yr)[2]
True	3	42	4,110
Traulsen	8	9	3,500
Victory	7	9	3,200
Bev Air	13	6	3,050
Delfield	14	5	2,790
Continental	4	4	3,570
Northland	1	1	2,380
Others	22	24	3,210
Average (weighted by market share)			**3,590**
Sources: 1. Appliance Magazine 2008 2. CEC 2008			

Table 3-27 summarizes the performance data for the compressor and associated design temperature data. A common evaporator temperature is -20°F and the condenser temperature is about 20°F above ambient. The compressor efficiency at the listed condition is 56%.[11] This is comparable to the efficiencies achieved by good residential refrigerator/freezer compressors.

Table 3-27: Compressor Description and Operating Temperatures (Baseline Reach-In Freezer)

Typical Compressor in One-Door Reach-in Freezer	
Type	Hermetic
Horsepower	½
Capacity (Btu/h)	2200
Power Draw (W)[1]	470

Source: Adapted from ADL 1996
[1]Power draw reduced by 10% compared to ADL 1996

Typical Temperatures (°F)	
Cabinet	0
Evaporator	-20
Ambient	90
Condensing	110

Source: ADL 1996

The energy consumption of the freezer is shown in Table 3-28. The compressor duty cycle is somewhat higher than a comparison of load and compressor capacity (Table 3-29) would suggest. Additional load is due to frequent door openings and some pull-down of food placed in the unit.

[11] *Efficiency* of a compressor is defined as the ideal power input for isentropic adiabatic compression divided by the actual power input. The coefficient of performance (COP) refers to the cooling load (W) divided by the actual power input (W). A freezer can have a higher compressor efficiency while still having a lower COP than a refrigerators because the temperature lift between evaporating and condensing temperatures is considerably higher for a freezer.

Table 3-28: Electrical Energy Consumption Breakdown (Baseline Reach-In Freezer)

Component	Power Consumption (W)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)
Compressor	470[1]	65[2]	2,670	67.5
Evaporator Fans	21	100	184	4.5
Condenser Fan	70	65[2, 3]	398.5	10
Anti-Sweat Heaters	43[4]	100	372	9.5
Electric Defrost	600	6.25	328.5	8
Lighting	25	3.125[5]	7	0.5
Total			**3,960**	**100.0**

Source: Adapted from ADL 1996
[1] 1/3 hp compressor nominal power draw; actual compressor power draw varies.
[2] Duty cycle at 70°F ambient temperature is reduced by 10% from ADL 1996
[3] Condenser fan cycles with the compressor.
[4] There are 3 W of anti-sweat heaters per linear foot of door perimeter (27” x 58”).
[5] One incandescent 25 W light bulb operates when door is open (0.5 to 1 hour per day).
Note: Energy consumption values have been rounded to the nearest whole number, and therefore the total may not exactly equal the sum of the energy use values for each component.

Table 3-29 shows the typical load breakdown for the baseline reach-in freezer.

Table 3-29: Thermal Load Breakdown (Reach-in Freezers)

Component	Thermal Load (Btu/h)
Evaporator Fans	68
Lighting	3
Infiltration	41
Wall Losses	329
Defrost	128
Anti-sweat Heating	73*
Total	**642**
Compressor Capacity	2200

Source: Adapted from ADL 1996
*Includes an estimated 50% of anti-sweat consumption that contributes to the cabinet load.

Reach-In Refrigerators
To characterize the energy consumption breakdown for the baseline reach-in refrigerator, we chose the two-door unit since it is the most common unit currently used.

The same method of calculation of annual UEC that was used for reach-in freezers above was used for reach-in refrigerators, but in this case, we considered refrigerator models between 42 and 56 ft^3, the most common size for reach-in refrigerators. **Table 3-30** shows the data used for the refrigerator calculations.

Table 3-30: Derivation of Typical Reach-in Refrigerator Unit Energy Consumption

Manufacturer	# of Units in Database with Volume 42 to 56 ft^3	2007 Market Share (%)[1]	Average Unit Energy Consumption (kWh/day)[2]
True	9	42	2,320
Traulsen	20	9	2,510
Victory	12	9	2,200
Bev Air	13	6	2,310
Delfield	28	5	1,930
Continental	4	4	1,650
Others	93	25	2,380
Average (weighted by market share)			**2,270**
Sources: 1. Appliance Magazine 2008 2. CEC 2008			

The annual energy consumption of a typical reach-in refrigerator is estimated using the California Energy Commission Appliances Database (CEC 2008). The average unit energy consumption of current models is assumed to be 2,270 kWh per year.

Table 3-31 summarizes the characteristics of the compressor and the associated design temperatures. The evaporator temperature is usually about 20°F and the condenser temperature typically runs about 20°F above ambient air temperature. The compressor efficiency at the listed condition is 48%. This compares with efficiencies in the high 50's, which are achieved with good residential refrigerator compressors. There has historically been little incentive to develop high efficiency compressors for commercial refrigeration due to the lack of stringent federal standards until recently, the far lower sales volumes of these compressors compared to the residential market, and the perceived need for instant restart in commercial applications due to the severe duty cycle that results from frequent door openings. For instant restart, the starting circuit must be much more powerful than is necessary for residential compressors.

Table 3-31: Component Description (Baseline Reach-in Refrigerator)

Typical Compressor in 2-Dr Reach-in Refrigerator	
Type	Hermetic
Horsepower	1/3
Capacity (Btu/h)	2700
Power Draw (W)	337

Typical Temperatures (°F)	
Cabinet	40
Evaporator	20
Ambient	90
Condensing	110

Source: ADL 1996

The energy consumption of the refrigerator is shown in Table 3-32 below.

Table 3-32: Electrical Energy Consumption Breakdown (Baseline Reach-in Refrigerators)

Component	Power Consumption, (W)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption, (%)
Compressor	337[1]	50[2]	1,477	60%
Evaporator Fans	42	100	368	15%
Condenser Fan	42	50[2, 3]	184	7%
Anti-Sweat Heaters	50[4]	100	434	17%
Lighting	50 (2x25)	3.125[5]	14	1%
Total			**2,477**	**100%**

Source: Adapted from ADL 1996
[1] 1/3 hp compressor nominal power draw; actual compressor power draw varies.
[2] Duty cycle at 70°F ambient temperature is reduced by 10% from ADL 1996.
[3] Condenser fan cycles with the compressor.
[4] There are 1.75 W of anti-sweat heaters per linear foot of door perimeter (2 doors x 27” x 58”). Fifty percent reduction from ADL 1996.
[5] Two incandescent 25 W lights operate when either refrigerator door is open (0.5 to 1 hour per day).
Note: Energy consumption values have been rounded to the nearest whole number, and therefore the total may not exactly equal the sum of the energy use values for each component.

Table 3-33 below shows the typical load breakdown for the reach-in refrigerator. The compressor capacity is much higher than the steady state load so that it can handle times of frequent door openings and also provide quick pull down of warm food.

Table 3-33: Thermal Load Breakdown (Reach-in Refrigerators)

Component	Thermal Load (Btu/h)
Evaporator Fans	164
Lighting	5
Infiltration	62
Wall Losses	265
Anti-sweat Heating[1]	84*
Total	**580**
Compressor Capacity[2]	2700

Source: Adapted from ADL 1996

[1] It is estimated that 50% of anti-sweat consumption contributes to the case load.

[2] Compressor capacity is reduced by 10% from ADL1996.

3.4.3 Purchase and Installation Costs

Table 3-34 shows purchase prices for a one-door reach-in freezer, a two-door reach-in refrigerator, and a two-door reach-in refrigerator-freezer from one major manufacturer. The prices are used to calculate equipment purchase price in **Table 3-35**, which shows total installed cost for reach-ins. Installation for reach-ins requires machine delivery and placement. The one-door reach-in freezer and the two-door reach-in refrigerator are designated to be the baseline models for the energy-savings analysis.

Table 3-34: Reach-In List Prices

Product Type	Model Number	Door Type	# of Doors	Volume (ft^3)	List Price ($)
Freezer	EF24-1AS	Solid	1	23.1	$6,132
Refrigerator	ER48-1AS	Solid	2	46.6	$6,736
Refrigerator	ER48-1AG	Glass	2	46.6	$8,156
Dual-Temp	PRF24-24-1AS02	Solid	2	23.1/23.1	$14,423
Source: Beverage-Air 2008b					

Table 3-35: Reach-In Total Installed Cost ($2008)

Costs	Reach-In Freezer (1 Door)	Reach-In Refrigerator (2 Doors)	Reach-In Refrigerator Freezer (2 Doors)
Equipment Purchase Price	$5,100	$5,600	$12,100
Installation Cost	$150	$150	$150
Total Installed Cost	$5,250	$5,750	$12,250
Source: List prices from Beverage-Air online catalog were converted to equipment purchase prices using the markups described in 0. Installation cost estimated by NCI. Note: Costs are for solid-door models.			

3.4.4 Life, Reliability, and Maintenance Characteristics

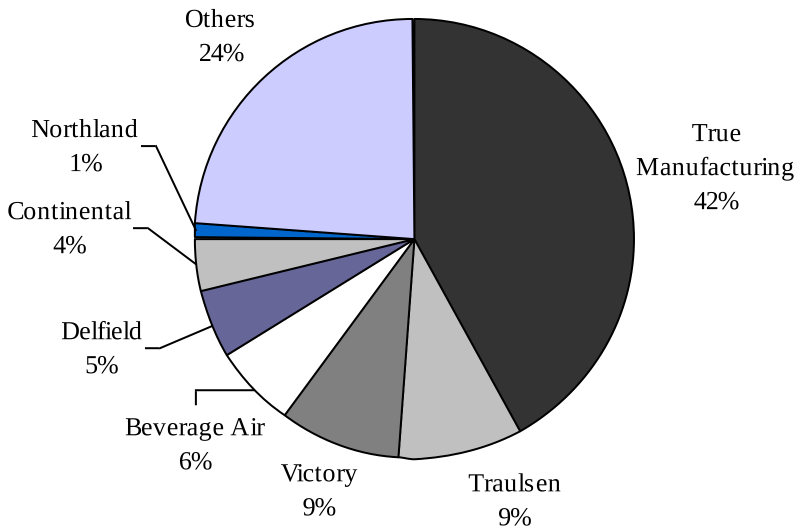
The typical life of a reach-in is 8 to 10 years. According to the North American Food Equipment Manufacturers (NAFEM), roughly 50% of units purchased by restaurants are of used equipment.

Typical regular maintenance requirements are to keep the condenser coil clear of debris and dust. Generally, however, maintenance is only done when there is a problem.

3.4.5 Major Manufacturers

Figure 3-22 shows market share figures for reach-in manufacturers. Reach-ins are typically marketed as "standard-line" or "specification-line". Standard-line reach-ins, which represent about 70% of 200,000 annual sales, are sold primarily to commercial food establishments. Specification-line reach-ins, which represent the remaining 30% of sales, are sold to institutional foodservice establishments. There are differences between the two in cosmetics and durability, but not necessarily in energy consumption.

Within the standard-line market, True Manufacturing is the dominant player, followed by Beverage-Air. Within the specification-line, Traulsen has the largest market share. Hobart has historically been another strong player and has recently established a partnership with Traulsen. Both companies currently sell their reach-ins under the Traulsen brand. Victory also maintains a large share of the spec-line market.



Source: Appliance Magazine 2008

Figure 3-22: Manufacturer Market Shares for Commercial Refrigerators as of 2007

3.4.6 Major End-Users

Reach-ins are used in full-service restaurants, fast-food restaurants, and institutional foodservice establishments in buildings such as hospitals, schools, and office buildings. The largest end-users are large fast-food chains. The market for reach-in refrigerators and freezers is very fragmented due to the large variety and number of restaurants and other users.

3.5 Beverage Merchandisers

3.5.1 Equipment Description and Illustrations

Beverage merchandisers hold and display cold beverages (canned or bottled) for self-service sales in convenience stores, supermarkets, retail stores, and small foodservice establishments.

Since beverage merchandisers are evaluated primarily on sales enhancement, they must:

- maintain a cold beverage temperature (~ 35°F)
- make the beverages look appealing (with bright lighting, logo, etc.)
- hold a high volume of beverages (~ 900 12 oz. cans maximum without special organizers)

Bottling companies (Coca-Cola, Pepsi, etc.) who sell their products through these merchandisers provide refrigeration system performance specifications (pull-down time, holding temperature, etc.) and merchandiser aesthetics specifications to the manufacturer. **Figure 3-23** shows the physical characteristics of a typical one-door beverage merchandiser. Its capacity is about 27 ft^3. The case is typically insulated to R-11.5 with 1.5 inches of blown polyurethane foam. Doors on most merchandisers in the United States are made of triple-pane, insulated glass. Double-pane glass with a low-emissivity coating, a less efficient alternative to triple-pane glass, is used internationally. Some manufacturers also supply open beverage merchandisers (i.e. without a door). They represent approximately 5% of the inventory (Beverage-Air 2008a).

Source: Beverage Air 2008

Figure 3-23: Beverage Merchandiser - Equipment Illustration

In the past most beverage merchandisers were equipped with T-12 fluorescent lighting (1 1/2”- diameter lamps) to illuminate the beverages and the logo. Today the major manufacturers have switched to more-efficient T-8 lighting and electronic ballasts (Beverage-Air 2008a). A 20-watt lamp is usually used for the logo. Either a 20-watt or a 31.5-watt lamp is used for product illumination. A single ballast can operate both lamps.

Table 3-36 summarizes the physical characteristics of the prototypical beverage merchandiser.

Table 3-36: Physical Characteristics of Baseline Beverage Merchandiser

Overall Exterior Dimensions			Overall Interior Dimensions			Insulation		Shelves		Doors
W (in.)	D (in.)	H (in.)	W (in.)	D (in.)	H (in.)	Thickness (in.)	R-Value per inch ($ft^2 \cdot °F/Btuh$)	#	Total Shelf Space (ft^2)	Type
30	35*	78	27	28.5	61.75	1.5	7.7	4	19	Triple-pane insulated glass

Sources: Beverage-Air 2008a, Beverage-Air 2008b, True Manufacturing 2008.
* Main cabinet exterior depth is 32”. Additional depth is due to the handle.

The refrigeration system is commonly located near the bottom of the unit. This allows the beverages to be displayed at the proper height above the floor and leaves room for a brand-identified logo near the top of the unit. However, a top-mounted system provides easy access for maintenance and servicing.

The baseline refrigeration system components consist of a 1/3 hp hermetic compressor, two PSC evaporator fans and one PSC condenser fan. Both fan types typically have 9 Watts of output power, which draw 21W per PSC motor. Refrigerant flow is governed by a capillary tube expansion device. Sizing of the capillary tube is generally a compromise between the need for rapid pull-down of temperature when beverages are warm and the reduced thermal load associated with steady-state conditions. To compensate for this variation in load, many beverage merchandisers have a refrigerant charge that is suitable for pull-down conditions, using a suction accumulator to store the excess refrigerant during times when the interior is cool. The excess refrigerant passes to the condenser during pull-down, which increases mass flow through the capillary tube by increasing high-side pressure and condenser-exit sub-cooling. All fans are equipped with shaded-pole motors. Nearly all units are manufactured using HFC-134a refrigerant (Beverage-Air 2008a).

3.5.2 Energy Consumption

To characterize the energy consumption breakdown for the baseline beverage merchandiser, we chose the one-door unit since it is the most common unit currently used. Table 4-37 summarizes typical compressor performance for a compressor at the high end of the efficiency range for this type of equipment. The evaporator temperature is 20°F and the condenser temperature is typically 20°F higher than ambient. The compressor efficiency at the rating point condition is 48%.[12] This compares with efficiencies in the mid 50's, which are achieved with good residential refrigerator compressors.

Table 3-37: Baseline Beverage Merchandiser - Compressor Data

Compressor				Temperatures			
Type	HP	Capacity (Btuh)[1]	Power (W)[1]	Cabinet (°F)	Evaporator (°F)	Ambient (°F)	Condensing (°F)
Reciprocating Hermetic	1/3	2,250	350	35	20	100	120

Source: Beverage-Air 2008b, Tecumseh 2008, Adapted from ADL 1996
[1]Compressor capacity and power reduced by 10% from ADL 1996.

Table 3-38 provides estimated refrigeration load breakdowns for the unit. Total average load is significantly lower than the refrigeration system cooling capacity to allow for quick temperature pull-down.

[12] This efficiency is equal to the power for ideal adiabatic compression divided by the actual power.

Table 3-38: Beverage Merchandiser Load Breakdown

	Thermal Load (Btu/h)
Evaporator Fans	143
Lighting	337
Infiltration	125
Wall Losses	204
Door Losses	50
Total	**859**
Cooling Capacity	2,500

Source: Adapted from ADL 1996

Table 3-39 shows the energy usage breakdown for the baseline one-door beverage merchandiser.

Table 3-39: Energy Usage Breakdown for Baseline Beverage Merchandiser (One-Door)

Component	Power Consumption (W)	Duty Cycle (%)	Unit Energy Consumption (kWh/yr)[5]	Energy Consumption (%)
Compressor	350[1]	40%[3]	1,221	48%
Evaporator Fans (2)	42	100%	368	15%
Condenser Fan	21	40%[3,4]	74	3%
Lighting	99[2]	100%	864	34%
Totals			**2,527**	**100%**

Source: Adapted from ADL 1996
[1]1/3 hp compressor nominal power draw. Actual compressor power draw varies. Reduced by 10% from ADL 1996.
[2]Reduced by 10 percent from ADL 1996.
[3]Estimated duty cycle based on a 70°F ambient temperature: 30% (reduced by 5% from ADL 1996). 10% was added for pull-down.
[4]Condenser fan cycles with the compressor
[5] UEC is typical for current new models

3.5.3 Purchase and Installation Costs

Table 3-40 shows purchase prices for four one-door beverage merchandisers from two major manufacturers. The prices are used to calculate equipment purchase price in **Table 3-42**, which shows total installed cost for reach-ins. Installation for beverage merchandisers requires machine delivery and placement. The one-door beverage merchandiser is designated to be the baseline model for the energy-savings analysis.

Table 3-40: Beverage Merchandiser List Prices ($2008)

Manufacturer	Model Number	Volume (ft^3)	Overall Exterior Dimensions			List Price ($)
			W (in.)	D (in.)	H (in.)	
Beverage-Air	MT21	21	27	27.5*	78	3,012
Beverage-Air	MT27	27	30	32*	78	3,178
True Manufacturing	GDM-23	23	27	29.5	78.25	3,021
True Manufacturing	GDM-23	26	30.25	78.25	78.25	3,139
Source: Beverage-Air and True Manufacturing online catalogs						

Table 3-41: Beverage Merchandiser Total Installed Cost ($2008)

Costs	Beverage Merchandiser (1-Door)	Reach-In Refrigerator (2 Doors)	Reach-In Refrigerator Freezer (2 Doors)
Equipment Purchase Price	$2,600	$5,600	$12,100
Installation Cost	$150	$150	$150
Total Installed Cost	$2,750	$5,750	$12,250
Source: List prices from Beverage-Air and True Manufacturing online catalog were converted to equipment purchase prices using the markups described in 0. Installation cost estimated by NCI.			

3.5.4 Life, Reliability, and Maintenance Characteristics

The typical life of a beverage merchandiser is 7 to 10 years. Since almost all units are trade-identified, there is no significant used equipment market in the US for beverage merchandisers. Bottling companies do not want their brand identity (i.e., the logo) to be misused. After the typical lifetime of 7 to 10 years, the bottling company will choose to:

- scrap the unit for parts,
- sell the unit overseas, or
- refurbish the unit for continued use in the same or a different location.

3.5.5 Major Manufacturers

The beverage merchandiser equipment market is dominated by two manufacturers: Beverage-Air and True Manufacturing.

3.5.6 Major End-Users

Nearly all beverage merchandisers are purchased directly from manufacturers by bottling companies for use in convenience stores, supermarkets, retail stores and small foodservice establishments. Major bottlers such as Coca-Cola and Pepsi account for 85-

90% of purchases. Smaller bottlers, such as Snapple and Clearly Canadian, account for less than 10%.

3.6 Ice Machines

3.6.1 Equipment Description

A typical ice machine consists of a refrigeration system, a water supply system, a case, and insulation. We assume a 500-lb air-cooled ice-making head that produces cube-type ice to be a representative baseline unit for the energy-savings analysis. Refer to section 0 for a definition of the three ice machine types and the three ice types. This section provides a general description of ice-making equipment.

Ice-making heads are the most versatile of the three types and are available in the widest range of capacities. They are generally mounted on top of a separately sold storage bin. Remote-condensing units are similar to IMHs, except that they have a remote condenser usually located outdoors, rejecting heat directly to the outside air without heating the air inside the building. Self-contained units are generally manufactured with smaller capacities than IMHs and RCUs.

Storage bins generally have a full-width door for user access to the ice. Ice production capacity can be expanded for non-self-contained machines either by stacking an additional machine on top of the first machine or by positioning a second machine adjacent to the first machine on top of a single ice storage bin. When the machines are stacked, ice produced in the upper machine falls through the lower machine into the storage bin. Drains must be provided for removal of the excess water from the ice machine and melting of the ice in the bin.

In ice cube machines, the ice is formed by passing a film of water over the evaporator, allowing some of the water to freeze, while some of the water falls to the sump and is recirculated with a pump. The ice layer gradually grows. Because not all the water freezes, solids and gases dissolved in the water are not captured in the ice layer and are instead carried away by the remaining liquid, which is drained from the sump when the ice cube batch is complete. This process allows production of clear ice with ice cube machines. Typical potable water consumption ranges from 15 - 40 gallons/100 lb of ice produced, compared to 12 gallons/100 lb. contained in the ice produced.

Flake and nugget machines do not produce clear ice and hence do not require water purging--the resulting ice retains the impurities, similar to ice production in residential refrigerators.

An ice machine consists of two major subsystems: the refrigeration system and water supply/ circulation/purge system. All commercially available ice machines use vapor compression refrigeration to produce the refrigeration needed for ice production. Ice machines use either air-cooled or water-cooled condensers. About 80 percent of ice machines have air-cooled condensers. Although water-cooled condensers increase energy efficiency slightly, they result in much higher water consumption since condenser water is most often drained after it is used to cool the condenser. (Communication with a major manufacturer in May 2008)

Primary refrigeration system components include:

- Compressor - Typically, conventional reciprocating compressors are used, with capacities of 1/3 to 3 hp, depending on ice machine capacity.
- Condenser - Conventional air-cooled fin-tube or water-cooled concentric tube heat exchangers are used. Air-cooled condensers are designed such that condensing temperatures are 20 - 25°F above the ambient temperature. Water-cooled condensers are controlled to maintain a constant preset condensing temperature by varying the water flow rate.
- Expansion device - Both thermostatic expansion valves and capillary tubes are used in ice machines.
- Evaporator - Typically the design consists of copper tubing attached to copper or stainless steel ice making surfaces.
- Liquid line/suction line heat exchanger
- Refrigerant piping
- Hot gas bypass line (cubers only) - This directs refrigerant directly from compressor to evaporator for harvesting the ice.
- Hot-gas solenoid valve (cubers only) - This controls hot gas refrigerant flow to the condenser during ice production and to the evaporator during ice harvest.
- Refrigerant - R-404A is the primary refrigerant used in all ice machines today. R-134a and R-22 are still used in a few select models.
- May have a suction accumulator

The water system consists of the following components:

- Potable water supply connection and water supply control valve
- Water sump
- Water circulation pump
- Water circuit - plastic tubing and evaporator water distributor
- Purge drain (cube machines only)

The batch process, used by cube machines, is described in detail as follows (ADL 1996):

1. Water fills the sump. The sump usually contains 10 - 40 percent more water than required to make a given batch of ice.
2. The refrigeration system is activated and sump water is circulated over the evaporator plate. During the freeze cycle the compressor, condenser fan (for air-cooled machines) and the water circulating pump are activated.
3. The water is cooled down and gradually freezes on the evaporator plate.
4. Ice builds up on the plate till the proper ice batch weight is detected by some means: sump water level, compressor suction pressure, or thickness of ice on the plate.

5. Upon reaching the prescribed ice weight, the machine switches to the harvest mode. Most machines use hot-gas harvest, in which hot refrigerant vapor is directed from the compressor to the evaporator to warm the evaporator and melt enough ice to free the ice on the plate. Typically about 10 - 20 percent of the ice is melted during the harvest process. Once free, the ice falls by gravity and/or mechanical assistance into the storage bin below. During the harvest process the condenser fan for air-cooled machines is off and the water circulating pump may be operating, depending on the design. Some machines use a limited amount of hot gas for meltage combined with mechanical means for removing the ice.
6. During the harvest process, water remaining in the sump is purged from the system and fresh, potable water is flushed through the system to remove impurities and purged.
7. Water fills the sump and the system returns to the freeze mode as detected by evaporator temperature and/or time.

Some ice machines utilize the incoming potable water stream to assist in the harvest process by directing the incoming water behind the evaporator plate or over the ice. The water can provide more than 50 percent of the energy required for harvest resulting in reduced harvest input energy and prechills the water for the next batch of ice.

Except for the evaporator, all of the components used in the ice machine are fairly conventional refrigeration components. The evaporator is constructed of copper tubing attached to copper or stainless steel ice making surfaces. There may be plastic attached to the ice making surface to act as an insulator to promote the formation of individual cubes. Evaporator design requires finding a careful balance between the ice growth behavior, water flow rate over the evaporator, localized water distribution, materials selection, and harvest performance (e.g., successful ice detachment, amount of meltage). Evaporator design is a complex process not amenable to analysis, and developing a successful evaporator design requires many hours of laboratory testing. Manufacturers are very reluctant to make changes to the evaporator design once a successful design has been developed.

Manufacturers generally produce one or at most two evaporator sizes, which are used in multiples across the product line matched with the appropriately sized compressor. This manufacturing strategy contributes to variations of energy efficiency across the product line due to the fact the evaporator/compressor combination cannot be optimized for each machine, resulting in some machines with undersized evaporators with oversized compressors to achieve the target production rate and correspondingly higher energy consumption. (Scotsman 2008)

3.6.2 Energy Consumption

Ice machine performance (capacity, energy consumption, and water consumption) is usually presented for operating conditions prescribed by the Air-Conditioning, Heating, and Refrigeration Institute (AHRI), which are for a 90°F ambient temperature, a 70°F inlet water temperature, and an inlet water pressure of 30 ± 3 psig (AHRI 2007).

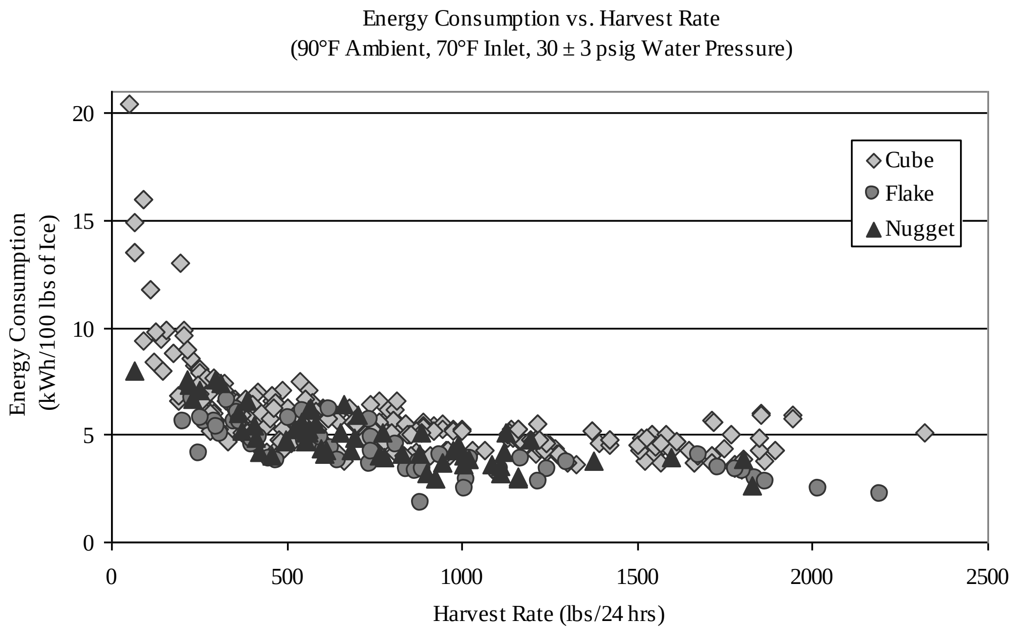
Table 3-42 lists the unit energy consumption in kWh per 100 lbs of ice broken down by equipment type, cooling type, harvest rate, and ice type. The first value in each box is the average unit energy consumption, and inside the parentheses is the range of values for each category. The data set includes all ice-machines, air- and water-cooled, currently on the market.

Table 3-42: Unit Energy Consumption by Equipment Type

Equip. Type	Condenser Cooling	Harvest Rate *lbs ice/24hrs*	Unit Energy Consumption *kWh / 100 lbs Ice* Average (Min – Max)		
			Cube	Flake	Nugget
IMH	Water	< 500	5.4 (4.2-9.9)	4.4 (3.8-5.6)	4.6 (4.2-4.9)
		≥ 500 and < 1436	4.3 (3.6-6.0)	3.5 (2.5-4.7)	3.7 (3.0-4.7)
		≥ 1436	3.9 (3.4-4.3)	2.7 (2.2-3.5)	2.7
	Air	< 450	6.8 (5.4-8.2)	5.6 (4.8-6.6)	6.2 (5.2-7.4)
		≥ 450	5.5 (4.6-7.5)	3.8 (0.6-5.8)	4.5 (3.5-6.4)
RCU w/o Remote Compressor	Air	< 1000	5.8 (4.7-7.1)	4.9 (3.4-6.2)	5.3 (4.1-6.2)
		≥ 1000	4.9 (4.2-6.0)	3.4 (2.8-4.0)	3.8 (3.6 – 4.2)
RCU w/ Remote Compressor	Air	< 934	5.8 (5.0-6.7)	4.5	4.6 (4.1-5.1)
		≥ 934	4.7 (4.3-5.0)	3.9	4.2 (3.9-4.8)
SCU	Water	< 200	7.8 (6.6-9.8)	N/A	N/A
		≥ 200	6.5 (5.7-7.3)	5.0 (4.9-5.0)	4.6 (4.0-5.1)
	Air	< 175	13.0 (8.4-20.4)	N/A	8.0
		≥ 175	9.1 (6.8-13.0)	5.5 (4.1-6.7)	6.5 (5.3-7.6)

Source: Survey of AHRI Ice Machine Directory (2008), CEC Appliances Database (2008), and ENERGY STAR Database (2009); confirmed by literature on manufacturer websites where possible

Figure 3-24 shows a plot of energy consumption versus capacity. Efficiencies tend to level out for production capacities above 500 lb/day, but can be significantly lower for lower production rates.



Source: Survey of AHRI Ice Machine Directory (2008), CEC Appliances Database (2008), and ENERGY STAR Database (2009); confirmed by literature on manufacturer websites where possible

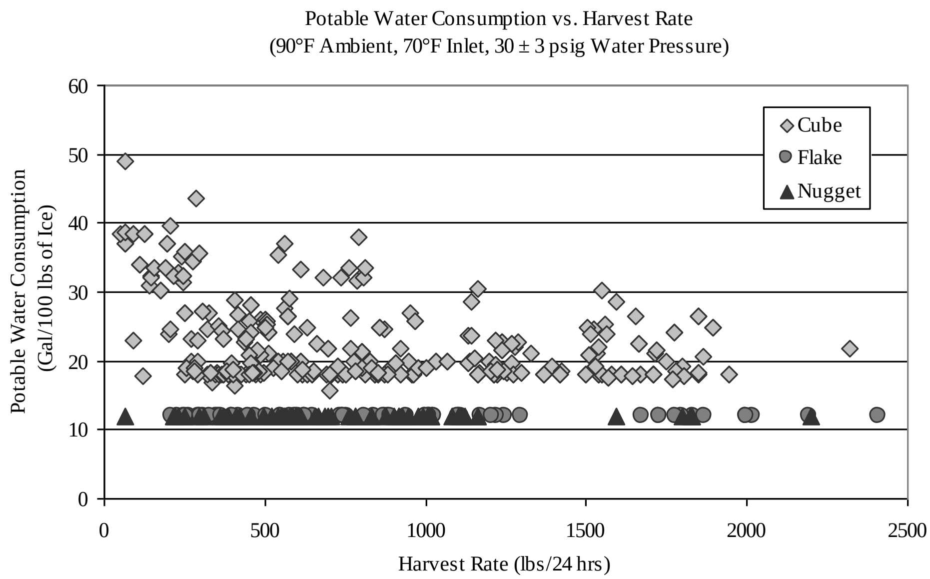
Figure 3-24: Ice Machine Energy Consumption versus Capacity (all equipment types)

The decrease of energy consumption with capacity can be attributed to several items described below: (ADL 1996)

- More efficient compressors: Nominal compressor capacities for the smaller machines are 1/2 hp and less, increasing to about 3/4 hp for machines in the 350 - 500 lb/day capacity range, further increasing to 2 hp for the machines of greater than 1000 lb/day capacity. Accompanying the increase of compressor capacity is an increase of efficiency. Compressor efficiencies for the small compressors are in the 45 - 50 percent range increasing to more than 60 percent in the larger sizes.
- Reduced ambient heat leak: Larger ice machines tend to have cold compartments which have less surface area exposed to the ambient per unit ice production and usually have better insulated cold compartments.
- Reduced water consumption: Smaller ice machines tend to have higher water consumption because manufacturers tend to use oversized sumps in the desire to maximize the use of common components.

The variation of energy efficiency over a small capacity range depends on the manufacturers' component selection and manufacturing strategies. Manufacturers try to maximize the use of common components across the product line, which include cabinets, evaporator size, and water sumps and the tradeoff of compressor efficiency level and cost. The more efficient machines tend to have larger evaporators for a given production rate, resulting in a higher evaporating temperature and higher resulting operating COP.

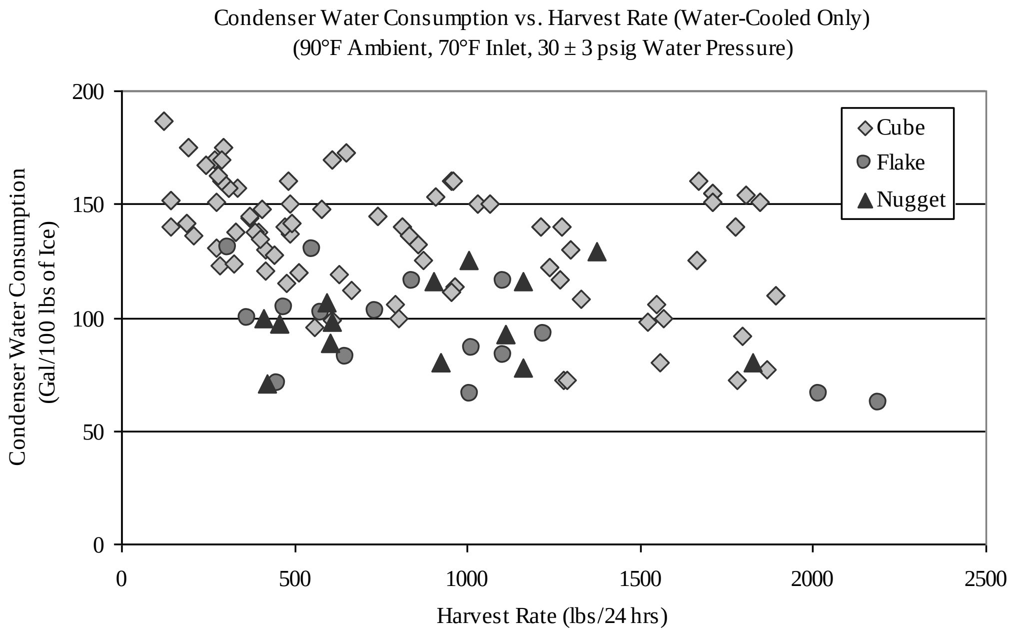
Figure 3-25 shows a plot of potable water consumption versus capacity. The minimum potable water consumption necessary to make 100 lbs of ice is 12 gallons. Flake and nugget machines convert all potable water to ice using the continuous process, and therefore only use 12 gallons/100 lbs of ice, not counting losses due to melting in the storage bin. Cube machine potable water use ranges from 15-50 gallons/100 lbs of ice, with the highest consumption rates at the lowest capacities.



Sources: Survey of AHRI Ice Machine Directory (2008), CEC Appliances Database (2008), and ENERGY STAR Database (2009); confirmed by literature on manufacturer websites where possible

Figure 3-25: Potable Water Consumption versus Capacity (all equipment types)

Figure 3-26 shows a plot of condenser water consumption versus capacity, for units with water-cooled condensers. Condenser water consumptions range from 60 – 190 gallons/100 lbs of ice. Cube machines use the greatest amount of condenser water.



Sources: Survey of AHRI Ice Machine Directory (2008), CEC Appliances Database (2008), and ENERGY STAR Database (2009); confirmed by literature on manufacturer websites where possible

Figure 3-26: Condenser Water Consumption versus Capacity (water-cooled only)

Estimated Total Energy Consumption

The annual unit energy consumption (UEC) of a typical baseline ice machine was estimated using the data found in the California Energy Commission Appliances Database (CEC 2008). To estimate the typical UEC, we considered the UEC of all air-cooled cuber ice-making heads with harvest rates between 400 and 600. From that subset, we then found the average UEC by manufacturer and took an average of those values weighted by manufacturer market share. Table 3-43 shows the data used for these calculations.

Table 3-43: Derivation of Typical Ice Machine Unit Energy Consumption

Manufacturer	# of Units with Harvest Rate > 400 and < 600 lbs/day	Market Share (%) [1]	Average Unit Energy Consumption (kWh/day)[2]
Hoshizaki	6	23%	5,805
Ice-O-Matic	4	15%	5,376
IMI Cornelius	2	4%	5,306
Manitowoc	4	35%	4,897
Scotsman	4	23%	5,227
Average (weighted by market share)			**5,270**

Source: CEC 2008
[1] Estimates from leading ice machine manufacturer
[2] Assumes 100% system duty cycle; UEC is typical of current new models.

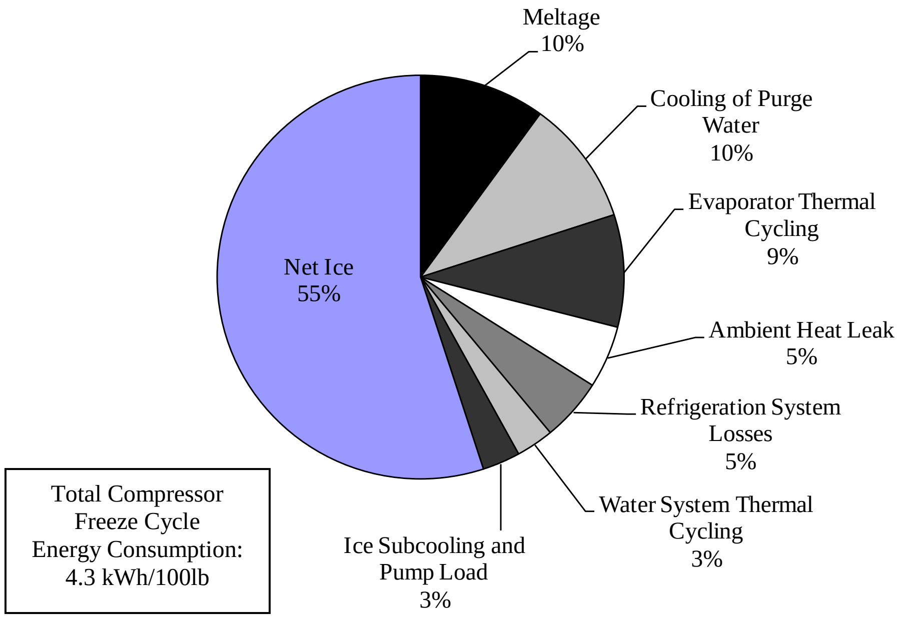
Energy Consumption Breakdown
The following discussion uses an air-cooled machine with a nominal capacity of 500 lb/day as the prototypical ice machine. Table 3-44 shows a power consumption breakdown for a typical 500 lb/day machine operating in a 70°F ambient with an inlet water temperature of 50°F (these conditions are more typical than the ARI rating conditions of 90°F ambient and 70°F water). As shown in Table 4-32, the total annual energy consumption is estimated to be about 5,250 kWh. Compressor energy consumption during the freeze cycle accounted for about 80 percent of the total while compressor energy for harvest accounted for about 9 percent of the total. The condenser fan and water pump accounted for about 8 percent and 1.6 percent of the total, respectively. The energy consumption associated with the hot-gas solenoid valve is negligible. Because the compressor accounts for about 90 percent of the total energy consumption, reducing the compressor energy consumption is the key to significant energy savings.

Table 3-44: Estimated Annual Energy Consumption by Component[1]

Component	Average Operating Power Draw (W)	Typical Duty Cycle, % (full capacity operation)	Estimated Unit Energy Consumption[2], kWh/yr (% total)
Compressor (during freezing)	1,050	90-95	4,250 (81%)
Compressor (during harvest)	1,400	5-10	460 (9 %)
Condenser Fan	110	90-95	450 (8 %)
Water Pump	20	90-100	80 (2%)
Hot-gas solenoid valve	15	5-10	5 (<1%)
Total	---	---	5,245 (100%)

Source: ADL 1996, CEC 2008
[1]For a 500 lb/day air-cooled IMH at full capacity operation; 90 deg F ambient, 70 deg F inlet water
[2]Assuming an overall annual duty cycle of 50%. UEC is typical for current new models.

Figure 3-27 shows the compressor freeze cycle energy consumption allocated among the various thermal loads. The energy required to produce net ice accounts for 55 percent of the total compressor energy input. Ice meltage during harvest and cooling of the purge water each accounts for 10 percent of the compressor freeze cycle energy consumption. Thermal cycling of the evaporator accounts for about 9 percent of the energy, the ambient heat leak about 5 percent and refrigeration system losses about 5 percent of the total energy consumption. Thermal cycling of the water system, sub-cooling of the ice, and heat input from the water pump motor accounts for 6 percent of the total compressor energy input.



Source: ADL 1996

Figure 3-27: Freeze Cycle Energy Consumption by Load[13]

Based on the results shown in Figure 3-27, reductions in energy consumption could be obtained through a combination of reducing the thermal loads to the system and refrigeration system improvements. Major parasitic thermal loads identified above include ice meltage during harvest, cooling of purge water, thermal cycling of the evaporator, and ambient heat leak, which account for over 30 percent of the compressor freeze cycle energy input. Refrigeration system improvements can be realized by utilizing the traditional methods applied to all vapor compression systems: increase compressor efficiency, reduce condensing temperature, raise average evaporating temperature, and reduce losses.

3.6.3 Purchase and Installation Costs

The ice machine market is very competitive with many producers, resulting in little price increase over the past few years. **Table 3-45** shows total installed cost for a typical baseline ice machine. Installation for ice machines requires machine delivery and placement, and connection of electric power, water supply, and drainage piping. The cost is estimated to be $500. Machines with remote condensers involve additional cost of about $100 for installation of the condenser. (ADL 1996, verified by Scotsman 2008)

[13] Source indicates that calculations are for a typical ice machine using the program FREEZE (ADL 1996). Source indicates that the model was validated by comparing with confidential test data of three manufacturer's machines

Table 3-45: Ice Machine Total Installed Cost ($2008)

Costs	Ice Machine (500lb/day, air-cooled IMH)
Equipment Purchase Price	$3,800
Installation Cost	$500
Total Installed Cost	$4,300
Source: List price from Scotsman online catalog was converted to equipment purchase price using the markups described in 0. Installation cost estimated by NCI.	

3.6.4 Life, Reliability, and Maintenance Characteristics

Reliability and first cost are important market drivers for ice machine selection. Reliability includes mechanical reliability, i.e., no component failures, and operational reliability, which means the machine produces ice of consistent shape and weight over long periods with little/no adjustments or attention. Customers are willing to pay slightly more for a particular machine if the purchase can be justified because of higher reliability.

Ice machine life is in the range of 7 - 10 years. Most manufacturers have a 5-year warranty on the refrigeration system.

Periodically (every 2 - 6 weeks) the ice machines must be cleaned to remove lime and scale and sanitized to kill bacteria and fungi. The cleaning/sanitizing process involves shutting the machine down, emptying the bin of ice, and adding cleaning/sanitizing solution to the machine. The machines are switched to a cleaning mode in which the mixture is circulated through the machine for a period of time, and then purged. The machine is switched into the ice production mode for several batches of ice to remove any residual cleaning/sanitizing solution from the machine. The machine is returned to normal operation after the ice is removed from the bin and the bin is cleaned.

Some ice machines have self-cleaning/sanitizing capabilities. These machines eliminate many of the manual steps of the cleaning/sanitizing process and can be programmed to clean/sanitize at prescribed intervals.

3.6.5 Major Manufacturers

Ice machine markets are highly fragmented due to the diversity of system types; complex distribution, sales, and service chains; and the large variety and size of food stores, food service establishments, hospitals, schools, hotels, and other users. Typically, manufacturers work through regional sales offices or manufacturers' representatives to sell equipment to equipment dealers, beverage and food distributors, or franchises. These various parties in turn sell equipment to end-users. However, manufacturers will often sell direct to large chains, cutting out the middlemen (Nadel 2002). Major ice machine manufacturers and approximate market share are summarized below.

Table 3-46: Major Ice Machine Manufacturers and Approximate Market Share

Manufacturer	Market Share (%)
Manitowoc	35%
Scotsman	23%
Hoshizaki America	23%
Ice-O-Matic	15%
IMI Cornelius	4%
Source: Estimates by a leading ice machine manufacturer	

Table 3-47 summarizes the ice machine market by manufacturer and equipment type. The brand name of the ice and/or machine is specified where applicable.

Table 3-47: Machines by Major Manufacturer and Equipment Type

Equipment Type	Ice Type	Condenser Cooling Type	Manitowoc	Ice-O-Matic	Scotsman	Hoshizaki	Cornelius
Remote Condensing w/o Remote Compressor	Cube	Air	x	x	x	x	x
	Flake	Air	x	x	x	x	x
	Nugget	Air	x	Pearl	x	Cubelet	Chunklet
Remote Condensing w/ Remote Compressor	Cube	Air	x		x		
		Water	x				
	Flake	Air	x				
	Nugget	Air	x				
Ice-making Head (IMH)	Cube	Air	x	x	x	x	x
		Water	x	x	x	x	x
	Flake	Air	x	x	x	x	x
		Water	x	x	x	x	x
	Nugget	Air	x	Pearl	x	Cubelet	Chunklet
		Water	x	Pearl	x	Cubelet	Chunklet
Self-Contained (SCU)	Cube	Air	x	x	x	x	x
		Water	x	x	x	x	x
	Flake	Air	x	x	x	x	
		Water		x	x		
	Nugget	Air	x		x	Cubelet	
		Water			x	Cubelet	

Source: Manufacturer websites

1.1.1 Major End-Users

Major classes of end-users include food service establishments (e.g., restaurants, institutional cafeterias, fast food establishments, delicatessens, and bars), food sales (e.g., supermarkets and convenience stores), hospitals, and hotels (Nadel 2002).

The largest machines are used in airline and airport food service industries, very large restaurants, stadiums, schools, and industrial processes. The smallest machines are sold to bars, restaurants, and office buildings.

3.7 Refrigerated Vending Machines

3.7.1 Equipment Description and Illustrations

Fifty-two percent of refrigerated vending machines are beverage vending machines, making beverage vending machines a good proxy for all refrigerated vending machines (Vending Times 2008a). A refrigerated packaged beverage vending machine is "a commercial refrigerator that cools bottled or canned beverages and dispenses the bottled or canned beverages on payment." (42 U.S.C. 6291(40))

Beverage vending machines must:

- achieve quick pull-down temperature
- maintain a cold beverage temperature (~ 35°F)
- attract customers (with bright lighting, attractive logo, etc.)
- hold a high volume of beverages (~ 400-800 12 oz. cans)
- be theft-resistant

Bottling companies purchase vending machines directly from the manufacturer. They will often specify all the aesthetics of the machine (light output, logo, etc.) and its refrigeration performance requirements (pull-down time, holding temperature, etc.). These performance requirements vary among the different bottling companies. Aside from these differences, however, all vending machines are similar.

As mentioned in section 2.7, refrigerated vending machines come in two configurations: fully-cooled and zone-cooled.

Fully-cooled vending machines typically have a glass or transparent polymer front. Figure 3-28 shows a typical fully-cooled packaged beverage vending machine. Its capacity is about 410 12 oz. cans. The case is typically insulated to R-8 with 1 inch of blown polyurethane foam. The unit has a clear air-filled, two-pane glass front with an aluminum frame.

The fully cooled unit is typically equipped with two 4-foot T8 fluorescent lamps (1-1/2 inch diameter tubes), on one ballast. The lighting is located within the refrigerated volume to provide the best illumination of the product. The most common lighting configuration is two 32-watt bulbs. Lighting can have a significant effect on energy use because it directly consumes electricity and indirectly increases compressor power consumption by adding heat to the cabinet load.

Source: Dixie-Narco 2008

Figure 3-28: Refrigerated Packaged Beverage Vending Machine (Fully-Cooled)

Table 3-48 summarizes the physical characteristics of the baseline fully-cooled packaged beverage vending machine.

Table 3-48: Baseline Refrigerated Beverage Vending Machine (Fully-Cooled) – Cabinet Description

External	Internal	Insulation		Capacity	Weight
Dimensions (in.)	Volume ($ft.^3$)	Thickness (in.)	R-Value per inch ($ft.^2$-°F-h/Btu)	Number of cans	Weight (lbs.)
72H x 47W x 32D	35	1	8	410	750

Source: Adapted from product literature available on the websites of major manufacturers

The refrigeration system of the fully cooled vending machine is packaged as a modular unit and is located in the bottom rear section of the vending machine. The refrigeration circuit shown in **Figure 2-1** (above) is typical for the self-contained systems found in refrigerated vending machines. The baseline refrigeration system components are assumed to consist of a 2,900 Btu/h hermetic compressor, one evaporator fan and one condenser fan, both with 6-Watt shaft-power shaded pole motors. Refrigerant flow is governed by a capillary flow expansion device. HFC-134a is used in most vending machine refrigeration systems.

Figure 3-29 shows a typical zone-cooled packaged beverage vending machine, which is assumed to be a representative baseline. Its capacity is approximately 800 12 oz. cans. The case is typically insulated to R-8 with 1 inch of blown polyurethane foam.

There are two doors on zone-cooled vending machines. An insulated inner door allows access to the refrigerated space where the cans are stored. An outer door houses the logo and its associated lighting equipment.

The zone-cooled unit is typically equipped with two 6-foot T8 fluorescent lamps (1-1/2 inch diameter tubes), on one ballast, to illuminate the logo. The lighting is located outside of the refrigerated volume. The most common lighting configuration is two 49-watt bulbs.

Source: SandenVendo 2008

Figure 3-29: Refrigerated Packaged Beverage Vending Machine (Zone-Cooled)

Table 3-49 summarizes the physical characteristics of the baseline zone-cooled packaged beverage vending machine.

Table 3-49: Baseline Refrigerated Packaged Beverage Vending Machine (Zone-Cooled) – Cabinet Description

External	Internal	Insulation		Capacity	Weight
Dimensions (in.)	Volume ($ft.^3$)	Thickness (in.)	R-Value per inch ($ft.^2$-°F-h/Btu)	Number of cans	Weight (lbs.)
79H x 40W x 35D	31	1	8	800	700

Source: Adapted from product literature available on the websites of major manufacturers

The refrigeration system is similar to a fully-cooled vending machine. However, since only a portion of the unit is refrigerated, the compressor is typically smaller (i.e., 2,400 Btu/h). The evaporating and condensing coils are also smaller than those used in fully-cooled vending machines.

3.7.2 Energy Consumption

Table 3-50 summarizes performance data for the compressor and the associated design temperature data of a baseline refrigerated vending machine. The evaporator temperature is typically about 10°F and the condenser temperature is usually about 20°F above ambient. The rating-point ambient conditions are 75°F with 45% relative humidity. The baseline compressor efficiency at this condition is 48%. This compares with efficiencies in the mid 50's, which are achieved with good residential refrigerator compressors. Residential refrigerator compressor efficiencies are higher due to the more stringent minimum efficiency standards applicable to home refrigerators. In addition, commercial equipment is designed for instant restart, necessitating a more powerful starting circuit for single phase power, which reduces the efficiency of commercial refrigeration compressors.

Table 3-50: Baseline Refrigerated Packaged Beverage Vending Machine – Refrigeration Component Description

Unit Type	Compressor				Temperatures			
-	HP	Type	Capacity (Btu/h)	Power Draw (W)	Cabinet (°F)	Evap. (°F)	Ambient (°F)	Cond. (°F)
Fully-Cooled	1/2	Hermetic	2,900	480	36	20	75	120
Zone-Cooled	1/3	Hermetic	2,400	420	39	20	75	120

Table 3-51 shows the steady state thermal load breakdowns for typical packaged beverage vending machines. The refrigeration system cooling capacity is much higher than the steady state load because of the need for quick pull-down of beverage temperatures.

Table 3-51: Packaged Beverage Vending Machine Refrigeration Load Breakdown

Component	Cooling Load (Btu/h)	
	Fully-Cooled	Zone-Cooled
Evaporator Fan	72	143
Lighting	205	0
Infiltration	164	263
Conduction	223	248
Radiation	192	0
Total	**856**	**654**

Source: DOE 2008b

Table 3-52 and Table 3-53 show the energy consumption breakdown for typical packaged beverage vending machines.

Table 3-52: Baseline Refrigerated Beverage Vending Machine (Fully-Cooled) – Energy Consumption Breakdown

Component	Power Consumption (W)	Duty Cycle (%)	Unit Energy Consumption (kWh/yr)[1]	Energy Consumption (%)
Compressor	520	35%	1,594	64%
Evaporator Fan	21	100%	184	7%
Condenser Fan	53	35%	163	7%
Lighting	63	100%	552	22%
Dispensing Mechanism	120	~ 0%	1	0%
Total	**-**	**-**	**2,494**	**100%**

Source: DOE 2008b
[1]These UECs are typical of current new models.

Table 3-53: Baseline Refrigerated Beverage Vending Machine (Zone-Cooled) – Energy Consumption Breakdown

Component	Power Consumption (W)	Duty Cycle (%)	Unit Energy Consumption (kWh/yr)[1]	Energy Consumption (%)
Compressor	447	30%	1,175	52%
Evaporator Fan	21	100%	184	8%
Condenser Fan	53	30%	139	6%
Lighting	87	100%	759	34%
Dispensing Mechanism	120	~ 0%	1	0%
Total	**-**	**-**	**2,258**	100%

Source: DOE 2008b
[1]These UECs are typical of current new models.

3.7.3 Purchase and Installation Costs

Table 3-54 shows total installed cost for a typical baseline ice machine. Installation costs, which include profit and overhead, are estimated to be $100.

Table 3-54: Refrigerated Vending Machine Total Installed Cost ($2008)

Costs	Fully Cooled RVM (410 Can capacity)	Zone-Cooled RVM (800 Can capacity)
Equipment Purchase Price	$4,300	$3,200
Installation Cost	$100	$100
Total Installed Cost	$4,400	$3,300
Source: Communication with leading manufacturers in May 2008. [1] Accurate for quantities of 100 units or less		

Bottling companies “recycle” their machines through their refurbishing centers. Some owner-operators will buy or sell these used machines. The bottling companies will strip the machine of its logo before it is brought into the used market because they do not want their brand-identity to be misused.

If a refrigeration system module fails at a time beyond its typical 5-year warranty, a replacement system costs about $400 to the manufacturer (PepsiCo 2008).

3.7.4 Life, Reliability, and Maintenance Characteristics

The typical life of a packaged beverage vending machine is about 14 years. Regular maintenance for refrigerated vending machines consists of cleaning the condenser coil and cleaning and replacing lamps when necessary. Annual maintenance is done on the machine at a cost of about $30 per year. During the life of the vending machine, it is usually refurbished about twice in a refurbishing center run by the bottling company. Packaged beverage vending machines are usually fully refurbished about once every three to five years at a cost of approximately $930.

Most manufacturers provide a packaged refrigeration system. If a service technician discovers a problem in the refrigeration loop, the old packaged refrigeration system is replaced with a new one. Most manufacturers have a 5-year warranty on the refrigeration system. Beyond the warranty, bottling companies may or may not wish to have the old system repaired.

3.7.5 Major Manufacturers

According to industry representatives, about 75% of the packaged beverage vending machine equipment market is dominated by three manufacturers (i.e., Dixie-Narco, Royal Vendors, and SandenVendo). The remaining portion of the industry is comprised of several smaller manufacturers.

3.7.6 Major End-Users

The major bottling companies that purchase packaged beverage vending machines are:

- Coca-Cola Company
- PepsiCo, Inc.
- Dr. Pepper Company
- Seven-Up Company

4 Energy Savings Potential Using Current Technologies

This section describes the energy saving potentials achievable using currently available technologies that are applicable to commercial refrigeration equipment, performs a simple economic analysis, and discusses the technical and market barriers to implementation. By "currently available" we mean that the technology has been integrated into commercially available commercial refrigeration equipment. Technologies that are not currently available are discussed in Section 5.

The list of current technologies remains similar to the 1996 Arthur D. Little report, with a few changes. LED lighting has recently become a commercially available lighting option in the commercial refrigeration applications. See section 4.1 for equations defining how the energy and cost figures were calculated for each technology option. The variables for each of the equipment types are defined in **Table 4-4** at the end of section 4.1. In general, for cases where more recent cost data were not available, we adapted the data from ADL 1996 according to several assumptions to bring the cost figures up to 2008 dollars. Specifically, we doubled retooling costs and inflated equipment costs according to the producer price index (discussed in 0).

We present the results of the economic analysis for each equipment type in the form of simple payback period. For walk-ins, reach-ins, and refrigerated vending machines, we conducted separate analyses for the two types of cabinets (i.e. refrigerator and freezer, zone-cooled and fully-cooled). For each equipment type, the table of results is followed by explanations of the origination of the energy use reduction and cost premium values.

$$PBP = \frac{Installed\ Cost\ Premium\,(\$)}{Annual\ Energy\ Savings\left(\frac{kWh}{yr}\right) \times Electricity\ Price\left(\frac{\$}{kWh}\right)}$$

The results include three scenarios, each involving a different price for electricity. **Table 4-1** lists the electricity prices used to calculate operating cost savings and payback period. These electricity prices use state average commercial retail electricity rates published by the Energy Information Administration. The low electricity rate scenario is the average of the 19 lowest-priced states weighted by state population, comprising 25 percent of the U.S. population. Likewise, the high electricity rate scenario is the average of the 10 highest-priced states weighted by state population, comprising 25 percent of the U.S. population.[14] The medium electricity rate is the average of the electricity prices from the remaining 22 states, weighted by population. See Appendix B for the raw electricity price data used to calculate the three rates.

Table 4-1: Electricity Price Scenarios

[14] The top 25 percent section includes Washington, DC, as a state.

	Electricity Price ($/kWh)
Low Electricity Rate	$0.0711
Medium Electricity Rate	$0.0993
High Electricity Rate	$0.1627
Source: EIA 2009	

4.1 Currently Available Technologies

This section includes a description of each current technology option as well as the equations and assumptions used in order to calculate the energy savings potential and installed cost premium for each technology. **Table 4-2** shows which technologies apply to which equipment types.

Table 4-2: Currently Available Energy Saving Technologies

		Refrig. Systems	Walk-ins	Service Equip.	Merchandisers	Reach-ins	Ice Machines	Vending Mach.
Insulation	Thicker Insulation	✓	✓	✓	✓	✓	✓	✓
Fans	High-Efficiency Fan Motors	✓	✓	✓	✓	✓	✓	✓
	Fan Motor Controllers	✓	✓					✓
	High-Efficiency Fan Blades	✓	✓	✓	✓	✓	✓	
Compressors	High Efficiency Compressors	✓	✓	✓	✓	✓	✓	
	Variable Capacity Compressor	✓		✓	✓	✓		✓
Heat Exchangers	Evaporator Design (Enhanced UA Evap. Coil)	✓						✓
	Condenser Design	✓						✓
	Reduced Evaporator Thermal Cycling						✓	
Doors	Advanced Door Technologies	✓						
	Low Heat Doors		✓					
	Strip Curtains		✓					
	Auto-Door Closer		✓					
Anti-Sweat Heaters	Hot Gas Anti-Sweat		✓	✓		✓		
	Anti-sweat Heat Controls	✓	✓					
Defrost	Hot Gas Defrost		✓			✓		
	Defrost Control					✓		
Lighting	LED Lighting	✓	✓		✓			✓
	Lighting - High-Lumen Bulb, Low BF		✓		✓			✓
Controls	Case Controller & EEV	✓						
	Reduced Meltage During Harvest						✓	

	Floating Head-Pressure Control		✓					
	Ambient Subcooling		✓					
	Smart Proximity Sensor				✓			✓
	Distributed Refrigeration	✓						

4.1.1 Insulation

Either increasing the insulation thickness or reducing insulation thermal conductivity will reduce the energy consumption of commercial refrigeration equipment. Typical insulation thicknesses range from 1 to 2 inches. Blow-in polyurethane foam is used for most equipment. We assume an increase in total wall resistance, including inside and outside air layers.

Much work has been done over the years to develop lower-conductivity foam insulation. Reducing the conductivity would reduce the cabinet heat leak. A number of approaches have been tried to reduce conductivity, including adjustment of the foam blowing process or chemistry to allow formation of smaller air cells, use of opacity additives, use of foam blowing gases with reduced conductivity, reduction of foam cell wall thickness, etc. The state of the art in attainable foam insulation conductivity levels has remained relatively unchanged in the past 10 years. However, new developments in this area may still be possible.

Equations for the thicker insulation (TI) option:

$$Energy\ Savings_{TI}\left(\frac{kWh}{yr}\right)=\frac{Thermal\ Load\ (W)\times Reduction\ (\%)}{COP}\times Duty\ Cycle_{System}(\%)\times\frac{8760\ hours/yr}{1000\ W/kW}$$

$$\begin{aligned}&Installed\ Cost\ Premium_{TI}\ (\$)\\&=\left(Material\ Cost\left(\frac{\$}{ft^2}\right)\times Wall\ Area\ (ft^2\ )+Unit\ Retooling\ Cost\ (\$)\right)\\&\times Manufacturer\ \ Markup\times Dealer\ \ Markup_{SC}\end{aligned}$$

Where for food service equipment, reach-ins and beverage merchandisers,

$$\begin{aligned}&Unit\ Retooling\ Cost_{TI}\ (\$)\\&=\frac{Total\ Retooling\ Cost\ (\$)}{Annual\ \ Sales\ (units)\times\%\ Retooled\times Product\ Line\ Lifetime\ (yrs)}\end{aligned}$$

And for walk-ins, there are not retooling costs associated with thicker insulation.

The energy savings and cost premium for the thicker insulation option for display cases and refrigerated vending machines are estimated using the engineering analysis spreadsheets from the DOE standards rulemakings for commercial refrigeration equipment (DOE 2009a) and beverage vending machines (DOE 2008b). We used the "VCT.RC.M" classification for display cases, the "Large Class A" classification for fully-cooled vending machines and "Large Class B" for zone-cooled vending machines. For these equipment types, we present the overall installed cost premium, which includes the added material cost and retooling costs for thicker insulation as calculated in the respective rulemakings.

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

4.1.2 Fans

High Efficiency Fan Motors

Fan motors move air across the evaporator or condenser coil and typically run at one speed. The manufacturer will match the motor size and blade to the coil to meet the expected load under most operating conditions. Higher efficiency fan motors reduce energy consumption by requiring less electrical power to generate motor shaft output power. There are a variety of types and sizes of motors, depending on the application. Key defining motor characteristics include shaft power output, power supply (AC or DC, voltage, frequency, single or multiple phases), type of motor cooling, environmental conditions, design life, etc. Motors used in commercial refrigeration applications in the U.S. typically use AC power, with power input either single phase 60 Hz at 115 volts or 230 volts, or, for some larger condenser fan motors, three phase 60 Hz 208 volts or 460 volts. They have shaft power output ranging from 6W to multiple horsepower for large condenser fan motors.

Electric motors operate based on the interaction between the magnetic fields of the rotor and the stator. Induction motors are very common, and these motors have no magnets, generating magnetic fields in the rotor by inducing current flow in the rotor windings. Some motors have permanent magnets, leading, in some cases, to more efficient designs. Single-phase induction motors require separate starting windings to assure proper start rotation and sufficient starting torque. The type of start-up differentiates the three main types of single phase induction motors, which include the shaded pole motor, the permanent split capacitor motor (PSC), and the electronically commutated permanent magnet motor (ECM). In a shaded-pole motor, the starting windings are "shaded" by a copper loop. The interactions between the magnetic field generated by the shaded portion and that generated by the un-shaded portion induce rotation when the motor is powered. The imbalance between the shaded and un-shaded portions of the magnet remains throughout operation. As a result, shaded-pole motors used in commercial refrigeration applications with shaft power output ranging from 6W to 37W are inefficient, with typical motor efficiencies less than 20 percent (Heinecke 2006). Shaded-pole motors are, however, electrically simple and inexpensive.

In a PSC motor, a smaller, start-up winding is present in addition to the main winding. The start-up winding is electrically connected in parallel with the main winding and in series with a capacitor. At start-up, the interactions between the magnetic field generated by the start up winding and that generated by the main winding induce rotation. Because of the capacitor, however, the current to the start-up winding is cut off as the motor reaches steady state. Because of this, PSC motors are more energy efficient than their shaded-pole counterparts, with efficiencies for motors with shaft power ranging from 6W to 37W in the range 50 to 70 percent. Like shaded-pole motors, PSC motors are produced in large quantities and are relatively inexpensive (DOE 1997). The brushless motor offers a 50 to 60 percent reduction in wattage.

A third type of electric motor, the electronically commutated permanent magnet (ECM) motor (also known as a brushless permanent magnet motor), is more energy-efficient than either shaded-pole or PSC motors. ECM motors are more complex than either shaded pole or PSC motors, particularly for commercial refrigeration applications, because they are internally powered with DC power. A power supply is required to convert from AC line power to DC, and control electronics are required to handle the electronic commutation, i.e. switching the power to the motor windings in synchronization with motor rotation. For this reason, ECM motors can weigh more than shaded pole or PSC motors, and they are more expensive.

Evaporator fans save additional energy due to the reduced refrigeration load and less compressor energy required to remove it from the cabinet.

Energy savings equations for the PSC and ECM fan options:

$$Energy\ Savings_{PSC,Cond}\left(\frac{kWh}{yr}\right) = (Power_{Base}(W)\text{-}\ Power_{PSC}(W)) \times Number\ of\ Fans \times Duty\ Cycle_{Cond\ Fan}(\%) \times \frac{8760\ hours/yr}{1000\ W/kW}$$

$$Energy\ Savings_{ECM,Cond}\left(\frac{kWh}{yr}\right) = (Power_{Base}(W)\text{-}\ Power_{ECM}(W)) \times Number\ of\ Fans \times Duty\ Cycle_{Cond\ Fan}(\%) \times \frac{8760\ hours/yr}{1000\ W/kW}$$

$$Energy\ Savings_{ECM,Cond}\left(\frac{kWh}{yr}\right) = (Power_{Base}(W)\text{-}\ Power_{PSC}(W)) \times Number\ of\ Fans \times Duty\ Cycle_{Cond\ Fan}(\%) \times \frac{8760\ hours/yr}{1000\ W/kW}$$

$$Energy\ Savings_{ECM,\ Evap}\left(\frac{kWh}{yr}\right) = (Power_{Base}(W)\text{-}\ Power_{ECM}(W)) \times Number\ of\ Fans \times Duty\ Cycle_{Evap\,Fan}(\%) \times \left(1+\frac{1}{COP}\right) \times \frac{8760\ hours/yr}{1000\ W/kW}$$

Installed cost premium equation for PSC and ECM fan options:

$$Installed\ Cost\ Premium_{PSC}\ (\$) = (OEM\ Cost_{PSC}\ (\$) - OEM\ Cost_{Baseline}\ (\$)) \times Number\ of\ Fans \times Manufacturer\ Markup \times Dealer\ Markup_{SC}$$

$$Installed\ Cost\ Premium_{ECM}\ (\$) = (OEM\ Cost_{ECM}\ (\$) - OEM\ Cost_{Baseline}\ (\$)) \times Number\ of\ Fans \times Manufacturer\ Markup \times Dealer\ Markup_{SC}$$

Table 4-3 shows typical power input and OEM costs for shaded pole, PSC, and ECM motors for a range of shaft power levels relevant for commercial refrigeration.

Table 4-3: Fan Motor Typical Efficiencies and Costs ($2008)

Rated Shaft Output (W)	SPM		PSC		ECM	
	Power Input (W)	OEM Cost	Power Input (W)	OEM Cost	Power Input (W)	OEM Cost
373 (1/2 hp)		-	530	$75.23	450	$94.04
249 (1/3 hp)			370	$57.00	304	$71.00
125 (1/6 hp)	329	$50.15	202	$63.94	155	$80.24
50 (1/15 hp)			90	$53.91	65	$67.71
37 (1/20 hp)	110	$37.61	70	$50.15	49	$65.20
25	100	$31.35	51	$46.39	33	$60.18
20	90	$25.08	42	$43.88	27	$56.42
15	75	$18.81	33	$41.38	20.5	$52.66
9	53	$12.95	21	$17.35	12.5	$27.96
6	40	$12.17	15	$16.57	8.5	$27.19
Source: Communication with the Motor and Motion Association (SMMA)						

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

Variable-Speed Control of Fan Motors

Variable-speed operation can reduce evaporator fan energy consumption and the associated fan-heat load when cooling capacity requirements permit lower evaporator air flows. However, under full-capacity operation, there is no energy savings benefit with condenser fan motor controllers.

Equations for the evaporator fan control (EFC) option:

$$Energy\ Savings_{EFC}\left(\frac{kWh}{yr}\right) = Power_{EvapFan}(W) \times Number\ of\ Fans \times Duty\ Cycle\ Reduction_{EFC}(\%) \times \left(1+\frac{1}{COP}\right) \times \frac{8760\ hours/yr}{1000\ W/kW}$$

$$Installed\ Cost\ Premium_{EFC}(\$) = OEM\ Cost\ of\ Contractor(\$) \times Number\ of\ Fans \times Manufacturer\ Markup \times Dealer\ Markup_{SC}$$

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

High-Efficiency Fan Blades

High efficiency fan blades reduce motor shaft power requirements by moving air more efficiently. Most evaporator and condenser fans use stamped sheet metal or plastic axial fan blades. These fan blades are lightweight and inexpensive. The blades are typically supplied by a fan blade manufacturer and mounted to the motor by the equipment manufacturer. These fan systems are mass produced for a broad range of applications, and are not necessarily optimized for specific equipment types. For example, evaporator fans may operate at compromised efficiencies, because the standard-design sheet metal fans are not well suited for the relatively high pressure drops required. .

In some cases tangential fans, also known as cross-flow blowers, have been used to decrease refrigeration equipment energy consumption. Tangential fans have long thin impellers, allowing them to be packaged into some applications better than propeller fans. This can provide benefits for evaporators and condensers by improving the distribution of airflow and reducing transition losses. A single long tangential fan can meet the airflow requirements for an entire condenser, while only requiring one high efficiency fan motor. However, selection of such fans must be done with careful consideration of the application and verification of results, because they are not always the most efficient option, especially for high pressure rise situations.

Fan blades optimized for the ice-machine application can lower condenser fan power by about 15%. This represents about 61 kWh for the baseline ice machine. A $2 OEM cost premium per blade is estimated for the economic analysis (assuming $32,000 tooling costs distributed over 32,000 fan blades.

Equations for the high efficiency fan blade (HEFB) option:

$$Energy\ Savings_{HEFB}\left(\frac{kWh}{yr}\right) = \begin{bmatrix} Power_{EvapFan}(W) \times Number\ of\ Evap\ Fans \times Duty\ Cycle_{EvapFan}(\%) \times \left(1+\frac{1}{COP}\right) \\ + Power_{Cond\ Fan}(W) \times Number\ of\ Cond\ Fans \times Duty\ Cycle_{Cond\ Fan}(\%) \end{bmatrix} \times Power\ Reduction\ (\%) \times \frac{8760\ hours/yr}{1000\ W/kW}$$

$$Installed\ Cost\ Premium_{HEFB}(\$) = \begin{pmatrix} Cost\ of\ Condenser\ Blade_{Baseline}(\$) \times Number\ of\ Cond\ Fans \\ + Cost\ of\ Evaporator\ Blade_{Baseline}(\$) \times Number\ of\ Evap\ Fans \end{pmatrix} \times Cost\ Increase\ (\%) \times Manufacturer\ Markup \times Dealer\ Markup$$

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

4.1.3 Compressors

High-Efficiency Compressors

Several technologies exist to increase the efficiency of compressors. High efficiency reciprocating and scroll compressors, which often incorporate variable-speed motors are found in some supermarket refrigeration systems. These compressors have higher efficiencies over a wide operating range than the traditional reciprocating compressors commonly used in small, self-contained commercial refrigeration equipment.

However, for equipment types such as self contained refrigerators and vending machines where the rating test procedure and efficiency standards are based on single operating points, the efficiency advantages of variable speed compressors are not apparent. Variable speed compressors may have similar or even lower efficiency than single speed units at the rating point condition. Consequently, current efficiency standards for these equipment types does not reward the application of variable speed compressors, which do involve substantial additional costs.

Embraco supplies small, high-efficiency compressors to manufacturers of residential refrigerator-freezers (i.e. Whirlpool, GE, Frigidaire, etc.). These compressors are low-suction pressure units ranging from 600 Btu/h to 950 Btu/h in capacity. HFC-134a is the refrigerant used. Reported compressor efficiencies for the Embraco EGX are up to 6.26 EER at -10°F evaporator temperature and 130°F condensing temperature. These compressors are used as a basis for the achievable efficiencies of hermetic reciprocating compressors. The theoretical maximum (isentropic) EER for the -10/130 rating condition is 10.2. The overall efficiency of the 6.26 EER Embraco compressor is, therefore, 61%. The prototypical beverage merchandiser compressor could be modified to achieve similar efficiency by use of a higher-efficiency motor (80%), reducing suction gas pressure losses, reducing the valve clearance gap, reducing the heating of suction gas within the compressor shell, reducing pressure drop through the discharge valve, and reducing mechanical losses. Improvement in the compressor to achieve a 60% overall efficiency would result in a 20% reduction in the electric load.

For ice machines, energy savings are conservatively based on savings during the freeze cycle only, since the savings associated with ice harvest is not well understood.

Scroll compressors compress gas in a fundamentally different manner from reciprocating compressors—between two spirals, one fixed and one orbiting. High efficiency reciprocating compressors are as efficient, or more efficient, than scroll compressors. However, some drawbacks exist including noise, cost and reliability, compared to scroll compressors.

Equations for the high efficiency compressor (HEC) option:

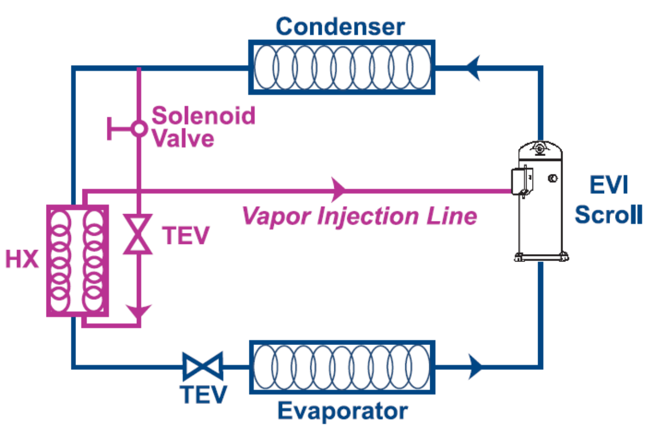
$$Energy\ Savings_{HEC}\left(\frac{kWh}{yr}\right)=\left[Capacity_{Compressor}\,(W)\times\left(\frac{1}{COP_{Baseline}}-\frac{1}{COP_{HiEff}}\right)\times Duty\ Cycle_{Compressor}\,(\%)\right]\times\frac{8760\ hours/yr}{1000\ W/kW}$$

$$Installed\ Cost\ Premium_{HEC}\,(\$)=OEM\ Compressor\ Cost\,(\$)\times Manufacturer\ Markup\times Dealer\ Markup_{SC}$$

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

Vapor Injection (Supermarket Refrigeration only)

Some designs of scroll compressors incorporate an intermediate-pressure vapor inlet port, which allows the compressor to be used in dual-stage configurations, for example the Copeland ZF series (see **Figure 4-1** for refrigeration circuit). The intermediate port can accept refrigerant vapor from a liquid sub-cooling heat exchanger or from an intermediate pressure flash tank. This approach improves efficiency by allowing part of the refrigerant load to be served by compression of refrigerant from the higher intermediate pressure level rather than from the lower suction pressure level. It allows implementation of mechanical sub-cooling without use of a separate compressor.



Source: Emerson Climate Technologies 2007

Figure 4-1: Vapor Injection Circuit for Low Temperature Applications

ECM Compressor Motor

Improvement in compressor efficiency could also be achieved with the use of an ECM compressor motor. Data for these motors in 1/3 and 1/2 hp sizes are presented in **Table 4-3**. Currently such compressors are only available in limited numbers for special order. Equations for the ECM compressor motor (ECM) option:

$$Energy\ Savings_{ECM\ Comp}\left(\frac{kWh}{yr}\right) = Power_{Compressor}\ (W) \times Power\ Reduction\ (\%) \times Duty\ Cycle_{Compressor}\ (\%) \times \frac{8760\ hours/yr}{1000\ W/kW}$$

$$Installed\ Cost\ Premium_{ECM\ Comp}\ (\$) = OEM\ ECM\ Motor\ Cost\ (\$) \times Manufacturer\ Markup \times Dealer\ Markup$$

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

Variable Capacity Compressor (Compressor Modulation)

Two types of variable capacity compressors that can also reduce energy consumption have emerged in recent years: (1) variable-speed compressors which are implemented through the use of an electronic control on the compressor motor, which allows the motor to operate at different speeds, and (2) digital scroll compressors, which are implemented through electronic control of the time duration for the engagement of the scroll elements where 100% capacity is achieved (state 1) when the scroll elements are engaged and 0% capacity (state 0) when they are disengaged. Continuous capacity modulation from 10% to 100% is achieved by rapid control of the duration of time in each state. For example, 50% capacity is accomplished with 10-sec on and 10-sec off. This modulation approach has also been developed for semi-hermetic reciprocating compressors used for supermarket applications. An older form of compressor capacity modulation that does not provide as great an efficiency benefit is cylinder unloading for reciprocating compressors.

Variable-speed compressors are implemented either through use of an inverter operating with an induction motor, or with a permanent magnet motor operating with a motor controller that allows frequency modulation.

Variable-capacity compressors reduce energy consumption in three ways:

1. When refrigerant flow is reduced during part-load operation, the condenser and evaporator (sized for full-load conditions) operate more efficiently and, hence, lower the overall refrigerant circuit temperature lift.
2. In a self-contained refrigeration system using a capillary tube refrigerant metering device, the pressure in the system equilibrates during a compressor off-cycle. During this equalization, warm refrigerant is passed from the condenser to the evaporator, where it adds to the thermal load.
3. The compressor startup adds energy use during a time when the system is not providing much or any useful capacity.

Equations for the variable capacity compressor (VCC) option:

$$Energy\ Savings_{VCC}\left(\frac{kWh}{yr}\right) = Energy\ Savings_{ECM\ Comp}\left(\frac{kWh}{yr}\right) \times (1 + Savings\ Improvement\ \%)$$

$$Installed\ Cost\ Premium_{VCC}\ (\$) = Installed\ Cost\ Premium_{ECM\ Comp}\ (\$) + VC\ Controls\ Cost\ (\$)$$

The energy savings of compressor modulation (CM) in supermarkets is estimated to be 7 to 10 percent of total compressor rack energy use (Emerson 2009). We estimate the cost to be approximately 10 percent of the baseline compressor purchase price of $100,000, along with the manufacturer and dealer markups.

$$Energy\ Savings_{CM}\left(\frac{kWh}{yr}\right) = 8.5\% \times Compressor\ Rack\ Energy\ Use_{Baseline}\left(\frac{kWh}{yr}\right)$$

$$Installed\ Cost\ Premium_{CM}\,(\$) = 10\% \times Compressor\ Purchase\ Price_{Baseline}\,(\$) \times Manufacturer\ Markup \times Dealer\ Markup$$

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

4.1.4 Heat Exchangers

Evaporator and Condenser Design

Evaporator or condenser performance can be enhanced by increasing surface area and/or turbulence on either the refrigerant side (inside the tubes) or on the air side. Refrigerant-side heat transfer can be enhanced by using:

- Internally finned (Rifled or diamond-pattern) tubing
- Turbulence-promoting inserts in tubes
- Alternative cross-sectional tube shapes, such as flattened tubes (used in micro-channel heat exchangers), which provide the added benefit of lowered air-side pressure drop
- More, smaller-diameter, parallel tubes per circuit
- Diverging circuits (in evaporators) or converging circuits (in condensers) to maintain good refrigerant velocity, without excess pressure drop, as the refrigerant evaporates or condenses.

Enhancements to the air-side heat transfer include:

- Increasing fin density (decreased fin spacing), subject to the constraints of increased air-side pressure drop, frost bridging (for evaporators), and dust/dirt buildup (for condensers).
- Fin patterns (wavy, raised lance, louvered),

Increasing the overall size of the coil in one or more dimensions without changing other aspects of the coil is another way to increase the surface area and, hence, promote heat transfer. However, space (and cost) constraints generally limit the coil size. Approaches to increasing the coil heat transfer must also be balanced against the power requirements for air moving (especially in evaporators, where the fan power adds to the cooling loads).

Micro-channel heat exchangers offer a way to decrease the overall size of the coil while still achieving the same performance. Micro-channel coils use channel widths of 10 to 1,000 µm. By constraining the flow to such narrow channels, thermal diffusion lengths are short, and the characteristic heat transfer coefficients are very high. With such high thermal performance, relatively short flow passages are required, and with many flow passages in parallel in a small device, the pressure drop can also be small (PNL 2008). Studies have shown that both the COP and cooling capacity of systems using a micro-channel condenser were higher than those for the round-tube condenser (International Journal of Refrigeration 2008b). Micro-channel technology is currently used widely for condensers in automotive air-conditioning and is starting to be adopted in other HVAC applications, where small size and low weight is important and/or where need for a very large amount of heat exchanger surface area makes the technology cost effective. In this report, micro-channel technology is considered an existing technology but omitted from the economic analysis due to its low market penetration in the refrigeration market. The technology is not appropriate for refrigeration evaporators and is typically not cost effective for sizes needed for the condensers of most of the equipment discussed in the report. However, for large supermarket condensers, micro-channel technology might become cost effective.

The energy savings and cost premium for the enhanced evaporator option in display cases is estimated for the vertical, remote-condensing display case with transparent doors using the engineering analysis spreadsheet from the DOE standards rulemaking for display cases (DOE 2009a).

The energy savings and cost premium for the enhanced condenser option for refrigerated vending machines is estimated using the engineering analysis spreadsheet from the DOE standards rulemaking for beverage vending machines (DOE 2008b). We used the “Large Class A” classification for fully-cooled vending machines and “Large Class B” for zone-cooled vending machines.

Supermarket condensers can achieve energy savings through the use of both higher efficiency fan motors and variable frequency drive (VFD) of the fan motors. High efficiency fan motors use an electronically commutated motor (ECM) and can achieve up to 90 percent motor efficiency, therefore requiring less power to produce the same amount of air flow. VFD can be implemented as a stand-alone option, but it is increasingly incorporated into the overall refrigeration control system. The EIA gives an estimate for energy savings using a set of high efficiency remote air-cooled condensers to be 27 percent, with an associated cost premium of $5,000.

Reduced Thermal Cycling of the Evaporator (Ice Machines only)

In the baseline ice machine, the thermal cycling of the evaporator accounts for about 9% of the compressor input energy during the freeze cycle. The prototypical evaporator design is a copper serpentine attached to the rear of plated copper waffle ice-making surfaces. Copper has a high thermal conductivity, but also a high thermal mass. Assuming the thermal mass could be reduced by a factor of two with no change in thermal conductivity, a savings of about 4-5%, or about 230-290 kWh/yr could be realized. Realistically, a reduction of the thermal mass for this evaporator design would probably result in lower thermal conductivity, which would offset some of the projected savings for the ideal case.

Equations for the reduced thermal cycling (RTC) option:

$$Energy\ Savings_{RTC}\left(\frac{kWh}{yr}\right) = Compressor\ Energy\ Use_{Baseline}\left(\frac{kWh}{yr}\right) \times Energy\ Use\ Reduction\ (\%)$$

See **Table 4-4** for definitions of the equation variables. The installed cost premium is adapted directly from ADL 1996 to approximately $25 using the inflation factor described in 0.

External Heat Rejection (Walk-in Freezers only)

Some smaller walk-in coolers and freezers with packaged refrigeration systems are installed in interior spaces and reject heat to the interior space, for the convenience of installation. Efficiency of these systems could be improved by external rejection of heat. This would require external placement of the condenser or of the entire condensing unit. The use of internal heat rejection will also impact energy requirements for space conditioning either (1) due to the increased make-up air requirement associated with exhausting the heat, or (2) by directly impacting the air-conditioning load. For the representative city, Washington D.C., the walk-in freezer compressor and condenser duty cycle is reduced from 70% to 61% (ADL 1996). The cost premium represents the additional installation labor required, which we assume to be 8 hours of work by two workers at a wage of $63/hr (ADL 1996, adjusted for inflation).

Equations for external heat rejection (EHR) option:

$$Energy\ Savings_{EHR}\left(\frac{kWh}{yr}\right) = Reduced\ \ Duty\ Cycle_{EHR}(\%) \times (Power_{Compressor}\ (W) + Power_{Condenser\ Fan}(W)) \times \frac{8760\ hours/yr}{1000\ W/kW}$$

$$Installed\ Cost\ Premium_{ASC}(\$) = \#\ of\ \ workers\left(\frac{\$}{hr}\right) \times Installation\ Time\ (hrs)$$

See **Table 4-4** and **Table 4-5** for definitions of the equation variables.

4.1.5 Anti-Sweat Heaters

Hot Gas Anti-Sweat Heating

It is possible that electric anti-sweat heaters (ASH) could be replaced by a hot refrigerant gas line running around the doorframe. This technology is used extensively in residential freezers. Implementation in self-contained commercial refrigerators and freezers has begun due to increased interest in high efficiency technologies, despite concerns of durability compared to the electric heating option. Implementation in supermarket display cases is complicated because it would require additional piping runs to the cases. Some display cases use hot gas defrost, and these cases would have a source for the hot gas needed to supply such an ASH loop. However, the refrigerant leaving this loop may not be completely condensed, and hence feeding the exit refrigerant into the evaporator may be counterproductive. For both supermarket and self-contained products, another consideration is the possibility of the operators penetrating the anti-sweat loop with fasteners, something that is much more likely to occur in commercial than residential settings.

Equations for hot gas anti-sweat (HGAS) option:

$$Energy\ Savings_{HGAS}\left(\frac{kWh}{yr}\right) = Power_{ASH}(W) \times Duty\ Cycle_{ASH}(\%)$$

$$Installed\ Cost\ Premium_{HGAS}(\$) = Unit\ Retooling\ Cost_{HGAS}(\$) + Pipe\ Cost\left(\frac{\$}{ft}\right) \times Perimeter\ of\ Doors\,(ft)$$

Where,

$$Unit\ Retooling\ Cost_{HGAS}(\$) = \frac{Total\ Retooling\ Cost_{HGAS}(\$)}{Annual\ Sales\,(units) \times \%\ Retooled \times Product\ Line\ Lifetime(yrs)}$$

See **Table 4-4** and **Table 4-5** for definitions of the equation variables.

Anti-Sweat Heater Control

Anti-sweat heaters (ASH) are necessary to prevent condensation on door-gasket surfaces and glass (for display cases and beverage merchandisers), the temperatures of which can be below the ambient air dew point. ASHs often operate continuously. ASH controllers can lower energy consumption during low-humidity conditions by operating heaters only as needed to maintain surface temperatures above the dew-point temperature. A heater can be turned on when the temperature of the heated surface falls below the dew point, or the heaters can be cycled with on-times increasing with increasing dew-point temperature. We assume that anti-sweat-heater controls lower the total anti-sweat heating load by 1/3. Reducing ASH on-time also yields refrigeration energy savings, since ASHs contribute to case heat load. We assume that half of the anti-sweat heating load contributes to the refrigeration load. Installed cost of sensor and controller is approximately $627.

Equations for anti-sweat heater control (ASHC) option:

$$Energy\ Savings_{ASHC}\left(\frac{kWh}{yr}\right) = Power_{ASH}(W) \times Power\ Reduction_{ASHC}(\%) \times \left(1 + \frac{1}{2 \times COP}\right)$$

$$Installed\ Cost\ Premium_{ASHC}(\$) = Sensor\ \&\ Controller\ Cost(\$)$$

See **Table 4-4** and **Table 4-5** for definitions of the equation variables.

4.1.6 Doors

Display-Case Door Technologies

Doors to refrigerated display cases can be improved by using both anti-sweat heater control and better materials. Instead of an aluminum frame, a less-conductive vinyl-composite frame can be used. Also, the doors can be constructed using glass that lowers both thermal radiation and heat conduction.

The energy savings and cost premium for the improved display case door option are estimated for the vertical, remote-condensing display case with transparent doors using the engineering analysis spreadsheet from the DOE standards rulemaking for display cases (DOE 2009a).

Automatic Door Closer (Walk-ins only)
The automatic door closer is a device that automatically closes the door when it is left open. This reduces ambient air infiltration and decreases the refrigeration load. Energy savings and cost data is taken directly from the analysis in PG&E 2004.

Strip Curtains (Walk-ins only)
Strip curtains are clear flexible strips located at the opening of a walk-in cooler or freezer. They reduce ambient air infiltration when the door is open. Energy savings and cost data is taken directly from the analysis in PG&E 2004.

High Efficiency Low/No Heat Reach in Doors (Walk-ins only)
Walk-ins with reach-in doors can have gas-filled, multi-pane glass that reduces or eliminates use of anti-sweat heaters. The insulated glass panels also reduce the heat conducted into the refrigerated space and thus decrease the refrigeration load. Energy savings and cost data is taken directly from the analysis in PG&E 2004.

4.1.7 Defrost

Defrost Mechanisms
There are three methods available for defrosting the evaporator coil in this case: off-cycle defrost, electric defrost, and hot gas defrost. Off-cycle defrost involves shutting off refrigerant flow to the coil while leaving the evaporator fan running. This method is used in refrigerators where cabinet air is above the freezing point of water and can be used to melt the frost. Electric defrost is used in freezers where the air temperature is not high enough to defrost the coil, and where defrost must occur quickly to prevent any significant rise in product temperature. Electric defrost involves melting frost by briefly turning on an electric resistance heater, which is in contact with or near the evaporator coil. Hot-gas defrost involves piping and valves that direct hot gas from the compressor discharge into the evaporator that would otherwise be waste heat, therefore using energy more efficiently. Control of the defrost cycle can also lead to increased energy savings.

Costs would increase due to additional controls and refrigerant piping.

Equations for the hot gas defrost (HGD) option:

$$Energy\ Savings_{HGD}\left(\frac{kWh}{yr}\right) = Power_{Defrost}(W) \times Duty\ Cycle_{Defrost}(\%) \times \frac{8760\,{}^{hrs}\!/\!{}_{yr}}{1000\,{}^{W}\!/\!{}_{kW}}$$

Installed cost premium is based on research from ADL 1996 related to HGD in supermarkets, where the total cost was $3,800 for 46 circuits in 1996. We assumed twice the cost for the same number of circuits equaling $165 per circuit. Here, we use the price for one circuit as the installed cost premium.

$$Installed\ Cost\ Premium_{HGD}(\$) = Cost\ per\ circuit_{HGD}(\$) + 1\,circuit$$

See **Table 4-4** and **Table 4-5** for definitions of the equation variables.

Defrost Control (Reach-in Freezers only)

The most promising defrost technique involves monitoring the temperature drop across cooling coils to determine whether air flow rates have dropped. Defrost control is estimated to eliminate half of the required defrost energy for the six cooler months of the year, with additional compressor energy savings due to reduced internal load (ADL 1996).

Equations for defrost control (DC) option:

$$Energy\ Savings_{DC}\left(\frac{kWh}{yr}\right) = Power_{Defrost}(W) \times Duty\ Cycle_{Defrost}(\%) \times Energy\ Reduction\,(\%)$$

$$\times (0.5\,of\ \ year) \times \left(1 + \frac{1}{COP}\right) \times \frac{8760\,{}^{hrs}\!/_{yr}}{1000\,{}^{W}\!/_{kW}}$$

$$Installed\ Cost\ Premium_{DC}(\$) = Cost\,of\ \ 2\ Sensors\,\&\,Controls(\$)$$

See **Table 4-4** and **Table 4-5** for definitions of the equation variables.

4.1.8 Lighting

High Efficiency Lighting

Efficiency in lighting is commonly measured as efficacy (lumens/watt), or the quantity of light output (measured in lumens) divided by electrical power input (measured in Watts).

For display cases, beverage merchandisers, and vending machines, since sales levels strongly correlate with lighting levels, energy savings will most likely be accomplished through use of high-efficiency lamps and ballasts, rather than reduction in light output. The refrigeration industry has transitioned into the use of T-8 and fluorescent lighting in commercial refrigeration equipment with lighting, which is substantially more efficacious than T-12 lighting.

T-8 lighting is predominantly used with electronic ballasts, which are more efficient than magnetic ballasts commonly used in T-12 lighting. Electronic ballasts use solid state electronics to modulate power provided to fluorescent lamps. Electronic ballasts, which convert power at high frequency, have lower electrical resistance losses compared to magnetic ballasts which operate at line frequency. Fluorescent lamps also operate more efficiently at the higher frequency provided by electronic ballasts. In addition to the direct reductions in electrical power consumption, heat generated by the lighting and the lighting ballast contributes to the cabinet or case heat load, since these components are typically installed in the refrigerated space. Therefore, increasing ballast and/or lamp efficiency reduces refrigeration loads. We assume that approximately two thirds of the heat from lighting enters the refrigerated space.

An even more recent trend is the use of light emitting diode (LED) technology. Although LEDs are currently less efficacious than fluorescent technology, they are more directional than linear fluorescent bulbs, allowing for comparable illumination with less total wattage. In addition, they are more amenable to rapid on/off control than fluorescents, so they are more suitable to be used with proximity sensors, and they may be more amenable to low-temperatures operations. There have been recent advancements in LED efficacy as well as the adoption of LED technology. For example, Seaga Manufacturing uses LEDs in their new Premium Collection of refrigerated beverage vending machines. Research by the Lighting Resource Center indicates that lighted display cases using LEDs are attractive to consumers. LEDs are predicted to steadily increase in efficacy and decrease in cost as the technology improves (RPI 2002).

An estimated potential energy savings of 6% could be achieved through the use high efficient bulbs and low ballast factor. An estimated potential energy savings of 74% could be achieved through the use of LED lighting.

Equations for high efficiency fluorescent lighting (HEF) option:

$$Energy\ Savings_{HEF}\left(\frac{kWh}{yr}\right)=\begin{bmatrix}(Bulb\ Power_{Baseline}(W)-Bulb\ Power_{HEF}(W))\times\#\ of\ \ Bulbs+\\(Ballast\ Power_{Baseline}(W)-Ballast\ Power_{HEF}(W))\times\#\ of\ \ Ballasts\end{bmatrix}$$

$$\times\left(1+\frac{2/3}{COP}\right)\times Duty\ Cycle_{System}(\%)\times Duty\ Cycle_{Lighting}(\%)\times\frac{8760\,{}^{hrs}\!/\!_{yr}}{1000\,{}^{W}\!/\!_{kW}}$$

$$Installed\ Cost\ Premium_{HEF}(\$)=\begin{bmatrix}OEM\ \ Bulb\ Cost_{HEF}\left(\frac{\$}{bulb}\right)+OEM\ \ Ballast\ Cost_{HEF}\left(\frac{\$}{bulb}\right)\\ -\ OEM\ \ Bulb\ Cost_{HEF}\left(\frac{\$}{bulb}\right)-OEM\ \ Ballast\ Cost_{HEF}\left(\frac{\$}{bulb}\right)\end{bmatrix}$$

$$\times\#\ of\ \ Bulbs\times Manufacturer\ \ Markup\times Dealer\ \ Markup$$

Equations for LED lighting (LED) option:

$$Energy\ Savings_{LED}\left(\frac{kWh}{yr}\right)=\begin{bmatrix}(Bulb\ Power_{Baseline}(W)\times\#\ of\ \ Bulbs_{Baseline}-Bulb\ Power_{LED}(W)\times\#\ of\ \ Bulbs_{LED}\ )+\\Ballast\ Power_{Baseline}(W)\times\#\ of\ \ Ballasts\end{bmatrix}$$

$$\times\left(1+\frac{1}{COP}\right)\times Duty\ Cycle_{System}(\%)\times Duty\ Cycle_{Lighting}(\%)\times\frac{8760\,{}^{hrs}\!/\!_{yr}}{1000\,{}^{W}\!/\!_{kW}}$$

$$Installed\ Cost\ Premium_{LED}(\$) = \left[\begin{array}{l} OEM\ Bulb\ Cost_{LED}\left(\frac{\$}{bulb}\right) \times \#\ of\ Bulbs_{LED} - OEM\ Bulb\ Cost_{Baseline}\left(\frac{\$}{bulb}\right) \\ \times \#\ of\ Bulbs_{Baseline} - OEM\ Ballast\ Cost_{Baseline}\left(\frac{\$}{bulb}\right) \end{array} \right] \times Manufacturer\ Markup \times Dealer\ Markup$$

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

The energy savings and cost premium for both lighting options in display cases is estimated for the vertical, remote-condensing display case with transparent doors using the engineering analysis spreadsheet from the DOE standards rulemaking for display cases (DOE 2009a).

The same information for refrigerated vending machines is estimated using the engineering analysis spreadsheet from the DOE standards rulemaking for beverage vending machines (DOE 2008b). We used the “Large Class A” classification for fully-cooled vending machines and “Large Class B” for zone-cooled vending machines.

Beverage merchandisers are assumed to use the same lighting as fully-cooled refrigerated vending machines.

4.1.9 Controls

Supermarket Controls

Control systems for supermarket refrigeration systems have evolved significantly over the years. Most systems now use electronic controls using temperature and pressure sensors to monitor operating conditions and sequence rack compressor operation. Much more sophisticated control algorithms can be implemented with these controls than could be achieved with older mechanical controls, which are now no longer common. The increased communications capability of such systems also allows feedback of display case conditions to the compressor controllers, which allows the rack suction pressure set points to be floated upwards if all case temperature set points are being met.

High Efficiency Expansion Valves (Supermarkets Only)

Expansion valves are refrigerant metering devices whose purpose is to control the amount of refrigerant flowing to the evaporator coil. In doing so, they simultaneously decrease the temperature and pressure of the refrigerant, creating a cold liquid-vapor mixture. The low temperature refrigerant leaving the expansion valve can absorb the heat of the refrigerated space in the evaporator.

The most basic type of expansion device is a capillary tube. The capillary tube is a long thin tube that imposes a pressure drop in between the condenser and the evaporator as the refrigerant flows through it. Capillary tubes must be sized to the particular application and cannot adjust for variations in load or ambient operating conditions. They are often oversized for worst-case conditions, and therefore may operate at reduced efficiency during normal operation. Capillary tubes are used in most self-contained commercial refrigeration equipment. They have the advantage of allowing easy integration with the suction line to form a suction line heat exchanger.

Two alternative expansion valve technologies are thermostatic expansion valves (i.e., TXVs) and electronic expansion valves (i.e., EEVs). TXVs are mechanical control devices which adjust refrigerant flow to control the refrigerant superheat leaving the evaporator. They work by application of two pressures balancing forces on opposite sides of the valve plunger, the evaporating pressure and the pressure of a thermostatic bulb in contact with the evaporator exit piping. By appropriate selection of the "charge" within the bulb and adjustment of a counterbalance spring, the desired level of superheat is achieved over a range of evaporating pressures and conditions. TXVs are commonly used in refrigerated display cases.

EEVs are valves with actuation motors that are controlled electronically, based on sensor input representing the evaporating temperature and the evaporator exit temperature. By controlling superheat more precisely, EEVs provide better assurance that evaporators are optimally utilized, which provides more margin for raising of the suction pressure set point. EEVs also have a much wider flow range, which allows floating head pressure control to be set for lower minimum condensing temperatures. Electronic evaporator pressure regulators allow for more precise case temperature control, which also provides more margin on suction pressure set points. These controls can help keep the system operating with minimized pressure lift, thus reducing the required compressor power input.

Reducing Meltage During Harvest (Ice Machines only)
Ice meltage during harvest is assumed to be 15% for the baseline ice machine, based on performance measurements of a machine similar to the baseline machine described in Section 3.6.1 (ADL 1996). Meltage can be reduced by reducing the time the ice is exposed to the ice-making surface that has been warmed for harvest. The baseline design uses gravity to pull the ice off the plate. Assisting gravity in pulling the ice off the plate will reduce the time the ice is in contact with the warmed ice-making surface. One manufacturer uses a mechanical assist to push the ice off the ice-making surface. Another manufacturer uses a patented design involving a series of plastic baffles to separate the ice cubes, to which the ice does not adhere. Ice is removed after an 8-second application of hot gas to the evaporator. The manufacturer claims that ice meltage is negligible (ADL 1996).

Assuming the meltage rate can be reduced by 50%, the energy consumption required for the freeze cycle can be reduced by about 5%. **Figure 3-27** shows that the cooling associated with the meltage water amount is about 10% of the total freeze cycle thermal load.

Equations for reduced meltage during harvest (RM) option:

$$Energy\ Savings_{RM}\left(\frac{kWh}{yr}\right) = Energy_{FreezeCycle}\left(\frac{kWh}{yr}\right) \times Energy\ Use\ Reduction_{RM}\ (\%)$$

$$Installed\ Cost\ Premium_{RM}\ (\$) = OEM\ Cost\ (\$) \times Manufacturer\ Markup \times Dealer\ Markup$$

See Appendix A for the markups and **Table 4-4** and **Table 4-5** for the remaining equation variables.

Floating Head Pressure (Walk-ins only)

Better expansion valve and control technologies allow the head pressure of the compressor to float and, therefore, take advantage of the lower temperature lifts that are theoretically achievable at lower ambient temperatures. This decreases the load on the compressor.

Equations for floating heat pressure (FHP) option:

$$Energy\ Savings_{FHP}\left(\frac{kWh}{yr}\right) = Reduced\ Duty\ Cycle_{FHP}\ (\%) \times (Power_{Compressor} + Power_{Condenser\ Fan})$$

Modified expansion valves and head pressure control have the same per-circuit cost premium for floating head pressure as assumed for supermarkets. Installed cost premium is based on research from ADL 1996 related to FHP in supermarkets, where the total cost was $8,000 for 46 circuits in 1996. We assumed twice the cost for the same number of circuits equaling $348 per circuit. Here, we use the price for one circuit as the installed cost premium.

$$Installed\ Cost\ Premium_{FHP}\ (\$) = Cost\ per\ circuit\ (\$) \times 1\ circuit$$

See **Table 4-4** and **Table 4-5** for definitions of the equation variables.

Ambient Sub-Cooling (Walk-ins only)

Ambient subcooling is accomplished using a heat exchanger (subcooler) to further cool further the refrigerant leaving the condenser to boost refrigeration capacity with no increase in compressor power draw. We assume that the refrigeration system purchase price is half of the list price ($7,000 in ADL 1996, $8777 inflated to 2008), and that ambient sub-cooling adds 10% to the purchase price.

Equations for ambient sub-cooling (ASC) option:

$$Energy\ Savings_{ASC}\left(\frac{kWh}{yr}\right) = Reduced\ Duty\ Cycle_{ASC}\ (\%) \times (Power_{Compressor} + Power_{Condenser\ Fan})$$

$$Installed\ Cost\ Premium_{ASC}\ (\$) = Refrigeration\ System\ Cost\ (\$) \times 10\%\ Premium$$

See **Table 4-4** and **Table 4-5** for definitions of the equation variables.

Energy Management System (EMS)

These technologies have the ability to save energy through detection of the surrounding environment and adapting the energy consumption in real time. These technologies range from simple proximity sensors that simply turn off the lighting to more complex sensors that control the cooling load as well. In this report we will consider the more complex technology as an energy-saving option in our analysis. These advanced sensors learn the behavior of the surrounding environment and adapt to it in real time. They apply learning algorithms to real time conditions and in effect learn their surroundings, thus improving with time. Because they have the capability of reducing both lighting and cooling loads they can save up to 35 percent of the annual energy consumption in beverage merchandisers and 20 percent in refrigerated vending machines (Elstat 2009). The savings include 60% due to lighting and 40% due to cooling.

Equations for the energy management system (EMS) option:

$$Energy\ Savings_{EMS}\left(\frac{kWh}{yr}\right) = Energy\ Reduction_{EMS}\ (\%) \times Unit\ Energy\ Consumption\left(\frac{kWh}{yr}\right)$$

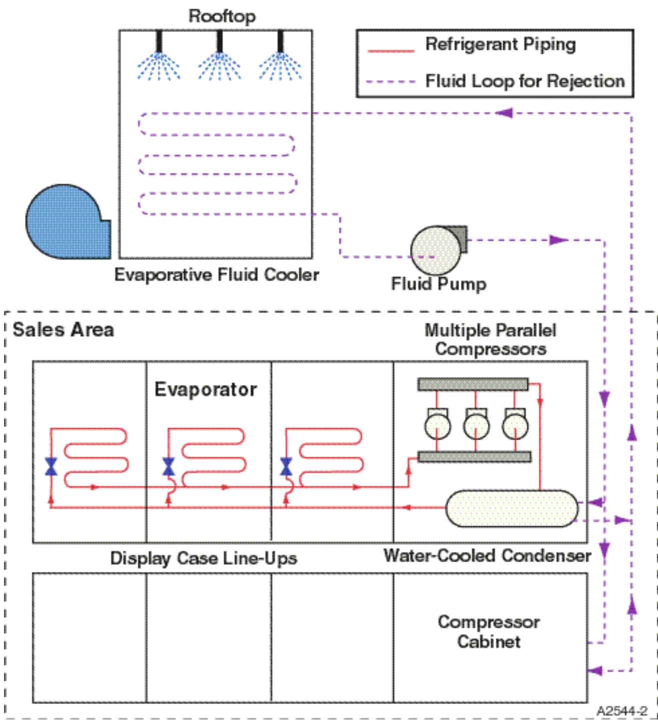
$$Installed\ Cost\ Premium_{EMS}(\$) = \$100$$

See **Table 4-4** and **Table 4-5** for definitions of the equation variables. Installed cost is estimated based on conversation with leading manufacturer of an EMS (Elstat 2009).

Distributed Refrigeration (Supermarket Refrigeration only)

Distributed refrigeration systems use multiple compressors that have been arranged in cabinets located near the loads they serve. The heat from the display cases is rejected either using air-cooled condensers located on the rooftop above the compressor cabinets or using a glycol loop that connects the cabinets to a fluid cooler (See **Figure 4-2**). The balance has tipped almost completely to air-cooled condensers in new installations, because the installation and maintenance costs are cheaper without the fluid cooler (Emerson 2009). Scroll compressors are used due to their low noise and vibration levels since they are located in the sales area. Such a configuration requires 50-70 percent less refrigerant charge than a multiplex direct expansion system. Using simulation tools, the energy savings of distributed refrigeration using evaporative-cooled fluid coolers have been estimated to be 11 percent compared to a baseline air-cooled multiplex refrigeration system, with an added cost of $60,000 per store. However, approximately 8 percent of the savings are associated with evaporative cooling (IEA 2003).

Implementation of distributed refrigeration systems in new supermarket construction is becoming more common, because supermarkets are under increasing pressure to reduce refrigerant charge, but energy use is generally not significantly reduced compared to conventional rack systems. Consequently, we do not analyze distributed systems in our energy savings analysis.



Source: IEA 2003

Figure 4-2: Water-Cooled Distributed Refrigeration System

Table 4-4: Equipment Specifications and Duty Cycle Assumptions

	Display Cases	Compressor Racks	Supermarket Condensers	Walk In Coolers	Walk In Freezers	Food Service Equip.
Annual Energy Use (kWh/yr)	8,688	1,000,000	138,000	42182	15524	3472
Compressor COP	4.04	1.71	----	3.42	1.00	2.04
Compressor Capacity (Btu/hr)	----	1,080,000	----	44,926	4,930	1,250
Compressor Power (W)	----	185,000	----	3,850	1,445	180
Wall Area (ft2)	----	----	----	648	353	67.85
Wall Thickness (in)	204	----	----	4	4	2.25
Glass Area (ft2)	1.5	----	----	----	----	----
Evaporator Fan Motor Output (W)	2610	----	----	37	20	6
Evaporator Fan Motor Power Input (W)	65	----	----	100	90	15
Number of Evaporator Fans	Shaded Pole	----	----	8	2	1
Condenser Fan Motor Output (W)	6	----	?	373	125	9
Condenser Fan Motor Power Input (W)	30	----	?	530	329	53
Number of Condenser Fans	5	----	?	2	1	1
Defrost Power (W)	----	----	----	----	2000	----
Hot Gas Solenoid (W) [1]	----	----	----	----	----	----
Anti-sweat Heater Power (W)	----	----	----	300	230	50.60
Lighting Type	----	----	----	T-8, Elec Ballast	Incand-escent	T-8, Elec Ballast
Number of Bulbs/Ballasts	----	----	----	4/1	2	----
Bulb Power (W/bulb)	----	----	----	72	40	----
Ballast Power (W/bulb)	500	----	----	?	----	----
System	T-8, Elec Ballast	100%	100%	100%	100%	100%
Compressor	6	0.0	----	66%	70%	65%
Evaporator Fan	58	----	----	100%	100%	100%
Condenser Fan	0	----	0	66%	70%	65%
Anti-sweat Heater	100%	----	----	100%	100%	100%
Defrost		----	----	----	4%	6%

Drip Pan Heater [2]	100%	----	----	----	4%	----
Lighting		----	----	66% (50%)	50%	----
[1] Ice Machines only [2] Walk-in Freezer only						

Table 4-4 continued: Equipment Specifications and Duty Cycle Assumptions

	Reach In Refrigerators	Reach In Freezers	Beverage Merchandisers	Ice Machines	Fully-Cooled Vending Machines	Zone-Cooled Vending Machines
Annual Energy Use (kWh/yr)	2477	3960	2527	5248	2494	2258
Compressor COP	2.04	1.25	1.72	----	1.77	1.67
Compressor Capacity (Btu/hr)	3,000	2,200	2,500	----	2,900	2,400
Compressor Power (W)	440	530	425	1,000	480	420
Wall Area (ft^2)	----	----	----	1400	----	----
Wall Thickness (in)	120	80	69	----	89.18	101.74
Glass Area (ft2)	2.25	2.25	1.5	----	1	1
Evaporator Fan Motor Output (W)	264	329	204	----	1074	389
Evaporator Fan Motor Power Input (W)	----	----	27.5	----	15.67	----
Number of Evaporator Fans	PSC	PSC	PSC	----	PSC	PSC
Condenser Fan Motor Output (W)	9	9	9	----	9	9
Condenser Fan Motor Power Input (W)	31.03	31.03	31.03	----	31.03	31.03
Number of Condenser Fans	2	1	2	----	1	2
Defrost Power (W)	PSC	PSC	PSC	Shaded Pole	Shaded Pole	Shaded Pole
Hot Gas Solenoid (W) [1]	20	37	9	25	9	9
Anti-sweat Heater Power (W)	90	70	31.03	100	45	45
Lighting Type	1	1	1	1	1	1
Number of Bulbs/Ballasts	----	600	----	----	----	----
Bulb Power (W/bulb)	----	----	----	15	----	----
Ballast Power (W/bulb)	49.5	42.5	----	----	----	----
System Duty Cycle	Incand-escent	Incand-escent	T-8, Elec Ballast (48")	----	T-8, Elec Ballast	T-8, Elec Ballast
Compressor Duty Cycle	2	1	2/1	----	2/1	2/1
Evaporator Fan Duty Cycle	25	25	30.1	----	30.1	41.3

Condenser Fan Duty Cycle	----	----	1.4	----	1.4	2
Anti-sweat Heater Duty Cycle	100%	100%	100%	50%	100%	100%
Defrost Duty Cycle	0.7	0.8	0.5	0.9	0.3	0.3
Lighting Duty Cycle	1	1	1	----	1	1
Compressor Duty Cycle (harvest) [1]	0.65	0.75	0.45	0.93	0.35	0.30
Water Pump Duty Cycle [1]	1	1	----	----	----	----
[1] Ice Machine only						

Table 4-5: Technology Option Parameter Definitions

Technology Option [1]	Parameter	Display Cases	Compressor Racks	Supermarket Condensers	Walk In Coolers	Walk In Freezers	Food Service Equip.
Thickness Increase	Thickness Increase (in)	0.5	----	----	1.0	1.0	1.0
	Load Reduction (%)	5.7%	----	----	20%	20%	22%
	Material Cost ($)	$40	----	----	$0.41	$0.41	$0.41
	Total Retooling Cost ($)	----	----	----	----	----	$1,000,000
Evap Fan Control	Duty Cycle Reduction (%)	----	----	----	20%	20%	----
	OEM Cost of Contactor($)	----	----	----	$63	$63	----
High Efficiency Fan Blades	Power Reduction (%)	----	----	----	15%	15%	15%
	Baseline Cost of Condenser Blade ($)	----	----	----	$18	$18	$1.04
	Baseline Cost of Evaporator Blade ($)	----	----	----	$5	$1.57	$1.04
	Cost Increase (%)	----	----	----	100%	100%	n/a
Hi-Eff Comp	Hi-Efficiency COP	----	----	----	4.14	1.20	2.15
	OEM Compressor Cost ($)	----	----	----	250	300	$10
ECM Comp Motor	Power Reduction (%)	----	----	----	----	----	18%
	OEM ECM Motor Cost ($)	----	----	----	----	----	$63
Variable Capacity Comp	Savings Improvement %	----	----	----	----	----	18%
	VC Controls Cost ($)	----	----	----	----	----	$63
Comp Modulation	Savings Improvement %	----	5.2%	----	----	----	----
Reduced	Energy Use Reduction	----	----	----	----	----	----

Thermal Cycling	(%)						
External Heat Rejection	Reduced Duty Cycle (%)	----	----	----	----	61%	----
	Cost of Installation ($)	----	----	----	----	$1,003	----
Hot Gas Anti-sweat	Perimeter of Doors (ft)	----	----	----	250	50	21
	Total Retooling Cost ($)	----	----	----	$2,000,000	$1,000,000	$1,000,000
Anti-Sweat Heater Control	Power Reduction (%)	----	----	----	33%	33%	----
	Sensor & Controller Cost ($)	----	----	----	$627	$627	----
Hot Gas Defrost	Cost per circuit ($)	----	----	----	----	$104	----
Defrost Control	Energy Reduction (%)	----	----	----	----	50%	----
	Cost of 2 Sensors & Controls ($)	----	----	----	----	$63	----
Lighting	Baseline OEM Bulb Cost ($/bulb)	11	----	----	$1.64	----	----
	Baseline OEM Ballast Cost ($/bulb)	14	----	----	$5.50	----	----
High Efficiency Fluorescent	Hi-Eff Bulb Power (W/bulb)	----	----	----	72.0	----	----
	Hi-Eff Ballast Power (W/bulb)	----	----	----	0	----	----
	Hi-Eff OEM Bulb Cost ($/bulb)	----	----	----	$3.68	----	----
	Hi-Eff OEM Ballast Cost ($/bulb)	----	----	----	$7.00	----	----
LED	Number of Bulbs	5	----	----	4 (60in)	----	----
	LED Power (W/bulb)	29	----	----	74.9	----	----
	LED OEM Cost ($/bulb)	$115	----	----	$86	----	----
Reduced Meltage During Harvest	Freeze Cycle Energy (kWh/yr)	----	----	----	----	----	----
	Energy Use Reduction (%)	----	----	----	----	----	----
	OEM Cost ($)	----	----	----	----	----	----
Floating Head Pressure	Reduced Duty Cycle (%)	----	----	----	50%	----	----
	Cost per circuit ($)	----	----	----	$348	----	----

Ambient Sub-cooling	Reduced Duty Cycle (%)	----	----	----	58%	----	----
	Refrigeration System Cost ($)	----	----	----	$439	----	----
Energy Management System	EMS Energy Reduction	----	----	----	----	----	----
Retooling	Annual Sales (units)	----	----	----	40000	40000	125,000
	% Retooled	----	----	----	5%	5%	10%

Table 4-5 Continued: Technology Option Parameter Definitions

Technology Option[1]	Parameter	Reach In Refrigerators	Reach In Freezers	Bev. Merchandisers	Ice Machines	Fully-Cooled Vending Machines	Zone-Cooled Vending Machines
Thickness Increase	Thickness Increase (in)	1.0	1.0	1.0	0.5	0.1	0.1
	Load Reduction (%)	29%	29%	38%	?	20%	10%
	Material Cost ($)	$0.41	$0.41	$0.41	$0.22	$18.87	$15.76
	Total Retooling Cost ($)	$1000000	$1000000	$1000000	$1000000	----	----
Evap Fan Control	Duty Cycle Reduction (%)	----	----	----	----	59%	64%
	OEM Cost of Contactor($)	----	----	----	----	$12	$12
High Efficiency Fan Blades	Power Reduction (%)	15%	15%	15%	15%	----	----
	Baseline Cost of Condenser Blade ($)	$1.04	$1.04	$1.04	$1.04	----	----
	Baseline Cost of Evaporator Blade ($)	$1.04	$1.04	$1.04	$1.04	----	----
	Cost Increase (%)	n/a	n/a	n/a	n/a	----	----
Hi-Eff Comp	Hi-Efficiency COP	2.55	1.67	2.15	925	504.4	----
	OEM Compressor Cost ($)	$10	$15	$10	$25	$6	----
ECM Comp Motor	Power Reduction (%)	15%	15%	15%	----	15%	15%
	OEM ECM Motor Cost ($)	$63	$69	$63	----	$63	$63
Variable	Savings Improvement	15%	15%	15%	----	15%	15%

Capacity Comp	%						
	VC Controls Cost ($)	$63	$63	$63	----	$63	$63
Comp Modulation	Savings Improvement %	----	----	----	----	----	----
Reduced Thermal Cycling	Energy Use Reduction (%)	----	----	----	4.5%	----	----
External Heat Rejection	Reduced Duty Cycle (%)	----	----	----	----	----	----
	Cost of Installation ($)	----	----	----	----	----	----
Hot Gas Anti-sweat	Perimeter of Doors (ft)	36	18	----	----	----	----
	Total Retooling Cost ($)	$1,000,000	$1,000,000	----	----	----	----
Anti-Sweat Heater Control	Power Reduction (%)	----	----	----	----	----	----
	Sensor & Controller Cost ($)	----	----	----	----	----	----
Hot Gas Defrost	Cost per circuit ($)	----	$104	----	----	----	----
Defrost Control	Energy Reduction (%)	----	45%	----	----	----	----
	Cost of 2 Sensors & Controls ($)	----	$63	----	----	----	----
Lighting	Baseline OEM Bulb Cost ($/bulb)	----	----	$1.64	----	$1.64	$5.63
	Baseline OEM Ballast Cost ($/bulb)	----	----	$5.50	----	$5.50	$7.75
High Efficiency Fluorescent	Hi-Eff Bulb Power (W/bulb)	----	----	30.1	----	30.1	----
	Hi-Eff Ballast Power (W/bulb)	----	----	1.4	----	1.4	----
	Hi-Eff OEM Bulb Cost ($/bulb)	----	----	$3.68	----	$3.68	----
	Hi-Eff OEM Ballast Cost ($/bulb)	----	----	$7.00	----	$7.00	----
LED	Number of Bulbs	----	----	2 (48 in)	----	2(48in)	2 (48in)
	LED Power (W/bulb)	----	----	11.6	----	11.6	33.8
	LED OEM Cost ($/bulb)	----	----	$62	----	$62	$210

Reduced Meltage During Harvest	Freeze Cycle Energy (kWh/yr)	----	----	----	4251	----	----
	Energy Use Reduction (%)	----	----	----	5%	----	----
	OEM Cost ($)	----	----	----	$50	----	----
Floating Head Pressure	Reduced Duty Cycle (%)	----	----	----	----	----	----
	Cost per circuit ($)	----	----	----	----	----	----
Ambient Sub-cooling	Reduced Duty Cycle (%)	----	----	----	----	----	----
	Refrigeration System Cost ($)	----	----	----	----	----	----
Energy Management System	EMS Energy Reduction	----	----	35%	----	20%	20%
Retooling	Annual Sales (units)	261000	51500	----	----	----	----
	% Retooled	10%	15%	----	----	----	----

4.2 Supermarket Refrigeration Systems

This section presents the energy savings potential of the energy savings technologies applicable to supermarket refrigeration systems and the barriers to implementation of these technologies. Refer to section 4.1 for detailed calculations of the energy savings and economics.

4.2.1 Economic Analysis

The economic analysis for supermarket refrigeration systems is presented in Table 4-6 through **Table 4-11**, which summarize the installed cost premium, annual energy savings, and the simple payback period for each technology considered for refrigerated display cases, compressor racks, condensers, and the central control system. The energy savings percentage compares the reduced energy to the baseline energy use. Refer back to Chapter 3 for descriptions of baseline equipment.

Refrigerated Display Cases

The baseline unit is a typical new medium-temperature vertical case with transparent doors and an energy consumption of 9,107 kWh/yr (DOE 2009a).

Table 4-6: Economic Analysis for Medium-Temp Open Display Cases

	Technology Option	Installed- Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 1.0 - 1.125 in	$76	109	1%	4.3	7.0	9.7
2	PSC Evap. Fan Motor (6W, 5 fans)	$50	916	11%	0.3	0.6	0.8
3	ECM Evap. Fan Motor (6W, 5 fans)	$171	2,150	25%	0.5	0.8	1.1
4	Lighting, High-Lumen Bulb, Low BF	$121	531	1%	1.4	2.3	3.2
5	LED Lighting	$5,434	1,678	19%	19.9	32.6	45.6
6	Enhanced Evaporator Coil [1]	$355	469	5%	4.6	7.6	10.6
	Max Tech (2,3,4)	$343	2,048	26%	1.0	1.7	2.4

Source: DOE 2009a
[1] Higher UA value of evaporator coil due to increased fin pitch and surface area.

Table 4-7: Economic Analysis for Low-Temp Glass-Door Display Cases

	Technology Option	Installed- Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 1.0 to 1.125 in	$76	205	2%	2.3	3.7	5.2
2	PSC Evap. Fan Motor (6W, 5 fans)	$42	594	7%	0.4	0.7	1.0
3	ECM Evap. Fan Motor (6W, 5 fans)	$143	1,333	15%	0.7	1.1	1.5
4	LED Lighting	$808	2,219	26%	2.2	3.7	5.1
5	High Performance Doors [1]	$1,825	4,763	55%	2.4	3.9	5.4
6	Enhanced Evaporator Coil [2]	$159	153	2%	6.4	10.4	14.6
	Max Tech (1 - 5)	$2,893	6,596	44%	2.7	4.4	6.2

Source: DOE 2009a
[1] Install high-performance doors with more efficient anti-sweat heating, vinyl/composite frame, and high-performance glass.
[2] Higher UA value of evaporator coil due to increased fin pitch and surface area.

Table 4-8: Economic Analysis for Medium-Temp Glass-Door Display Cases

	Technology Option	Installed- Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 1.0 to 1.125 in	$76	95	1%	4.9	8.0	11.2
2	PSC Evap. Fan Motor (6W, 5 fans)	$42	509	6%	0.5	0.8	1.2
3	ECM Evap. Fan Motor (6W, 5 fans)	$143	1,143	13%	0.8	1.3	1.8
4	LED Lighting	$808	2,219	26%	2.2	3.7	5.1
5	High Performance Doors [1]	$1,435	2,322	27%	66.7	109.2	152.6
6	Enhanced Evaporator Coil [2]	$94	66	1%	8.8	14.4	20.2
	Max Tech (2 - 4)	$992	2,694	31%	2.3	3.7	5.2

Source: DOE 2009a
[1] Install high-performance doors with more efficient anti-sweat heating, vinyl/composite frame, and high-performance glass.
[2] Higher UA value of evaporator coil due to increased fin pitch and surface area.

Compressor Racks
The baseline unit is a compressor rack consisting of 5 compressors operating at medium temperature, each having a capacity of approximately 75,000 Btu/hr and an annual energy use of 1,000,000 kWh/yr. The load on the rack is 350,000 Btu/hr, with the compressors sharing the load evenly. The compressors use refrigerant 404A.

Table 4-9: Economic Analysis of Compressor Racks

	Technology Option	Installed-Cost Premium[1]	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Vapor Injection Economizer for Low Temp [2]	$19,008	33,782	4%	3.5	5.7	7.9
2	Compressor Modulation (Digital) for Medium Temp [3]	$19,008	46,750	5%	2.5	4.1	5.7
	Max Tech (1 & 2)	$38,016	80,532	9%	2.9	4.8	6.6

[1] Both compressor technologies are assumed to have approximately 2-3 year payback. Cost premiums are estimated accordingly (Emerson 2009)
[2] Emerson ZF**KVE Scroll w/Vapor Injection (Low Temp)
[3] 8.5% decrease in energy use from Emerson (June 2009)

Supermarket Condensers
The baseline unit is a set of 4 condensers with 1,520 MBtu capacity and an annual energy use of 120,000 kWh/yr.

Table 4-10: Economic Analysis of Supermarket Condensers

Technology Option	Installed - Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
				High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
High Efficiency Condenser	$5,000	33,113	28%	0.9	1.5	2.1
Source: EIA 2008						

Supermarket Control System

The baseline supermarket includes 60 display cases, compressor racks, and four condensers with the characteristics described above. The overall annual energy use of the store is 2,037,000 kWh/yr.

Table 4-11: Economic Analysis of Supermarket Control System

Technology Option	Installed - Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
				High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
Case Controller & EEVs	$90,000	335,318	20%	1.7	2.7	3.8
Source: Lazzarin 2008						

Supermarket Walk-ins

In order to quantify the savings for supermarket walk-ins, we assume that the “maximum technology” savings achievable for stand-along walk-ins (see section 4.3.1) are also achievable for supermarket walk-ins except for the condenser and compressor savings. Therefore we calculate that supermarket walk-in coolers can save 46% and freezers 31%.

Supermarket Max Tech Combination Option

Table 4-12 shows the highest possible energy savings potential for supermarket combining the max tech options for each of the components above. Note that the savings from above are not additive. The total refrigeration load is reduced by an average of 29 percent when all design options are applied.[15] Before applying the savings percentages to compressors and condensers,

[15] The medium temp open door cases (VOP.RC.M) were reduced by 8% and the low temp glass door cases (VCT.RC.M) were reduced by 38% (DOE 2009a). The average of these two percentages, weighted by case load (30% low temp and 70% medium temp), is 29%.

we reduced their baseline energy consumption by 29 percent to account for the lower case load resulting from the display case improvements. Applying all technologies with payback periods of under 7 years yields supermarket primary energy savings of 196 TBtu/yr.

Table 4-12: Max Tech Supermarket Combination

Technology Option	"Typical New" UEC (kWh/yr)	Energy Reduction (%)	Max Tech UEC (kWh/yr)	Store Energy Savings (kWh/yr)	Total Electricity Savings (TWh/yr)	Primary Energy Savings (TBtu/yr)
Display Cases	8,894	35%	5,799	185,695	6.5	68
Compressor Racks	637,681 [1]	9%	580,621	57,060	2.0	21
Condensers	85,024 [1]	28%	61,563	23,462	0.8	9
Walk-ins[2]	Varies		Varies	59,119	2.1	22
Total Refrig System						118
Add Case Controllers & EEVs [3]				212,037	7.4	77
Total Max Tech Supermarket Savings:						**196**

Source: Lazzarin 2008
[1] Reduced by 29%
[2] See Table 4-16 for detail on walk-in savings calculation.
[3] 20% energy reduction of store refrigeration energy use

4.2.2 Barriers to Implementation in Supermarkets

Key barriers to improving supermarket refrigeration system efficiency include:

1) Display Cases: Supermarket chain marketing groups select display cases based on how well they present the product, i.e., to maximize sales. Energy consumption is often not considered. In fact, some of the characteristics that improve product presentation run counter to energy-efficient design. For example, display cases often forego doors (to make product more attractive and make it easier for shoppers to pick up), and they generally have aspect ratios resulting in large surface areas compared to the volume of product stored. Return on investment is considered better for design features which enhance sales than for energy-saving features.

2) Supermarkets tend to operate on very narrow margins and generally have limited capital for making energy-efficiency investments. Paybacks of 2 to 3 years are required by most supermarkets. Some supermarkets even require paybacks of less than one year.

3) Reliability (and the perception of reliability) is extremely important. As mentioned in section 3.1.4, refrigerated goods are estimated to represent approximately 45% of

supermarket sales, and at a given point, the value of refrigerated inventory in a supermarket generally ranges from $200,000 to $300,000 depending on store size (TIAX 2005). Unproven technology is not readily accepted. New technologies will have to be field tested and proven before they are generally accepted. Some large supermarket chains test new technologies in demonstration stores to show reliability before deploying the new technology widely.

4) Implementation of Energy Conservation Standards: Design practices for supermarket refrigeration systems vary widely and require significant site-specific engineering, making it difficult for DOE to establish energy conservation standards. Interactions with the supermarket HVAC system and building shell further complicate establishing standards.

5) Evaporative Condensers add maintenance and water costs when compared with air-cooled condensers. This technology does not have significant market penetration except in dry areas such as the southwest. In contrast, evaporative condensers are used almost exclusively in warehouse and food processing refrigeration applications, resulting in a 95°F typical design condenser temperature (compared with 110°F to 115°F for supermarkets). The desire to keep maintenance costs to a minimum is a key issue with supermarkets.

6) Display Cases: It can be difficult and expensive to incorporate LED lighting in display cases. Certain food products require specific lighting color and quality to maximize their customer appeal. Obtaining consistent lighting color and quality requires LED manufacturers to select LED chips by hand, which raises the price of LEDs for commercial refrigeration equipment.

4.3 Walk-In Cooler & Freezers

This section presents the energy savings potential of the energy savings technologies applicable to walk-in coolers and freezers and the barriers to implementation of these technologies. Refer to section 4.1 for detailed calculations of the energy savings and economics.

4.3.1 Economic Analysis

The results of the economic analysis for walk-ins is summarized in Table 4-13 and Table 4-14, including installed-cost premium, annual energy savings, and the simple payback period for each technology considered for walk-in coolers and freezers, respectively. The energy savings percentage compares the energy saved to the baseline energy use. Refer back to Chapter 3 for descriptions of baseline equipment.

Walk-in Coolers

The baseline unit is a typical new self-contained walk-in cooler with an annual energy consumption of 42,182 kWh/yr.

Table 4-13: Economic Analysis for Walk-In Coolers

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Floating Head Pressure	$348	6,882	16%	0.3	0.5	0.7
2	Ambient Subcooling	$439	3,441	8%	0.8	1.3	1.8
3	Economizer Cooling	$4,702	1,781	4%	16.2	26.6	37.1
4	Anti-sweat Heat Controls	$627	1,004	2%	3.8	6.3	8.8
5	Thicker Insulation - 4 to 5 in	$592	191	0%	19.1	31.3	43.7
6	Evaporator Fan Control	$138	1,993	5%	0.4	0.7	1.0
7	PSC Evaporator Fan Motors (37W)	$221	3,623	9%	0.4	0.6	0.9
8	ECM Evaporator Fan Motors (37W)	$487	5,525	13%	0.5	0.9	1.2
9	ECM Condenser Fan Motors (373W)	$83	925	2%	0.6	0.9	1.3
10	Lighting, High-Lumen Bulb, Low BF	$31	129	0%	1.5	2.4	3.4
11	Lighting, LED, 3500 K	$718	1,592	4%	2.8	4.5	6.3
12	High Efficiency Fan Blades	$166	2,414	6%	0.4	0.7	1.0
13	Non-Electric Antisweat	$1,244	2,628	6%	2.9	4.8	6.7
14	Strip Curtains	$73	3,730	9%	0.1	0.2	0.3
15	Low Heat/No Heat Doors	$876	3,130	7%	1.7	2.8	3.9
16	Auto Door Closer	$142	3,535	8%	0.2	0.4	0.6
17	High Efficiency Compressor	$552	3,863	9%	0.9	1.4	2.0
	Max Tech (1,2,6,8,9,11,13,14)	$1,568	22,778	54%	0.4	0.7	1.0

Walk-In Freezers

The baseline unit is a typical new, self-contained walk-in freezer with an annual energy consumption of 15,524 kWh/yr.

Table 4-14: Economic Analysis for Walk-In Freezers

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	External Heat Rejection	$1,003	1,399	9%	4.4	7.2	10.1
2	Hot Gas Defrost	$104	589	4%	1.1	1.8	2.5
3	Defrost Controls	$138	367	2%	2.3	3.8	5.3
4	Anti-Sweat Heat Controls	$627	1,007	6%	3.8	6.3	8.8
5	Thicker Insulation - 4 to 5 in	$322	566	4%	3.5	5.7	8.0
6	Evaporator Fan Controls	$138	631	4%	1.3	2.2	3.1
7	PSC Evaporator Fan Motors (20W, 2 fans)	$83	1,682	11%	0.3	0.5	0.7
8	ECM Evaporator Fan Motors (20W, 2 fans)	$138	2,208	14%	0.4	0.6	0.9
9	PSC Condenser Fan Motors (125W)	$30	779	5%	0.2	0.4	0.5
10	ECM Condenser Fan Motors (125W)	$66	1,067	7%	0.4	0.6	0.9
11	High Efficiency Fan Blades	$46	776	5%	0.4	0.6	0.8
12	Hot Gas Antisweat	$414	2,015	13%	1.3	2.1	2.9
13	Strip Curtains	$73	3,730	24%	0.1	0.2	0.3
14	Low Heat/No Heat Doors	$876	3,130	20%	1.7	2.8	3.9
15	Auto Door Closer	$142	3,535	23%	0.2	0.4	0.6
16	High Efficiency Compressor	$662	1,477	10%	2.8	4.5	6.3
	Max Tech (1,2,6,8,10,12)	$1,523	8,310	54%	1.1	1.8	2.6

Table 4-15 and **Table 4-16** display the savings from replacement of typical installed with typical new equipment and from applying currently available technologies to the typical new equipment to reach “max tech”, for non-supermarket walk-ins and supermarket walk-ins respectively.

Table 4-15: National Energy Savings Potential for Non-Supermarket Walk-ins

	UEC (kWh/yr)	Total Electricity Consumption (TWh/yr)	Primary Energy Consumption (TBtu/yr)	Primary Energy Savings (TBtu/yr)
Walk-in Coolers				
Typical Installed	16,200	7.6	78.9	n/a
Typical New [1]	16,152	7.6	78.7	0.2
Max Tech [2]	7,452	3.5	36.3	42.6
Walk-In Freezers				
Typical Installed	21,400	5.0	52.1	n/a
Typical New [1]	21,364	5.0	52.0	0.1
Max Tech [2]	9,944	2.3	24.2	27.9
Combination Walk-Ins				
Typical Installed	30,200	1.6	16.6	n/a
Typical New [1]	30,130	1.6	16.6	0.0
Max Tech [2]	13,963	0.7	7.7	8.9
Total Savings:				**80**

[1]Typical New uses T-8 instead of T-12 fluorescent lighting.
[2] Max Tech shows 54% improvement over typical new

Table 4-16: National Energy Savings Potential for Supermarket Walk-ins

	Store Energy Consumption (kWh/yr)	Total Electricity Consumption (TWh/yr)	Primary Energy Consumption (TBtu/yr)	Primary Energy Savings (TBtu/yr)
Meat Coolers				
Typical Installed/Typical New [1]	17,432	0.6	6.3	n/a
Max Tech [2]	9,488	0.3	3.5	2.9
Other Coolers				
Typical Installed/Typical New [1]	89,586	3.1	32.6	n/a
Max Tech [2]	48,757	1.7	17.8	14.9
Freezers				
Typical Installed/Typical New [1]	32,887	1.2	12.0	n/a
Max Tech [2]	22,542	0.8	8.2	3.8
Total Savings:				**22.1**

[1]Typical New is assumed to be equal to typical installed.
[2] Max Tech shows 46% improvement over typical new for coolers, 31% for freezers;

4.3.2 Barriers to Implementation for Improved-Efficiency Walk-Ins

Barriers to widespread deployment of energy-savings technologies in walk-ins include:

1) Purchase decisions for walk-ins are generally not made based on life-cycle cost or payback considerations. A general contractor installing a walk-in has incentive to select the lowest cost equipment that meets energy efficiency specifications. Frequently, there is insufficient cash flow at the time of equipment purchase for consideration of future benefits to sway the decision. As discussed in previous sections, the long-term prospects for new start-up restaurants are not solid, and these establishments generally select the lowest-cost equipment. In many cases, used refrigeration equipment is purchased, if available.

2) It can be difficult for end-users to properly assess whether the added cost of energy-saving technologies will be recovered quickly enough through savings. Complicating factors include:

    - The complexity of refrigeration systems can make energy savings difficult to predict.
    - The complexity of commercial electric rate structures. Commercial electric rates can be complex, having both demand and energy usage components, which can vary with time of electricity usage. A further complication arises for convenience stores or restaurants chains that seek to apply standard design specifications across their establishments, but face differing electric rate structures in different locations.
    - Emergency replacements severely limit the time available to evaluate equipment alternatives, and make financing more difficult.

3) The walk-in market is very competitive, with many suppliers, none of whom have a dominant market position. First cost is generally the primary basis of differentiation among competitors. In addition, there are many supply options: an end-user can purchase the walk-in box from a walk-in manufacturer and purchase the refrigeration equipment elsewhere. Or, the entire system can be purchased from the walk-in manufacturer. Installation can be provided by the walk-in manufacturer or by a refrigeration contractor. Small walk-ins can be purchased as prefabricated units or can be assembled on-site.

    - Walk-ins generally consist of an insulated box section having coolers mounted inside it. There is minimal integration of these two parts of the walk-in (besides proper location of the evaporator within the box), and they are generally manufactured by different companies. This makes implementation of non-electric anti-sweat difficult, because it would require the walk-in box manufacturer to install refrigerant tubing and provide connections for the refrigeration system.

4) A number of market structure barriers hinder the increased use of energy saving technologies for walk-ins.

- Several of the technologies discussed (i.e. floating head pressure, ambient subcooling, demand defrost control, and evaporator fan shutdown) represent additional complexity for the refrigeration system and its control. Training would be required for most refrigeration service technicians providing service for walk-in systems.
- As mentioned in the beginning of Section 4.3, the use of floating head-pressure control would require the use of balanced-port expansion valves to allow satisfactory refrigerant flow over a range of head pressures. Implementing floating head pressure control would require coordination among the refrigeration controls manufacturer, the refrigeration system manufacturer, and the walk-in manufacturer. Such cooperation is possible, but takes initiative and represents a barrier to implementation.
- ECM motors are not widely available in the sizes required for walk-in fans. Even if a unit cooler with ECM motors was installed, finding a replacement motor would be difficult, and represent additional down time. These motors will have to break into the market and develop a larger supply network before the risk of not being able to quickly find a replacement is diminished.
- The market for walk-ins is fairly fragmented. Many manufacturers must successfully introduce an energy-saving technology to capture a significant portion of the market. Furthermore, there is no trade association that represents the walk-in manufacturers that would provide a forum for discussion of technical issues for this application. There has in the past been insufficient interest in such an association among manufacturers.

4.4 Refrigerated Food Service Equipment

This section presents the energy savings potential of the energy savings technologies applicable to refrigerated food service equipment and the barriers to implementation of these technologies. Refer back to section 4.1 for detailed calculations of the energy savings and economics.

4.4.1 Economic Analysis

The economic analysis for refrigerated food service equipment is summarized in **Table 4-17**, including the installed-cost premium, annual energy savings, and the simple payback period for each technology considered for food service equipment. The energy savings percentage compares the energy saved to the baseline energy use. Refer back to Chapter 3 for descriptions of baseline equipment.

The baseline unit is a typical new, self-contained preparation table with an annual energy consumption of 2,341 kWh/yr.

Table 4-17: Economic Analysis for Refrigerated Food Service Equipment

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 2.25 to 3.25 in	$106	71	3%	9.2	15.1	21.1
2	PSC Evaporator Fan Motors (6W)	$10	343	15%	0.2	0.3	0.4
3	ECM Evaporator Fan Motors (6W)	$23	507	22%	0.3	0.5	0.6
4	ECM Condenser Fan Motors (9W)	$23	179	8%	0.8	1.3	1.8
5	High Efficiency Compressor	$22	233	10%	0.6	1.0	1.3
6	Variable Speed Compressor	$138	438	19%	1.9	3.2	4.4
7	ECM Compressor Motor	$201	219	9%	5.6	9.3	12.9
8	Hot Gas Anti-Sweat	$80	427	18%	1.2	1.9	2.6
9	High Efficiency Fan Blades	$7	98	4%	0.4	0.7	1.0
	Max Tech (3,4,5,8)	$379	1,306	56%	1.8	2.9	4.1

Table 4-18 displays the savings from replacement of typical installed with typical new equipment and from applying currently available technologies to the typical new equipment to reach "max tech".

Table 4-18: National Energy Savings Potential for Food Service Equipment

	Shipment-weighted Avg UEC (kWh/yr)	Total Electricity Consumption (TWh/yr)	Primary Energy Consumption (TBtu/yr)	Primary Energy Savings (TBtu/yr)
Typical Installed	3,478	5.27	55	n/a
Typical New [1]	3,162	4.79	50	5
Max Tech [2]	1,399	2.12	22	28
Total Savings:				**33**

[1]Typical New is assumed to use 10% less energy than typical installed.
[2] Max Tech shows 56% improvement over typical new

4.4.2 Barriers to Implementation for Improved-Efficiency Food Service Equipment

Space constraints limit acceptable cabinet insulation thicknesses. Therefore, simply increasing insulation thickness may not be acceptable.

The first-cost barrier can be substantial for food-service equipment. Many food-service establishments are very cash limited. Hence, purchasers of food service equipment may be unwilling to pay more for improved energy efficiency. Furthermore, food-service establishments often don't even consider energy costs when making purchase decisions. Even food-service chains that, in aggregate, have energy costs that would normally justify consideration of the economics of energy savings, may not do so because the franchisee (typically owning one or possibly two restaurants) pays energy costs.

4.5 Reach-Ins

This section presents the energy savings potential of the energy savings technologies applicable to reach-in refrigerators and freezers and the barriers to implementation of these technologies. Refer back to section 4.1 for detailed calculations of the energy savings and economics.

4.5.1 Economic Analysis

The economic analysis for reach-ins is presented in Table 4-19 and Table 4-20, which summarize the end-user cost premium, annual energy reduction, and the simple payback period for each technology considered for reach-in freezers and refrigerators, respectively. The energy savings percentage compares the energy saved to the baseline energy use. Refer to Section 3 above for descriptions of baseline equipment.

Reach-in Freezers

The baseline unit is assumed to be a typical new reach-in freezer with solid doors with an estimated annual energy consumption of 3,960 kWh/yr.

Table 4-19: Economic Analysis for Reach-In Freezers

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 2.25 to 3.25 in	$144	195	5%	4.6	7.5	10.4
2	Improved Insulation	$1,429	81	2%	108.3	177.4	247.8
3	ECM Evaporator Fan Motor (9W)	$23	274	7%	0.5	0.9	1.2
4	ECM Condenser Fan	$33	138	3%	1.5	2.4	3.4

	Motor (37W)						
5	High Efficiency Compressor	$33	544	14%	0.4	0.6	0.9
6	ECM Compressor Motor	$152	465	12%	2.0	3.3	4.6
7	Variable Speed Compressor	$215	535	14%	2.5	4.0	5.7
8	Hot Gas Defrost	$165	329	8%	3.1	5.1	7.1
9	Hot Gas Anti-Sweat	$121	372	9%	2.0	3.3	4.6
10	Defrost Control	$63	148	4%	2.6	4.3	6.0
11	High-Efficiency Fan Blades	$5	116	3%	0.2	0.4	0.6
	Max Tech (3,4,5,9)	$575	1,299	33%	2.7	4.5	6.2

Reach-in Refrigerators

The baseline unit is assumed to be a typical new reach-in refrigerator with solid doors with an estimated annual energy consumption of 2,477 kWh/yr.

Table 4-20: Economic Analysis for Reach-In Refrigerators

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 2.25 to 3.25 in	$131	96	4%	8.3	13.7	19.1
2	Improved Insulation	$423	40	2%	64.9	106.4	148.6
3	ECM Evaporator Fan Motor (9W, 2 fans)	$47	454	18%	0.6	1.0	1.5
4	ECM Condenser Fan Motor (20W)	$30	359	14%	0.5	0.9	1.2
5	High Efficiency Compressor	$22	171	7%	0.8	1.3	1.8
6	ECM Compressor Motor	$138	288	12%	3.0	4.8	6.8
7	Variable Speed Compressor	$201	331	13%	3.7	6.1	8.5
8	Hot Gas Anti-Sweat	$121	434	18%	1.7	2.8	3.9
9	High-Efficiency Fan Blades	$7	171	7%	0.2	0.4	0.6
	Max Tech (3,4,5,8)	$351	1,381	56%	1.6	2.6	3.6

Table 4-21 displays the savings from replacement of typical installed with typical new equipment and from applying currently available technologies to the typical new equipment to reach "max tech".

Table 4-21: National Energy Savings Potential for Reach-ins

	Shipment-weighted Avg UEC (kWh/yr)	Total Electricity Consumption (TWh/yr)	Primary Energy Consumption (TBtu/yr)	Primary Energy Savings (TBtu/yr)
Reach-in Freezers				
Typical Installed[1]	4,158	4.8	50	n/a
Typical New	3,960	4.6	47.6	2.4
Max Tech [2]	2,662	3.1	32.0	15.6
Reach-In Refrigerators				
Typical Installed[1]	3,455	5.4	55.9	n/a
Typical New	2,477	3.9	40.1	15.8
Max Tech [2]	1,096	1.7	17.7	22.4
Total Savings:				**38**
[1]Typical Installed is assumed to use 20% less energy than reported in ADL1996. [2] Max Tech shows 56% improvement over typical new for reach-ins refrigerators, 33% improvement for reach-in freezers.				

4.5.2 Barriers to Implementation for Improved-Efficiency Reach-Ins

1) High-efficiency equipment with a cost premium is undesirable for start-up restaurants, for which investment capital is limited. If the payback is not extremely quick, such equipment is typically not considered.
2) There are a relatively high number of reach-in manufacturers, and each must meet the needs of a wide range of end-users. The resulting low production volumes make it difficult to implement efficiency improvements.

4.6 Beverage Merchandisers

This section presents the energy savings potential of the energy savings technologies applicable to beverage merchandisers and the barriers to implementation of these technologies. Refer to section 4.1 for detailed calculations of the energy savings and economics.

4.6.1 Economic Analysis

Table 4-22 summarizes the economic analysis for beverage merchandisers, including the installed-cost premium, annual energy savings, and the simple payback period for each alternative technology considered, compared to the baseline equipment. The energy reduction percentage compares the energy savings to the baseline energy use. Refer back to Chapter 3 for descriptions of baseline equipment.

The baseline equipment is a typical new one-door beverage merchandiser with an annual energy consumption of 2,527 kWh/yr.

Table 4-22: Economic Analysis for Beverage Merchandisers

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 1.5 to 2.5 in	$80	116	5%	4.3	7.0	9.8
2	Improved Insulation	$345	37	1%	58.0	95.1	132.8
3	ECM Evaporator Fan Motors (9W)	$47	482	19%	0.6	1.0	1.4
4	ECM Condenser Fan Motors (9W)	$23	69	3%	2.1	3.4	4.8
5	High Efficiency Compressor	$22	183	7%	0.7	1.2	1.7
6	ECM Compressor Motor	$138	206	8%	4.1	6.8	9.4
7	Variable Speed Compressor	$201	237	9%	5.2	8.5	11.9
8	Lighting, High-Lumen Bulb, Low BF	$16	49	2%	2.0	3.2	4.5
9	Lighting, LED, 3500 K	$161	551	22%	1.8	2.9	4.1
10	High Efficiency Fan Blades	$7	252	10%	0.2	0.3	0.4
11	Smart Proximity Sensor	$100	884	35%	0.7	1.1	1.6
	Max Tech (1,3,4,7,9)	$620	1,386	55%	2.7	4.5	6.3

Table 4-23 displays the savings from replacement of typical installed with typical new equipment and from applying currently available technologies to the typical new equipment to reach “max tech”.

Table 4-23: National Energy Savings Potential for Beverage Merchandisers

	Shipment-weighted Avg UEC (kWh/yr)	Total Electricity Consumption (TWh/yr)	Primary Energy Consumption (TBtu/yr)	Primary Energy Savings (TBtu/yr)
One-Door				
Typical Installed	3,076	1.4	14.7	n/a
Typical New [1]	2,527	1.2	12.1	2.6
Max Tech [2]	1,141	0.5	5.5	6.6
Two-Door				
Typical Installed	6,080	2.5	26.2	n/a
Typical New [1]	3,489	1.4	15.0	11.2
Max Tech [2]	1,575	0.7	6.8	8.2
Three-Door				
Typical Installed	8,960	0.4	4.3	n/a
Typical New [1]	4,592	0.2	2.2	2.1
Max Tech [2]	2,073	0.1	1.0	1.2
Total Savings:				**32**

[1] Typical Installed is assumed to use 20% less energy than is reported in ADL 1996.
[2] Max Tech shows 55% improvement over typical new

4.6.2 Barriers to Implementation for Improved-Efficiency Beverage Merchandisers

1) Majority of beverage merchandisers are owned by bottling companies, such as Coca Cola. The bottling companies do not pay utility bills for the buildings where the merchandisers are located, which eliminates incentive for them to reduce energy consumption.

2) Energy costs are small compared with beverage sales revenues, so manufacturers tend to overlook energy issues in favor of sales-boosting design changes such as increases in lighting intensity.

3) Increasing insulation thickness is difficult due to space constraints in beverage merchandisers. Reducing the storage capacity of a given machine is generally not acceptable.

4) Engineering and tooling costs associated with manufacturing commercial equipment are not easily absorbed due to low production volumes.

5) Rating standards such as the DOE efficiency standard are based on steady state operation, so technologies such as energy management systems that turn off lights or allow temperatures to float do not receive credit for energy savings using those test procedures. Similarly, variable speed compressors save energy at off-design conditions rather than at

rating conditions, so manufacturers have limited incentive to implement these expensive technologies, even if they save substantial amounts of energy.

4.7 Ice Machines

This section presents the energy savings potential of the energy savings technologies applicable to ice machines and the barriers to implementation of these technologies. Refer back to section 4.1 for detailed calculations of the energy savings and economics.

4.7.1 Economic Analysis

The economic analysis for ice machines is summarized in Table 4-24, including the installed-cost premium, annual energy savings, and the simple payback period for each energy-saving technology considered. The energy reduction percentage compares the energy saved to the baseline energy use. Refer back to Chapter 3 for descriptions of baseline equipment.

The baseline unit is a typical new self-contained, air-cooled ice machine with an annual energy consumption of 5,248 kWh/yr. Refer to Section 4 above for a description of the baseline unit.

Table 4-24: Economic Analysis for Ice Machines

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 0.5 to 1 in	$77	157	3%	3.0	4.9	6.9
2	PSC Condenser Fan Motor (25W)	$33	200	4%	1.0	1.7	2.3
3	ECM Condenser Fan Motor (25W)	$64	273	5%	1.4	2.3	3.3
4	High Efficiency Compressor	$55	306	6%	1.1	1.8	2.5
5	Reduced Meltage During Harvest	$110	213	4%	3.2	5.2	7.3
6	Reduced Evaporator Thermal Cycling	$25	236	5%	0.7	1.1	1.5
7	High Efficiency Fan Blades	$2	61	1%	0.2	0.4	0.5
8	Max Tech (1, 3-7)	$334	1,221	23%	1.7	2.8	3.8

Table 4-25 displays the savings from replacement of typical installed with typical new equipment and from applying currently available technologies to the typical new equipment to reach "max tech".

Table 4-25: National Energy Savings Potential for Ice Machines

	Shipment-weighted Avg UEC (kWh/yr)	Total Electricity Consumption (TWh/yr)	Primary Energy Consumption (TBtu/yr)	Primary Energy Savings (TBtu/yr)
Typical Installed [1]	5,429	8.10	84.2	n/a
Typical New [1]	5,248	7.8	81.4	2.8
Max Tech [2]	4,027	6.0	62.5	18.9
Total Savings:				**22**

[1]Typical Installed is assumed to use 15% less energy than is shown in ADL 1996 (7 kWh/100lbs ice).
[2] Max Tech shows 23% improvement over typical new

4.7.2 Barriers to Implementation for Improved-Efficiency Ice Machines

1) The trend in ice machines, as with other commercial refrigeration equipment, is for reduced physical size. This trend makes increase in insulation thickness and installation of purge water interchangers more difficult to implement.
2) Reductions in purge water amounts have the potential to decrease energy use. However, such reductions are associated with the risk of increased scale buildup, which can in itself reduce efficiency. If the purge water flow is too low, frequent cleaning of the machines is required to eliminate scale and reduce the risk of waterborne diseases.
3) Manufacturers generally require paybacks of at most 1 to 2 years for retooling costs associated with design modifications (the engineering costs are usually not taken into consideration in evaluating changes). This makes manufacturers reluctant to implement product changes if they cannot quickly recover investment costs through increased product prices. The competitiveness of the market makes such price increases difficult to obtain, even if the payback to end-users through energy savings is swift.

4.8 Refrigerated Vending Machines

This section presents the energy savings potential of the energy savings technologies applicable to refrigerated vending machines and the barriers to implementation of these technologies. Refer back to section 4.1 for detailed calculations of the energy savings and economics.

4.8.1 Economic Analysis

The economic analysis for refrigerated vending machines is presented below in **Table 4-26** and **Table 4-27**, which summarize the end-user cost premium, annual energy reduction, and the simple payback period for each technology considered for fully-cooled and zone-cooled refrigerated vending machines, respectively. The energy savings percentage compares the energy saved to the baseline energy use. Refer back to Chapter 3 for descriptions of baseline equipment.

Fully-Cooled Refrigerated Vending Machines

The baseline unit is a typical new fully-cooled refrigerated vending machine with an annual energy consumption of 2,494 kWh/yr.

Table 4-26: Economic Analysis for Refrigerated Vending Machines (Fully-Cooled)

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 1.0 to 1.125 in	$42	311	12%	0.8	1.3	1.9
2	ECM Evap. Fan Motor (9W)	$23	238	10%	0.6	1.0	1.4
3	ECM Cond. Fan Motor (9W)	$33	95	4%	2.1	3.5	4.9
4	Low BF T8 Lighting (4 bulbs)	$16	55	2%	1.8	2.9	4.0
5	LED Lighting	$242	546	22%	2.7	4.5	6.2
6	Evaporator Fan Controller	$26	252	10%	0.6	1.1	1.5
7	Enhanced Glass Pack	$446	139	6%	19.7	32.3	45.1
8	Enhanced Condenser Coil	$28	101	4%	1.7	2.8	3.9
9	Hi-h Single Speed Hermetic Compressor	$13	47	2%	1.6	2.7	3.7
10	ECM Compressor Motor	$138	236	9%	3.6	5.9	8.2
11	Variable Speed Compressor	$201	272	11%	4.6	7.5	10.4
12	Energy Management System	$100	499	20%	1.2	2.0	2.8
	Max Tech (1,2,3,5,6,8,9,12)	$507	1,353	54%	2.3	3.8	5.3

Zone-Cooled Refrigerated Vending Machines

The baseline unit is a typical new zone-cooled refrigerated vending machine with an annual energy consumption of 2,258 kWh/yr.

Table 4-27 Economic Analysis for Refrigerated Vending Machines (Zone-Cooled)

	Technology Option	Installed-Cost Premium	Energy Savings (kWh/yr)	Energy Savings (%)	Simple Payback Periods (yrs)		
					High Rate ($0.1627 /kWh)	Medium Rate ($0.0993 /kWh)	Low Rate ($0.0711 /kWh)
1	Thicker Insulation - 1.0 to 1.125 in	$35	55	2%	3.9	6.4	8.9
2	ECM Evap. Fan Motor (9W, 2 fans)	$47	487	22%	0.6	1.0	1.4
3	ECM Cond. Fan Motor (9W)	$23	46	2%	3.2	5.2	7.2
4	LED Lighting	$865	739	33%	7.2	11.8	16.5
5	Evaporator Fan Controller	$26	554	25%	0.3	0.5	0.7
6	Enhanced Condenser Coil	$33	99	4%	2.0	3.3	4.7
7	ECM Compressor Motor	$138	176	8%	4.8	7.9	11.0
8	Variable Speed Compressor	$201	203	9%	6.1	10.0	14.0
9	Energy Management System	$100	452	20%	1.4	2.2	3.1
	Max Tech (1,2,3,5,6,9)	$264	751	33%	2.2	3.5	5.0

Table 4-28 displays the savings from replacement of typical installed with typical new equipment and from applying currently available technologies to the typical new equipment to reach “max tech”.

Table 4-28: National Energy Savings Potential for Refrigerated Vending Machines

	Shipment-weighted Avg UEC (kWh/yr)	Total Electricity Consumption (TWh/yr)	Primary Energy Consumption (TBtu/yr)	Primary Energy Savings (TBtu/yr)
Fully-Cooled				
Typical Installed	2,743	1.4	14.2	n/a
Typical New [1]	2,494	1.2	12.9	1.3
Max Tech [2]	1,141	0.6	5.9	7.0
Zone-Cooled				
Typical Installed	2,483	8.2	85.8	n/a
Typical New [1]	2,258	7.5	78.0	7.8
Max Tech [2]	1,507	5.0	52.0	25.9
Total Savings:				**42**

[1]Typical New is assumed to use 10% less energy than typical installed.
[2] Max Tech shows 54% improvement over typical new for fully-cooled machines, 33% for zone-cooled machines.

4.8.2 Barriers to Implementation for Improved-Efficiency Refrigerated Vending Machines

1) Bottling companies or distributors purchase nearly all vending machines from the equipment manufacturers, but do not pay the utility bills associated with their operation creating a split-incentive system. The American Council for an Energy Efficient Economy described the problem of split-incentives to be the largest barrier to improved efficiency for beverage vending machines (Nadel 2002). Most bottling companies and distributors are concerned primarily with first costs and have little interest in energy efficiency because they do not pay the utility bills. Accordingly, manufacturers do not include efficiency measures in their designs if the measures increase initial cost, even if payback periods are short.
2) The desire to maximize the amount of product stored puts space in vending machines at a premium. Therefore, increases in insulation thickness are undesirable. External vending-machine dimensions are also constrained, by the need to fit through the doorways.
3) Rating standards such as the DOE efficiency standard are based on steady state operation, so technologies such as energy management systems that turn off lights or allow temperatures to float do not receive credit for energy savings using those test procedures. Similarly, variable speed compressors save energy at off-design conditions rather than at rating conditions, so manufacturers have limited incentive to implement these expensive technologies, even if they save substantial amounts of energy.

5 Advanced Energy Saving Technologies and Tools

This section introduces commercial refrigeration technologies that are not yet fully developed, but have significant energy savings potential. **Table 5-1** characterizes advanced technologies based on the following criteria: (1) energy savings potential, (2) technological risk, (3) time to commercialization. Energy savings potential represents the percent of system energy use that can be saved by implementing the given technology, assuming RD&D programs are successful in achieving target performance levels. Technical risk is classified as low for technologies that are well-understood and demonstrated, medium for technologies that have been credibly demonstrated with some aspects left to resolve, and high for technologies for which a number of unknowns still remain. Table 6-1 describes each technology, including energy savings potential, research needs, and barriers to implementation. In addition to technologies, other technical resources such as modeling tools and design guides are discussed in this section.

We did not conduct detailed economic analyses for advanced technologies because credible cost forecasts are generally not available.

Table 5-1: Advanced Technology Options and Tools for Commercial Refrigeration

Technology	System Energy Savings Potential %	Technical Risk (Low/ Med/ High)	Time to Commercialization (<5 , 5-10, >10 yrs)
Overall System			
Magnetic Refrigeration	20-30	High	>10
Thermoacoustic Refrigeration	10-20	High	>10
Thermoelectric Refrigeration	>20	High	>10
Ground Coupled Supermarket Refrigeration System	20-30	Medium	<5
Secondary Refrigeration Loops/CO_2 Cascade Systems	<10	Low	5-10
Compressors			
Linear Compressors	10-15	Medium	<5
Insulation			
Vacuum Insulation Panels and Aerogels	10-15	Medium	<5
Advanced Air-Curtains	5-10	Low	<5
Heat Exchangers			
EHD	5-10	Medium	5-10

Micro-Channel Heat Exchangers	<10	Low	<5
Refrigerants			
Natural Refrigerants	0-10	Medium	<5
Nanoparticle Additives	5-20	High	5-10
Lighting			
Solid State Plasma Lighting	20-30 %	High	5-10
Fiber Optic Lighting	20-30 %	Medium	<5
LEDs	15-20 %	Low	<5
Fan Blades			
Whale Fins	20%	Low	<5
Modeling, Monitoring, and Tools			
Modeling and Design Guides	10-20%	Low	<5
Monitoring	10-20%	Low	<5
Best Practice and Design Guides	10-20%	Low	<5
Sources: See text descriptions in sections 5.1 to 5.8			

5.1 Alternative Refrigeration System Technologies

Several alternatives to the conventional air-to-air vapor-compression refrigeration cycle are in various stages of development, including magnetic refrigeration, thermoacoustic refrigeration, thermoelectric refrigeration, ground-coupled refrigeration, and secondary loop configurations.

Magnetic Refrigeration

Magnetic refrigeration (MR) is based on the magnetocaloric effect (MCE). MCE is a magneto-thermodynamic phenomenon in which a reversible change in temperature of a paramagnetic material is caused by exposing it to a changing magnetic field (measured in Tesla, T). In the magnetic refrigeration cycle, randomly oriented magnetic spins in a paramagnetic material are aligned via a magnetic field, resulting in a rise in temperature. This heat is removed from the material to ambient by means of heat transfer. Upon removal of the magnetic field, the magnetic spins return to its randomized state thus cooling the material to below ambient temperature. The material is then used to absorb heat from the refrigerated volume thus cooling that space and returning the paramagnetic material to its original state and the cycle starts again (**Figure 5-1** and **Figure 5-2**).

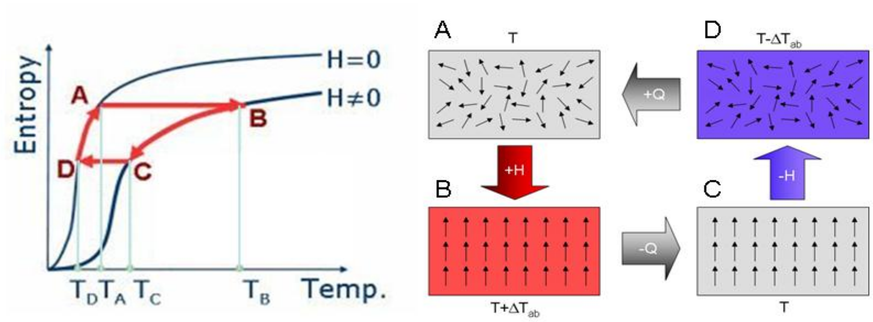


Figure 5-1: Thermo-magnetic Cycle.
(A-B) Randomly oriented magnetic spins align after applying a magnetic field (H). As the entropy is not reduced the temperature of the active material rises by Tab. (B-C) Excess heat is then removed as H remains constant. (C-D) When the magnetic field is turned off the spin moments re-randomize and the temperature is reduced by Tab due to the entropy reduction. (D-A) Heat from the refrigerated volume is absorbed by the active material raising its temperature and the cycle can begin again.

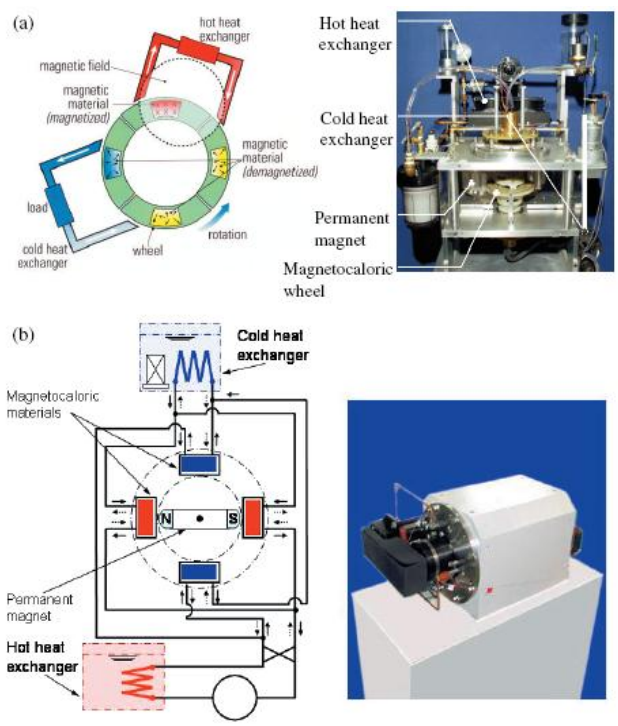


Figure 5-2: Rotary Magnetic Refrigerators
(*a*) The Astronautics Corporation of America rotary magnetic refrigeration and a schematic representation. A 1.4T magnetic field around the magnetocaloric wheel filled with Gd spheres is produced by a permanent magnet. The refrigerator operates near room temperature with a maximum temperature span of 20°C with a maximum cooling power of 95W and operates at a frequency between 1 and 4 Hz. The photograph is courtesy of Astronautics Corporation of America, Inc., Milwaukee, Wisconsin (Zimm 2003).
(*b*) The Chubu/Toshiba rotary magnetic refrigerator and a schematic representation. The 0.76T permanent magnet rotates inside of the four magnetocaloric beds, stopping momentarily to allow the appropriate fluid flows to occur before it moves to the next pair of beds. The beds contain Gd–Dy spheres of different Gd:Dy ratios. Using an

alcohol water solution as the heat transfer fluid a cooling power of 40W was obtained at a frequency of 0.28 Hz. The photograph and schematic is courtesy of Chubu Electric Power Co., Inc., Nagoya, Japan (Hirano 2003).

For room temperature applications, materials are needed that have a Curie temperature (the temperature above which ferromagnetic materials loss their permanent magnetism) around 295 K. Gadolinium and Gadolinium alloys have a large MCE around this temperature range and they are among the most widely used materials for room temperature refrigeration and space cooling applications. By using such materials and applying a 2 Tesla(T) magnetic field, researchers have demonstrated temperature differences of 9ºF (~5°C). Higher temperature differentials can be reached by increasing the magnetic field. For example, with a 10T field a temperature differential of 45ºF has been demonstrated. However, using such large magnetic fields is problematic as they require expensive and energy-hungry superconducting electromagnets such as the ones found in MRI machines. Using MR in practical applications will likely require a permanent magnet with magnetic fields between 1-2T to be economic and energy-efficient. The use of such small fields provides a challenge for applications such as air conditioning and refrigeration. This is because the small magnetic field is not enough to provide the required temperature lift. The primary obstacle is finding a refrigerant material that exhibits sufficient MCE in magnetic fields produced by permanent magnets (see Figure 5-3).

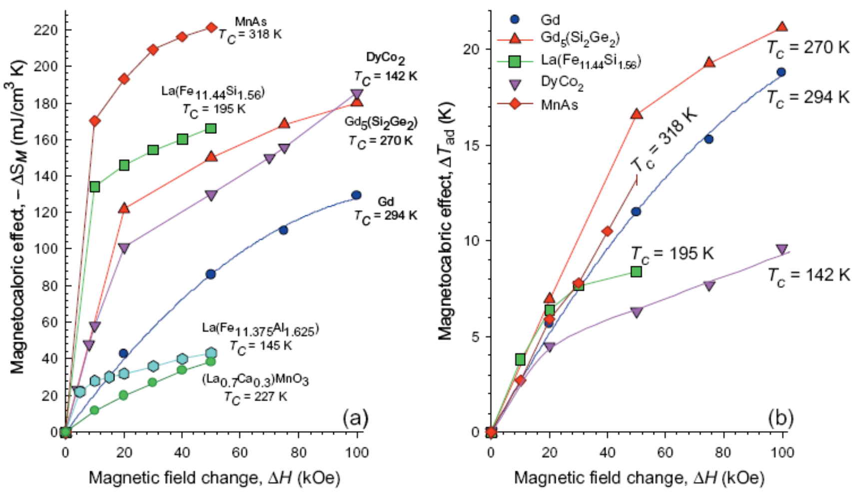


Figure 5-3: Entropy and Temperature Change vs. Magnetic Field
(a) Isothermal entropy change as a function of the magnetic field change for various materials.
(b) Adiabatic temperature rise as a function of the magnetic field change for different materials (Gschneidner 2005). (1Tesla = 10 kiloOersted or kOe)

There are four main hurdles that have to overcome before MCE materials could be practically used in commercial applications:

- Producing such materials at high volume and at acceptable cost.
- MCE materials are usually brittle and will not withstand a 15 year lifetime.
- Hysteresis effects.(although researchers have recently shown that this problem can be much reduced by adding iron to the alloy (Provenzano 2004)). The time delay for □T_{ad} to

reach its maximum value in the cycle. Solving this issue will mainly come through materials research.

Magnetic refrigeration has considerable energy savings potential. By coupling an active magnetic regenerator (AMR) with capacity modulation mechanisms, Seasonal Energy Efficiency Ratings (SEER) of 23 have been achieved (Dieckmann 2007). Analysis predicts efficiency gains of close to 20 percent when replacing a conventional vapor compression system with a magnetic system (Gschneidner 2008).

In 1997, the Ames Laboratory/Astronautics Corporation of America showed a MR with a cooling power of 600W with a 5T magnetic field and temperature lift of 18ºF. This refrigerator had a COP of 10 (60% of Carnot cycle efficiency) (Zimm 1998), showing the potential efficiency gains that can be achieved with MR. However, with larger temperature lifts (40ºF), both COP and cooling capacity were reduced to 2 and 150W, respectively. Other issues with this demonstration included a low operating frequency (0.16Hz)--for practical commercial refrigeration frequencies above 1*Hz* are needed. Since 1997 many demonstrations have been shown trying to address these issues (Figure 6-3), most using reciprocating or rotary designs with temperature lifts that rarely exceed 45ºF. Even with recent progress and increasing interest in MR, such low temperature lifts are not yet enough to compete with conventional systems.

The commercial viability of MR will depend on achieving efficiencies and costs similar to conventional vapor compression systems. The main hurdle is improving magnetocaloric materials by expanding their temperature lifts while operating at lower-intensity magnetic fields. However, cost reductions for the permanent magnets are also important. The field of MR has advanced rapidly in the past decade. However, without major leaps in performance, it is difficult to see how this technology can compete with conventional refrigeration systems in the next 5-10 years. Finally, it is worth noting that using MR posses another added value as it eliminates the use of conventional refrigerants with high global warming potentials (GWP).

Thermoacoustic Refrigeration

Thermoacoustic (TA) refrigeration is a technology that uses high-amplitude sound waves in a pressurized gas to pump heat. A device consisting of a series of small parallel channels, referred to as a “stack,” is fixed in place at a set location inside a resonator tube. In the case of a TA refrigerator, external work is supplied by the standing sound wave in the resonator. The longitudinal standing sound wave causes the gas particles to oscillate back and forth parallel to the walls of the stack. The alternating compression and rarefaction of the gas causes the local temperature of the gas to oscillate due to the adiabatic nature of sound waves. If the local temperature of the gas becomes higher than that of the nearby stack wall, heat is transferred from the gas to the stack wall. If the local temperature of the gas drops below that of the stack wall, heat is transferred from the wall to the gas. A schematic of the process is presented in **Figure 5-4**.

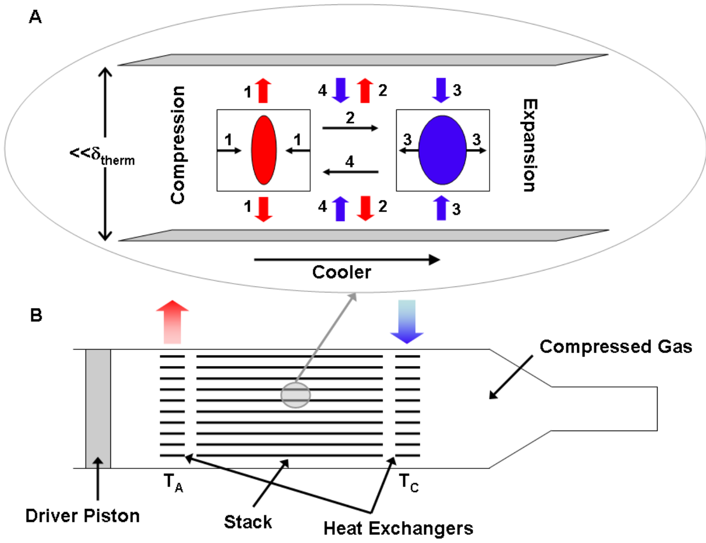


Figure 5-4: Operation of Thermoacoustic Refrigerators

(A) Operation of one pore in the TA refrigerator stack. In step 1 gas is compressed increasing pressure and excess heat is transferred to the nearby walls. In step 2 the gas moved towards the cooler side of the stack still transferring heat to the side walls. In step 3 the pressure is reduced causing the gas molecules to expand and absorb heat from the sidewalls. In step 4 the cooler gas moves back towards the high temperature side absorbing heat from the side wall. Finally, the cycle is repeated as the gas molecules are initiated to their starting phase in stage 1. It is worthy to note that heat transfer in steps 2 and 4 are equal and opposite, resulting in net heat transfer through steps 1 and 3 only. This leads to net heat transfer up the temperature gradient (Rossing 2007).
(B) Schematic of a TA refrigerator. Acoustic power is provided by the driver piston resulting in thermal power being absorbed and rejected from the system (indicated by red and blue arrows).

Prototypes of TA refrigerators have been demonstrated for refrigeration applications. A prototype TA refrigeration unit designed for an ice-cream freezer has been measured to be as efficient as the vapor compression unit it replaces (COP = 0.81 or 19% relative to the Carnot COP) (PSU 2009). Researchers expect TA technology to improve in efficiency with further development of heat exchangers and other subsystems. It is also likely that efficiency in many applications could improve because TA refrigerators are well suited to proportional control, where the cooling capacity can be continuously controlled so that the output can be adjusted to match varying load conditions. TA refrigeration also has the environmental benefit of being refrigerant-free.

TA refrigeration is still in the early design stages. Issues include a lack of suppliers that produce vital parts such as inexpensive loudspeakers or heat exchangers that are optimized for high-frequency oscillatory flow of compressed gases (PSU 2009); however this is a common issue for many early stage technologies. TA freezers have been demonstrated in some applications. Funded by substantial investments from Ben and Jerry's Ice Cream, engineers at Pennsylvania State University are building TA freezers, based on a proven prototype, to replace conventional ice cream freezers in the company's stores (Newman 2006). Due to high costs and other aforementioned issues, it is difficult to see a clear pathway where TA refrigeration can compete with conventional refrigeration systems in the next 5-10 years.

Thermoelectric Refrigeration

Thermoelectric (TE) refrigeration is similar to magnetic refrigeration. However, instead of using a magnetic field to induce temperature changes they use electrical current. TE devices are solid state, semiconducting systems that directly convert electrical power to thermal energy for either cooling or heating. These systems also have potential to reduce CO_2 and greenhouse gas emissions as they do not use fluorocarbon refrigerants. TE systems are simple compared to vapor-compression systems that need to compress and expand working fluids like fluorocarbon refrigerants to operate; they instead use electrons as their “working fluid” (Bell 2008). However, TE systems have not been used broadly thus far due to their lower efficiency and higher cost compared to vapor compression. On exception is portable beverage coolers that can be plugged into automotive lighters for power. These are convenient for providing a small amount of cooling when ac power is not available, but the cooling capacity required is very small and the efficiency is not a major concern due to the low overall cooling requirements.

TE effects arise due to charge carrier’s freedom to move like gas molecules that can carry heat as well. In the presence of a temperature gradient, carriers at the hot end tend to migrate towards the cold end. The charge build up results in an electrical potential (voltage) between the hot and cold ends, known as the Seebeck effect (V=□□T) (Seebeck coefficient, □). When operated in reverse, the Seebeck effect can be used as a cooling system. Applying a DC voltage to the system can drive an electrical current (I) and heat flow (Q), thereby cooling one surface due to the Peltier effect (Q=□TI).

The efficiency of a thermoelectric material for cooling and heating applications is determined by its figure of merit, *ZT* (see details below). *ZT* describes the efficiency of the n-type and p-type materials that make up the TE couple. Currently the best TE systems have a *ZT* value around 1 (COP=0.66). However, as ZT scales nonlinearly with efficiency, values around 9.2 need to be achieved to match the energy efficiency of current conventional systems (Bell 2008).

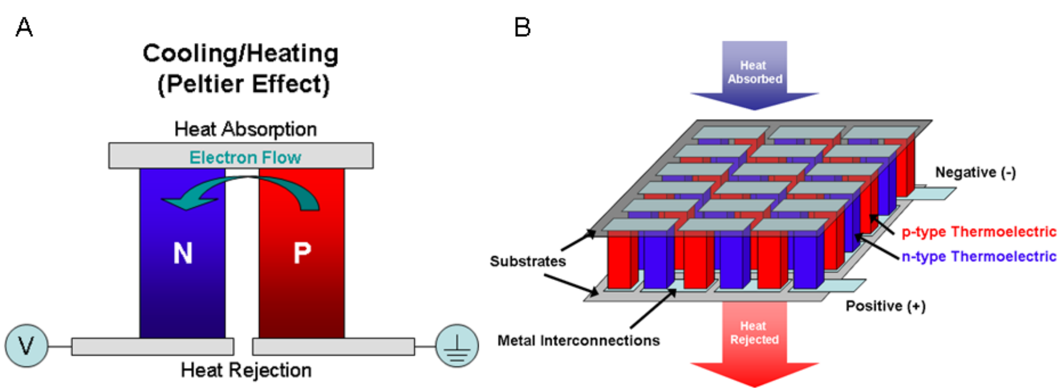


Figure 5-5. Thermoelectric heat engines.
(A) When current is applied across the TE device depending on its direction heating or cooling occurs via the Peltier effect. (B) TE module showing device architecture of a practical TE generator. In this case multiple TE junctions are connected together in series.

In the mid 1990's, Hick and Dresselhouse published theoretical predictions showing possibilities of making substantial TE efficiency gains using nanostructural engineering. These efforts led to experimental demonstration of proof-of principle and high-efficiency materials (Dresselhaus 2007, Chen 2003).

Highly efficient TE materials are very difficult to develop as they have a conflicting combination of physical properties. To maximize the *ZT* value of a material high thermo-power (Seebeck coefficient, □) and high electrical conductivity are needed, but also with low thermal conductivity. High Seebeck coefficients are present in insulators and semiconductors but such materials also have low electrical conductivity. Finding a combination of these properties in a single material remains a challenge.

Additional material design conflicts stem from the necessity for low thermal conductivity. Glasses exhibit some of the lowest lattice thermal conductivities but they also have very low charge carrier mobility, making them poor TE materials. On the other hand, good TE materials require a 'phonon-glass electron-crystal' material, which is a crystalline material that scatters phonons (thermal lattice vibrations) with minimal disruption to its electrical conductivity. The crystalline requirement results from the desire for high conductivity, whereas the phonon-glass requirement results from the need for low lattice thermal conductivity (Snyder 2008).

Different TE materials are used to match the operating temperature range of the application. For an application that rejects heat at room-temperature, the most common materials used are alloys of Bi_2Te_3 (p-type) and Sb_2Te_3 (n-type) due to their relatively high *ZT* values of approximately 1. By changing the alloy composition it is possible to adjust or tune the carrier concentration allowing *ZT* to be optimized to peak at different temperatures (Figure 5-6) (Snyder 2008). For cooling applications below RT, alloys of BiSb have been used in n-type legs coupled with legs of $(Bi,Sb)_2(Te,Se)_3$. However the poor mechanical properties of BiSb leave much room for improvement.

Despite significant improvements in recent years that increased ZT significantly on the nanoscale, scaling up to practical applications has proven to very difficult. However, TE devices are mass produced for cooling applications in several niche markets where size and convenience make them attractive. Current applications include portable coolers which are quiet and vibration free. They are also widely used as replacements for wine cabinets, mini-refrigerators, and water coolers.

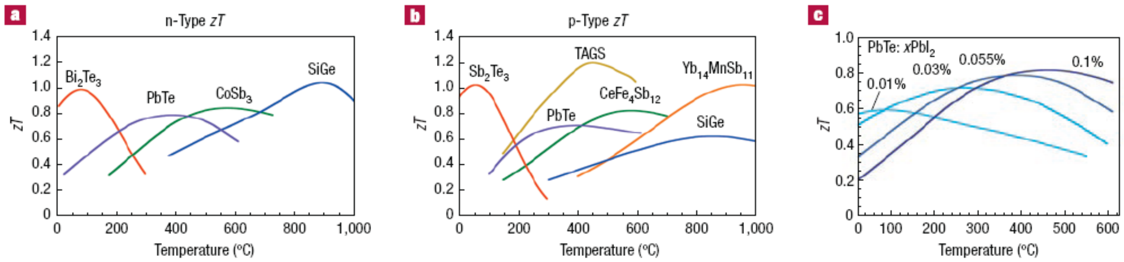


Figure 5-6: *ZT* of State-of-the-Art Materials

Includes materials used or being developed commercially or by NASA for thermoelectric power generation. (a), p-type and (b), n-type. Most of these materials are complex alloys with dopants. (c) Altering the doping concentration changes not only the peak *ZT* but also the temperature where the peak occurs. As the dopant concentration in n-type PbTe increases (darker blue lines indicate higher doping) the *ZT* peak increases in temperature (Snyder 2008).

Researchers claim that, once fully developed, thermoelectric refrigeration cycles could be 50 percent more efficient than current vapor compression ones. With refrigeration equipment accounting for a large percentage of national energy consumption, this means dramatic energy savings potential. Like magnetic refrigeration, electrocaloric cycles do not require a compressor, are more inclined to efficiency boosting controls, and could potentially operate without using refrigerants with high GWP (DOE 2000).

Currently, scientists have only been able to electrically induce 12°C temperature ranges in certain ceramics at room-temperature (Neese 2008). This temperature difference is not sufficient for room-temperature refrigeration applications, which require temperature differences around 40°C. Finding an economical material that is capable of producing the necessary temperature gradient is the primary barrier of implementation. Once overcome, thermoelectric refrigeration could be a practical, environmentally friendly replacement for vapor compression refrigeration (Patel-Predd 2008).

In summary, for TE refrigeration to become competitive with conventional vapor compression technology, some major advances must occur. They include increasing the hot and cold temperature differences ranges from around 12ºC to 40ºC as well as reducing the operating temperature. To achieve energy efficiencies that can compete with conventional refrigerators TE device efficiency needs to improve by a factor of four which translates to increasing *ZT* values from 1 to 9.2. Such advances are directly correlated to improvements in TE materials. While the field has been advancing rapidly in the past few years, without major leaps in performance it is difficult to see how TE technology can compete with conventional refrigeration systems in the next 5-10 years.

Ground-Coupled Supermarket Refrigeration Systems

Similar to ground-source (or "geothermal") heat pumps, a ground-coupled supermarket refrigeration system would reject heat to the ground rather than to ambient air. The advantage of this configuration is that the ground provides a stable and low temperature heat sink, unlike ambient air. Cooling water would be used to cool the condenser and it would then be circulated into ground water loops similar to those used for ground source heat pumps. This configuration avoids the maintenance, cost, and water treatment and water usage disadvantages to evaporative cooling of condensers. Bore holes could be located anywhere, including under the supermarket parking lots. Based on typical savings achieved with ground source heat pumps for comfort conditioning, it would be realistic to expect 20-30% savings relative to conventional supermarket rack systems.

Secondary Loop

Current supermarket refrigeration systems use about 1.8 million kWh of electricity per year for a store with 33,000 ft^2 selling area (EPA 2007). Furthermore, in large stores the amount of piping and fittings used in refrigeration systems can result in significant refrigerant leakage. Some systems lose as much as half of their refrigerant charge (3,000 - 5,000 lb) per year (ORNL 1997); more commonly the loss is around 15-25% per year. Lost refrigerant not only increases operating costs, but contributes to global warming.

Secondary loops place an additional heat-transfer loop between the refrigerant and the load (see **Figure 5-7**). In refrigeration applications, propylene glycol is typically used as the heat-transfer fluid used in the secondary loop. Propylene glycol is a non-toxic, non-flammable, liquid that has minor global warming impact.

Use of secondary loops in supermarket refrigeration systems provides several advantages:

- Requires about one-tenth the refrigerant to operate (less than 500 lb) compared to conventional systems
- Dramatically lowers refrigerant leakage
- Dramatically lowers refrigerant pressure drops and thermal losses in interconnecting piping
- Helps ensure good oil return to the compressors.

The key disadvantages of secondary loops in supermarket applications include:

- Adding an additional heat-transfer loop with heat exchangers tends to increase the overall temperature lift required, which tends to lower efficiency
- Requires additional pumps, adding system cost and parasitic energy consumption.

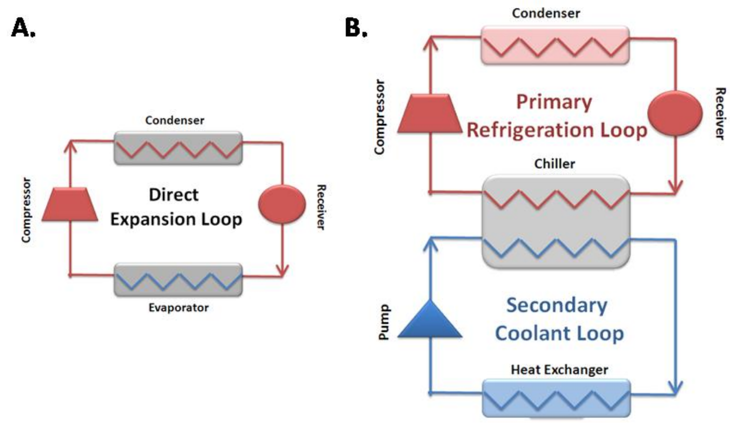


Figure 5-7: Secondary Loop used with Vapor-Compression Refrigeration. (A) Diagram of a direct expansion loop. (B) Diagram of a secondary refrigeration loop

Supermarket refrigeration systems with secondary loops have been demonstrated in the field. A study by Southern California Edison in 2004 describes a field test comparison between a secondary loop system and a conventional supermarket refrigeration system with evaporative condensing (CEC 2004). Although the field test showed modest savings for the secondary loop system, the source of these savings is unclear, and the comparison may not be a fair “apples-to-apples” comparison. In general, due to the extra heat exchange process, all other things being equal, secondary loop systems are not expected to show substantial energy savings advantages over a conventional system. (Emerson 2009)

The major advantage of secondary loop systems is the reduction in refrigerant charge (by up to a factor of 10), which results in far lower refrigerant leakage. Consequently, secondary loop systems have substantially lowers greenhouse gas emissions and can dramatically improve maintainability, thus reducing operation and maintenance costs. However, the installed cost of secondary loop refrigeration systems is higher than that of conventional systems, and secondary loop systems are unfamiliar to many contractors and supermarkets. Consequently, supermarkets are wary of adopting what they may perceive as a risky technology, but as the importance of direct greenhouse emissions from fluorocarbon refrigerants increases, secondary loop systems will likely receive greater attention.

5.2 Advanced Compressors

Linear Compressors

Linear compressors employ a different design than either reciprocating or rotary compressors and are reportedly more efficient than either. A diagram showing the differences between a linear and reciprocating design is presented in **Figure 5-8**. The first version of the design was developed by Sunpower for integration into refrigerators for the European market using isobutene (R-600a) as a refrigerant (Sunpower 1999).

The linear compressor has an axi-symmetric configuration. The compressor is comprised of a piston reciprocating in a cylinder and coupled directly to a linear motor. Since there is no conversion of rotating motion, all the forces of the linear compressor act along a single axis, the axis of the piston motion. This operation along a single axis together with direct coupling between the motor and piston generates minimal side loads allowing the use of a gas bearings system that prevents contact between the piston and cylinder. If oil lubrication is the method of choice, then the oil can be of a low viscosity, to minimize friction losses.

Linear Compressor Reciprocating Compressor

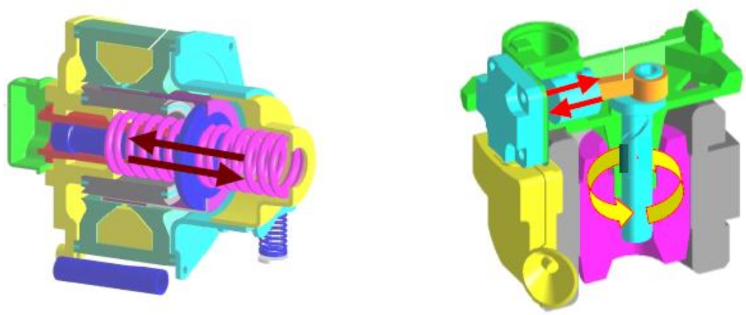


Figure 5-8: Linear and reciprocating compressor design diagrams.
Source: Increasing Appliance Energy Savings by Looking Beyond the Current Energy Star, Steven Nadel, ACEEE, 2004 Energy Star Appliance Partner Meeting

Recently a major Asian appliance manufacturer has developed an energy-efficient ‘free-piston’ linear compressor for a household refrigerator. While this product is targeted at the home refrigerator market, it could be equally applicable to small commercial refrigeration applications such as reach-in refrigerators, vending machines, etc. This refrigerator achieves its efficiency gains by employing a design with the following characteristics: (1) a highly efficient linear motor due to the elimination of end coil losses; (2) no crankshaft mechanism, which reduces friction and side-force losses; (3) direct suction and a straight flow path resulting in low heat exchange and reduced flow losses; and (4) a ‘free-piston’ system which allows for variable cooling capacity. The company claims that its line of linear compressors is up to 20 percent more efficient than reciprocating designs (LG 2004). In addition noise levels can also be reduced by utilizing linear compressors (FRPERC 2003). However, it should be noted that exact efficiency improvements of the linear compressors are hard to verify as the manufacturer has not used standard ASHRAE conditions. ASHRAE conditions for evaporating and condensing temperatures are -10ºF (-23.3ºC) and 130ºF (54.4ºC), respectively, while the manufacturers ratings are based on evaporating and condensing temperatures of -14.8ºF (-26ºC) and 100.4ºF (38ºC), respectively. In the trade press, the manufacturer has expressed willingness to license the linear compressor technology to competitors (ACEEE 2004). However, because the design is proprietary, the widespread use of linear compressors is highly uncertain.

5.3 Advanced Insulation

Vacuum Insulation Panels

Vacuum insulation panels (VIPs) exploit the virtually zero thermal conductivity of evacuated space to provide insulation. Multiple thermal-radiation barriers are typically included, along with a low-conductivity core material to provide structural integrity. Conventional insulation panels achieve R-values around 8 hr-sqft-˚F/Btu or less per inch, while commercially available VIP claim R-values of 30 hr-sqft-˚F/Btu per inch or greater (Glacier Bay 2009a). VIPs could, therefore, significantly reduce refrigeration loads without increasing insulation thickness—perhaps while even reducing insulation thickness.. VIPs are particularly attractive in applications like self-contained refrigerators (e.g. reach-ins, beverage merchandisers) where increasing the refrigerator outer dimensions or reducing refrigerated volume is unacceptable.

The energy savings potential of VIPs depends greatly on the application. In a Class A beverage vending machine, use of VIP at an R-value of 30 would decrease energy consumption by 10 percent, according to NCI simulations.

To date, lifespan concerns, inconsistent results from prototype testing, and high costs have prevented widespread use of VIPs. The core materials of VIPs provide structural support and also limit the mean free path of the gas molecules remaining in the panel. Typical core materials include mineral powder, mineral fiber, fiberglass and silica, all of which are costly after being processed for VIP applications (Glacier Bay 2009b).

Results of prototype tests have been inconsistent, partly due to the impacts of edge losses. The films encasing the core materials of VIPs often include metals to provide impermeability, but this can lead to high levels of conduction and consequent thermal losses around the edges of panels,. Further development could address some of the VIP performance issues and reduce their price to make them cost competitive with conventional insulation materials.

One possible approach to reducing the cost premium of VIPs would be to explore “active” vacuum panel insulation, particular for closed-door supermarket display cases. One of the factors that makes VIPs complex and expensive to manufacture is the difficulty of ensuring that vacuum is maintained for many years. However, it is conceivable that “active” vacuum panels could be developed that use small, built-in, vacuum pumps to maintain vacuum, which would enable far simpler and cheaper manufacturing techniques. The additional cost of the vacuum pump and associated hardware might more than offset the cost reduction associated with manufacturing simpler panels that do not need to maintain a high vacuum over many years. The parasitic electric consumption of the vacuum pump should be modest, assuming that leakage paths are minimal.

Advanced Air Curtains

Open refrigerated display cases are attractive to retailers because they allow consumers easy access to products. As described in section 3.1.1, open display cases generally incorporate circulated air curtains to help keep cold air inside the case. They effectively create invisible barrier between the cold air inside the display case and the ambient temperature air outside. Air curtains are planar jets of air with a large aspect ratio that have more momentum than the surrounding air molecules. They are mainly responsible for separating the cooling zone in the display case from the ambient air outside the display case (see **Figure 5-9**). However, other than temperature separation air curtains have other benefits, they also act as barriers for air-born particles and help maintain the relative humidity inside the display case.

Improved air-curtain design is aimed at reducing the impact of infiltration by reducing the entrainment of warm ambient air. While mixing between the cold air curtain and warm store air cannot be avoided as part of the cold air spills over the display case and is replaced by the outside air, making the air curtain flow as laminar as possible reduces entrainment (see **Figure 5-9**). This can be achieved by configuring the air profile before the air curtain discharge air grille (DAG) using a honeycomb grille to align the airstreams. This in turn encourages laminar flow of air and improves performance by reducing the infiltration load (DOE 2009a).

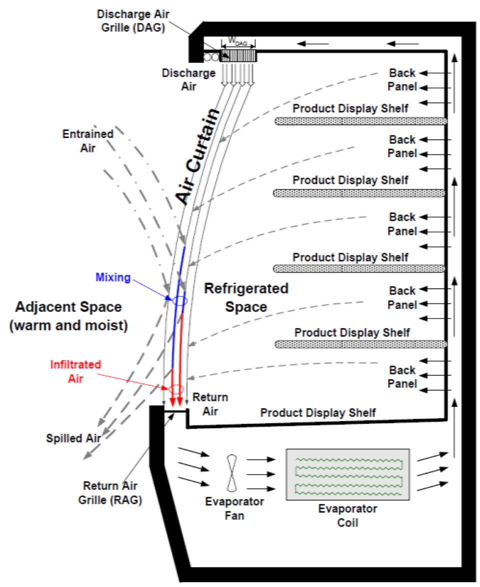


Figure 5-9: Schematics of a typical open refrigerated vertical display case and air circulation pattern (side view). Image taken from DOE 2009a.

The amount of warm air that moves into the display case through the DAG, the infiltration rate, is responsible for most of the cooling load and thereby the power consumption. Since roughly 80 percent of the refrigeration load in open refrigerators is caused by warm air infiltration, air curtain performance becomes a problem of interest. It is possible to control the infiltration rate by changing various parameters of the air curtain such as velocity and throw angle (DOE 2009a, ORNL 2008)). For example, in a report from ORNL they show a reduction of 8% in the infiltration rate by changing the velocity and throw angle of the air curtain (ORNL 2008). This reduction in infiltration rate in turn impacts the compressor load and results in a 4% daily power reduction. Additional savings of 9%, or a total of 13% savings, were observed by optimizing the suction pressure while maintaining product temperatures below 41°F. Simulations by ORNL show that turbulences in the return air grille (RAG) are also an important factor in reducing infiltration. By reducing turbulence in the RAG further improvements can be made and the infiltration rate can be reduced by an additional 8% from the current value of 14% to around 2% (ORNL 2008).

Overall, ORNL shows that a very limited and simple geometrical and flow alteration can reduce the infiltration rate by 8% and reduce compressor energy consumption by 4%. Simulations and experiments suggest there is more room for improvement.

5.4 Heat Exchangers

Micro-Channel Heat Exchangers

Micro-channel heat exchangers transfer heat through multiple flat, fluid-filled tubes containing small channels (10-1000μm) while air travels perpendicular to the fluid flow. Compared with current fin-tube heat exchangers, the air passing over the heat exchanger has a longer dwell time, thus increasing both the efficiency and the rate of heat transfer.
By constraining the flow to such narrow channels, thermal diffusion lengths are short, and the characteristic heat-transfer coefficients are very high. Since the thermal performance is so strong, relatively short flow passages are required, and with many flow passages in parallel in a small device, the pressure drop can be small as well (PNL 2008). This increase in heat exchanger effectiveness allows the micro-channel heat exchanger to be of smaller dimensions and maintain similar performance compared to a regular heat exchanger. It is possible to get improved performance for the same volume as a conventional heat exchanger. Micro-channel technology is very common for automotive air conditioning application due to its small size and reduced weight. They have also been introduced in some stationary a/c applications due to compact size considerations. This is an indication that the technology can overcome the critical manufacturing huddles.

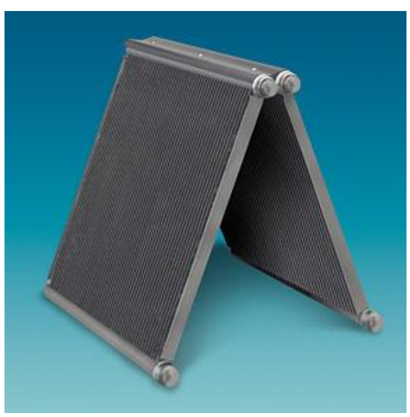


Figure 5-10: Micro-channel heat exchanger. Image taken from Delphi.

Studies show that systems using micro-channel heat exchangers show about 15% improvement in heat exchanger efficiency. This translates to a 5% improvement in energy saving potential compared conventional heat exchangers (ACEEE2004, International Journal of Refrigeration 2008b).

Electro-Hydrodyamically Enhanced Heat Transfer (EHD)

Heat transfer techniques, including electro-hydrodynamically (EHD) enhanced heat transfer, can improve the heat transfer duty of heat exchangers. These techniques can be divided into two main groups, active and passive. Active techniques require the use of external forces to the heat transfer surface, such as surface vibration, acoustic or electric fields, passive techniques are based on application of specific surface geometries with surface augmentation. The effectiveness of both techniques strongly depends on the mode of heat transfer, single phase or multiphase. Besides the improvement of the heat exchanger performance for the same area, heat transfer enhancement enables a decrease in the physical size of a heat exchanger while maintaining its performance.

Electro-hydrodynamic (EHD) enhancement of heat transfers refers to the coupling of an electric field with the fluid field in a dielectric fluid medium. In this technique, either a DC or an AC high-voltage low-current electric field is applied in the dielectric field medium flowing between a charged and a receiving (grounded) electrode. The applied electric field destabilizes the thermal boundary layer, thereby producing better mixing of the bulk fluid flow and, thereby, increasing the net heat-transfer coefficient. EHD appears to be more effective when applied to phase-change processes (e.g., boiling and condensation).

EHD is typically used to electronically control the capacity of a heat exchanger by adjusting the applied voltage to adjust the heat transfer. Therefore, the heat exchanger can be sized to operate conventionally during non-peak periods and use EHD during peak-load periods. EHD can replace, or work in conjunction with, enhanced surface heat exchangers.

To use EHD, an electrical voltage (from a few volts to thousands of volts) is applied to the heat exchanger. However, because heat-transfer fluids are typically dielectrics (of low electrical conductivity), even high voltages produce very little current. This low current helps keep the power (voltage x current), and the associated energy penalty, low.

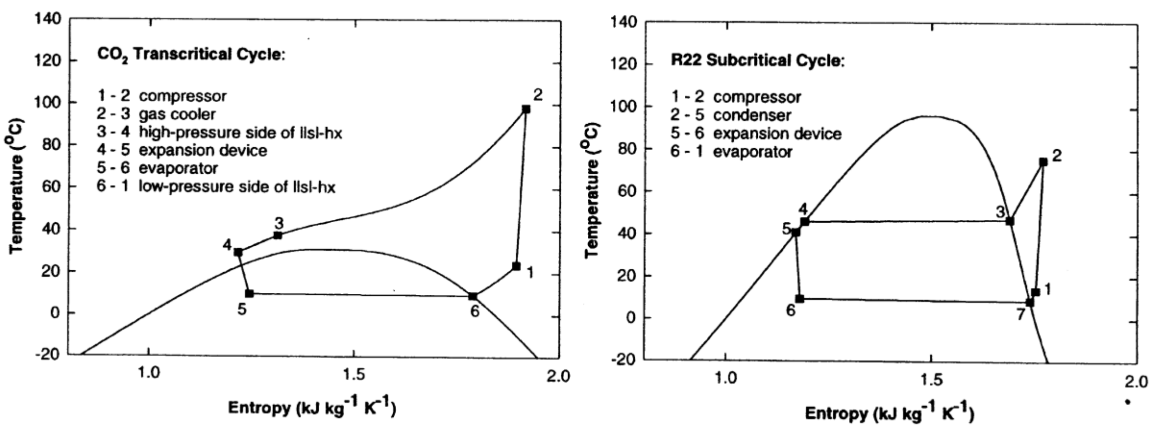
By using the EHD-enhanced heat transfer it is possible to calculate the COP of the system. Using experimental results of the EHD-enhanced heat transfer Al-Dadah et al. calculated the increase in COP for an R-22 vapor compression refrigerator. Their model showed an increase of 8% in the system COP (Al-Dadah, 1992). Eames and Sabir showed in their calculation that if enhancement is applied simultaneously to the condenser and evaporator significant increase in COP can be achieved. For this calculation they used material properties described in the literature to calculate a COP increase on the order of 40% (Eames 1997).

5.5 Alternative Refrigerants

Carbon Dioxide

CO_2 (R-744) was widely used as refrigerant in the early 20th century; however, during the middle of the 20th century, it was replaced by the fluorocarbons. Over the last several decades, chemical companies have introduced new refrigerants in response to government mandates for ozone friendly, chlorine free, refrigerants. When introduced, many expected hydrofluorocarbon (HFC) refrigerants to provide a permanent solution, but due to their global warming impact, future use of these refrigerants may be restricted. Use of natural refrigerants, such as CO_2, that are chlorine free and have low global-warming potential are gaining increased attention (Kyoto Protocol 1997). CO_2 refrigerant has a favorable global-warming potential (GWP) (GWP of 1, compared to 1300 for HFC-134a). As an additional benefit, there is no need to capture the CO_2 during maintenance, thereby simplifying handling and providing cost savings as well.

CO_2 is both non-toxic and non-flammable; furthermore, it is abundant and inexpensive. CO_2 has an operating pressure 3-10 times higher than for conventional refrigerants (7.38 MPa). CO_2 has a low critical temperature (31.1°C) thus requiring a transcritical refrigeration cycle. In a transcritical cycle, the condenser is replaced by a gas cooler. The refrigerant evaporates in the subcritical region and rejects heat at temperatures above the critical point in the gas cooler (**Figure 5-11**). The high operating pressures of CO_2 systems present both a challenge and an opportunity, as system size and weight can potentially be reduced.

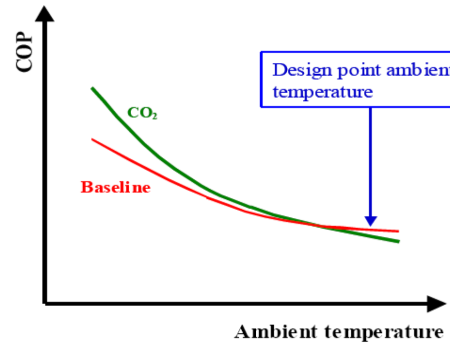


Source: Brown 2002

Figure 5-11: CO_2 transcritical cycle and R-22 subcritical cycle at an ambient temperature of 35C (95F) with baseline UA values.

Currently, CO_2 is drawing most interest in systems with high refrigerant leakage rates that are significant enough to attract regulatory attention (such as automotive air conditioning and supermarket rack systems) as well as in high-temperature heat-pump applications and selected military cooling applications where it is attractive for logistical reasons.

However, the use of CO_2 has its drawbacks. The thermodynamic efficiency of CO_2 cycles is lower than for conventional systems, especially at high ambient temperatures (**Figure 5-12**). This efficiency reduction can counter the environmental benefits of CO_2 systems, and may not be acceptable from both a regulatory and a marketing standpoint. Therefore, efficiency must be improved before CO_2 systems can be widely adopted (Topping 2004).



Source: Nekså

Figure 5-12: Principal COP behavior CO_2 system and conventional (baseline) systems at varying ambient temperatures.

The use of new, economical screw and scroll compressors could significantly improve the efficiencies of CO_2 systems working near, or above, the critical temperature. Currently available screw compressors are capable of providing up to 100 bar (gauge), but these compressors are not designed for refrigeration use. However, market demand for CO_2 compressors could possibly result in development of high-pressure screw compressors for refrigeration applications.

Carbon dioxide is likely to be more widely used as the low-temperature stage of cascade systems and, when economical CO_2 compressors become available, as a single-stage system for air conditioning and refrigeration. The use of CO_2 may present significant benefits for air conditioning and refrigeration applications. Benefits include better heat transfer, much smaller piping, reduced pumping power, and the elimination of chilled water (Pearson). Currently, CO_2 systems have a higher cost than conventional systems, but, with better designs that require less raw materials and manufacturing learning curves, can lead to cost reductions. If CO_2 becomes more widely adopted, costs could rival conventional system costs within several years (Antonijevic 2008).

Nanoparticle Refrigerant Additives

Nanomaterials are materials with dimensions on the order of nanometers. These materials include nanoparticles, quantum dots, and carbon nanotubes. Nanomaterials are being investigated for numerous applications due to their superior electrical, optical, mechanical, and chemical performance, among many other favorable properties. Researchers at NIST have found that the use of nanoparticles, especially ones made of copper oxide, have a beneficial effects when they are mixed in a common polyester lubricant and added to a refrigerant (HFC-134a). NIST observed a heat-transfer improvement of between 50 and 275% (Kedzierski 2007).

However, the mechanism behind the improved performance is not well understood yet, and performance benefits are sensitive to material selection and concentration

A partial explanation for how nanoparticles improve heat transfer is that highly thermally conductive nanoparticles increase the overall thermal conductivity of the fluid. This, however, accounts for only a about 20% of the total increase in system performance (Kedzierski 2007). One theory holds that, when present in sufficient concentrations, highly thermally conductive nanoparticles further enhance heat transfer by encouraging more vigorous boiling. The nanoparticles may serve as nucleation sites for boiling translating to improved heat transfer in the system.

Such improvements to refrigerant heat transfer can be beneficial for drop-in replacements as well as in new system designs. Furthermore, as more advanced nanoparticles are developed, they could be substituted for existing nanoparticles by simply flushing the system and replacing the refrigerant. Due to the improved energy efficiency, new system designs might allow downsizing of heat exchangers, with the associated cost benefits. Of course, this has to be balanced against the efficiency impacts of smaller heat exchangers.

The long-term health effects of nanomaterials are not yet fully understood. Addressing potential health issues and assuring the safety of nanomaterials is essential if they are to be widely used.

5.6 Advanced Lighting

Lighting efficiency improvements lower energy use in two ways--1) direct reduction of lighting electricity use; and 2) indirect reduction of refrigeration-system energy use due to less heat generation in refrigerated cabinets .

Light-Emitting Diodes (LEDs)

Light-Emitting Diodes (LED) provide an opportunity for increasing the efficiency of refrigeration lighting systems. Although current LEDs not as luminescent as fluorescent fixtures, they are rapidly closing the gap. Laboratory demonstrations show that white LEDs have already surpassed efficiencies of fluorescent bulbs. In some applications, the directional nature of LED-generated light provides additional benefits, providing comparable illumination with lower lumen output. For refrigeration application this is advantageous because the use of directional light can better illuminate merchandize in the display case. In research conducted by the Lighting Research Center (Raghavan 2002), subjects rated lighting display cases using LEDs to be more desirable than conventionally lit cases. Another benefit for using LEDs load reduction on the display case due to reduced heating. LEDs do not heat up as much as conventional fluorescent lights do.

LEDs are predicted to steadily increase in efficiency and decrease in cost as technology improves and production volumes grow. Although LED’s have started to penetrate the commercial refrigeration market, more research could reduce the price premium and foster development of application-specific lighting system designs that leverage the benefits of LEDs.

Solid-State Plasma Lighting

Plasma bulbs emit light when their contents (gas and metal halide) are vaporized to a plasma state. Conventional high-intensity discharge (HID) lamps, including Metal Halide (MH) lamps, generate plasmas using electrodes that penetrate the bulb casing. Solid-state plasma lighting eliminates these electrodes, using instead a focused electric field to heat the gas and metal halide mixture without penetrating the lamp. A focused radio frequency signal generated by a solid-state power amplifier produces the electric field. At 140 lumens/watt, plasma bulbs have demonstrated efficaciousness substantially higher than LEDs and they last just as long. In addition, plasma bulbs can handle high power inputs, allowing them to emit up to 30,000 lumens. Plasma bulbs are currently being marketed for use in street lamps, but a decrease in costs could make them attractive for a wide range of other applications, including refrigeration (LIFI 2009).

Fiber-Optic Lighting

Fiber-optic lighting systems use fiber-optic cables to distribute light from a remote source to the lighted area. A fiber-optic cable is made of glass or plastic, and uses the principal of internal reflection to guide light along its length. For refrigeration applications, fiber-optic lighting systems could save energy in two ways:

- The fiber-optic cables can direct light from a single source to multiple areas, reducing the overall number of lights needed for a given application
- The light source can be remote from the refrigerated space, significantly reducing the heat load placed on the refrigeration system.

In a pilot program by Southern California Edison, researchers retrofitted vertical reach-in freezers of a commissary with fiber-optic lighting (SCE 2006). Compared to the fluorescent system it replaced, the fiber-optic system reduced lighting energy consumption by 50% and compressor energy consumption by 17%. The combined energy savings reduced energy consumption by 25% for the entire system. The fiber-optic lighting delivered 60% less luminescence compared to a fluorescent system; however, researchers reported no perceptible difference in the lighting of the product.

The installed cost for the fiber-optic system was $30,000, with a payback period of 13.3 years. While this is significantly more expensive than either fluorescent or LED lighting systems, with economies of scale, price reductions can be expected. Test results reported by the Southern California Edison’s Refrigeration Thermal Test Center (RTTC) , show that, in a 3-door low-temperature display case, LED and fiber-optic systems drew, respectively, 25% and 64% less power than fluorescent lighting. For LEDs, the measured cooling-load reduction on the display case was small (125 Btu/hr), but significantly greater for fiber optics (574 Btu/hr) (SCE 2009). Materials research to make fiber-optic cables cheaper could make fiber-optic systems cost competitive.

5.7 Fan Blades

Whale Fins

Mimicking the tubercles found on humpback whale fins, scientists have developed fan blades lined with similar bumps. Although they are being developed primarily for use on wind turbines, whale-fin fan blades show a 20-percent increase in efficiency for industrial-sized ceiling fans (Hamilton 2008). This technology is expected to be commercially available for industrial use in less than a year. We found no evidence of ongoing research to adapt this technology for use in commercial refrigeration applications, but it appears theoretically plausible.

5.8 Modeling, Monitoring and Tools

A survey of commercial refrigeration industry stakeholders, including researchers, manufacturers, and store owners, revealed a strong interest in increasing the quality and availability of information related to the design, operation and improvement of commercial refrigeration equipment.

Modeling and Design

New and improved design tools/modeling methods would be beneficial to the commercial refrigeration industry. Performance and energy-efficiency enhancement of refrigeration-cycle components and systems are limited unless cutting-edge design tools, based on updated, more accurate models, are introduced. An arsenal of reliable, widely used design tools based on up-to-date science will also contribute to continuity, preventing knowledge loss as engineers and technicians retire.

In essence, design tools help turn energy savings potential into realized energy savings. Design tools are an effective way of packaging complex models. Usually based on higher order scientific principles, these models would be difficult for the lay person to apply. In the form of a design tool, however, these models become streamlined and user friendly. Design tools are also highly customizable. They can be made to optimize the design of a single component or an entire system. As a result, commercial refrigeration component and system designs become more informed. This improves the effectiveness and efficiency of the design process. Improvements to the design process make opportunities for increased energy efficiency more readily available and attainable.

Monitoring

By integrating a network of sensors and data-collecting mechanisms, users of commercial refrigeration equipment could gain access to real-time information detailing the energy use and performance of their equipment. Monitoring systems facilitate and improve equipment owners' ability to operate and maintain large arrays of refrigeration equipment more efficiently. Conventionally, equipment operators rely on whole-facility monthly energy bills to monitor performance and consumption. Anomalies in this information are hard to translate because so many systems outside of refrigeration also contribute to the monthly bill. With real-time data from a monitoring system, equipment operators would be able to recognize, pinpoint and diagnose unexpected fluctuations in performance or usage very quickly. .

Because performance and usage issues can be detected and diagnosed in a timely manner, monitoring systems provide significant energy savings potential. Monitoring systems have already been implemented on a small scale. In a popular American supermarket a leakage detection and energy usage monitoring system reduced electricity use by 23 million kWh per year, while avoiding emissions due to electricity generation of nearly 17,500 tons of CO_2, as well as 71 tons of SO_2 and 24 tons of NO_x (Shaws 2002). The supermarket also used the monitoring system data to develop long-term energy saving strategies and test the claims of energy efficient equipment manufacturers.

Complex monitoring systems have considerable up-front costs and potentially disruptive installations, and effective use may require operator retraining. In addition, because the concept is still developing, commercial availability of industry-specific systems is limited. These factors prevent widespread implementation. Further development, reduced initial costs and proven return on investment should lead to market adoption..

Best-Practice and Design Guides

Quality, readily accessible information on industry best practices and promising new technologies would be a helpful tool in the commercial refrigeration industry. In an evolving industry, where stringent energy-efficiency standards are becoming the norm, equipment manufacturers and operators would benefit from one-stop sources of trustworthy information: comprehensive best practices guides and unbiased reviews of emerging technologies.

With exhaustive guides on best practices and design options, equipment manufacturers and operators would be better informed of energy-saving opportunities. Exposure to this information would make the industry more aware of and likely to adopt energy-saving strategies and designs. Owners and operators would be better able to find operating strategies that lower energy consumption, manufacturers would be better able to compare design options to identify the most cost-effective technologies, and designers would be aware of industry innovations.

6 Impact of Regulatory and Voluntary Efficiency Programs

Government regulatory programs and voluntary energy-efficiency initiatives have set minimum efficiency requirements for many types of equipment and increased the awareness of energy efficiency, which has in turn become an increasingly important market driver for commercial refrigeration equipment. **Table 6-1** summarizes the major energy conservation standards (both existing and under development) and voluntary efficiency programs in the U.S. The two major U.S. voluntary efficiency programs in effect today are ENERGY STAR ® and the Consortium for Energy Efficiency (CEE). Section 6.1 provides information on regulatory programs and section 6.2 includes a summary of voluntary efficiency programs.

Table 6-1: Schedule of Energy-Conservation Standards and Voluntary Efficiency Programs

Equipment Type	**Effective Dates** [1]			
	Federal Standard (Final Rule Publish Date, Effective Date)	**State Standard** [4]	**ENERGY STAR**	**CEE Criteria**
Display Cases	Final Rule: March 2009 Effective: Jan. 2012	----	----	----
Compressor Racks	----	----	----	----
Supermarket Condensers	----	----	----	----
Walk-ins	Final Rule: Dec. 2007 [2] Effective: Jan. 2009	CA: 2007 CT, DC, MD, OR: 2009 RI: 2008	----	----
Food Service Equipment	----	----	Sept. 2001 (work-top tables only)	Dec. 2002, Jan. 2006 (work-top tables only) [5]
Reach-ins (including Bev Merchandisers)	Final Rule: July 2005 [3] Effective: Jan. 2010	CA: 2003, 2006 MD: 2005 WA: 2007 CT , OR: 2008	Sept. 2001 (solid door only)	Dec. 2002, Jan. 2006 (solid door only)
Ice Machines	Final Rule: July 2005 [3] Effective: Jan. 2010	CA: 2007 AZ, OR, WA: 2008	Jan. 2008 (air-cooled only)	Jan. 2006 (air-cooled only)
Vending Machines	Final Rule: due in Aug. 2009	CA: 2006	Phase 1: April 2004, Phase 2: April 2007	----

Sources: EPACT 2005, EISA 2007, and DOE 2009b for federal standards; ASAP 2009 for state standards; ENERGY STAR 2001 and 2008 for ENERGY STAR criteria; CEE 2006a and 2006b for CEE criteria
[1] The symbol, "----", means no program in place
[2] Passed as part of the Energy Independence and Security Act of 2007 (EISA)
[3] Passed as part of the Energy Policy Act of 2005 (EPACT)
[4] State standards are superseded by federal standards if/when they are put in place and become effective.
[5] The worktop surface may not add to the total energy consumption of the unit.

6.1 Regulatory Programs

The federal government (through DOE) and certain states have established energy-conservation standards for selected CRE. In most cases, standards are performance-based, meaning they place an explicit limit on energy consumption, based on prescribed test procedures. In the case of walk-ins, DOE has established prescriptive standards that regulate the physical characteristics of the unit to increase energy efficiency (i.e., R values for insulation). Federal and state regulatory programs are described further below.

6.1.1 Federal Energy Conservation Standards

Energy Policy Act of 2005

The Energy Policy Act of 2005 (EPACT), which was signed into law on August 8th, 2005, directed DOE to prescribe specific energy conservation standards for several residential and commercial products. The commercial equipment included solid-door reach-in refrigerators, freezers, and refrigerator-freezers, glass-door refrigerators and freezers, and automatic commercial ice machines manufactured on or after January 1, 2010. [16] Table 6-2 and **Table 6-3** provide EPACT 2005 current standards for reach-ins and ice machines, respectively. The maximum energy use for solid-door reach-ins according to the standard, as well as the CEE Tier 2 energy use criteria (described below), are shown as a function of cabinet volume in **Figure 6-1**. DOE expects to issue a revised standard for ice machines in 2010, which would become effective in 2015.

In addition, EPACT 2005 directs the U.S. Department of Energy to prescribe energy conservation standards (without specifying the standard) for beverage vending machines no later than August 8, 2009, and states that any such standards shall apply to beverage vending machines manufactured three years after the date of publication of the final rule that establishes those standards. The energy use of this equipment has never before been regulated at the Federal level.

[16] Ice machines covered under EPACT 2005 are limited to machines that produce cube type ice in capacities of 50 to 2500 pounds per 24-hour period.

Table 6-2: Standards for Reach-in Cabinets – EPACT 2005

Product Type	Door Type	Max. Daily Energy Consumption (kWh/day)	Typical Product Volume (ft^3)	Max. Annual Energy Consumption for Typical Volume (kWh/yr)
Refrigerator	Solid	0.10V + 2.04	48	2,497
	Glass*	0.12V + 3.34	48	3,322
Freezer	Solid	0.40V + 1.38	24	4,008
	Glass	0.75V + 4.10	24	8,067
Refrigerator-Freezer	Solid	0.27AV – 0.71 or 0.7 (if greater)	48	5,961
Refrigerator with pull-down application	Glass	0.126 V + 3.51	48	3,489

V = Internal Volume in ft^3
AV = Adjusted volume = (1.63 x freezer volume in ft^3) + refrigerator volume in ft^3
* Includes Beverage Merchandisers
Source: EPACT 2005

Table 6-3: Standards for Automatic Commercial Ice Machines - EPACT 2005 and State of California

Equipment Type	Cool Type	Harvest Rate, H (lbs ice/day) [1]	Maximum Energy Use (kWh/100lbs ice) [2]	Maximum Condenser Water Use (gal/100lbs ice)[3]
IMH	Water	< 500	7.80-0.0055H	200-0.022H
		≥500 and <1436	5.58-0.0011H	200-0.022H
		≥1436	4	N/A
	Air	< 450	10.26-0.0086H	N/A
		≥ 450	6.89-0.0011H	N/A
RCU (without remote compressor)	Air	< 1000	8.85-0.0038H	N/A
		≥ 1000	5.1	N/A
RCU (with remote compressor)	Air	< 934	8.85-0.0038H	N/A
		≥ 934	5.3	N/A
SCU	Water	< 200	11.4-0.019H	191-0.0315H
		≥ 200	7.6	191-0.0315H
	Air	< 175	18.0-0.0469H	N/A
		≥ 175	9.8	N/A

Source: EPACT 2005, CEC 2007 [1] Harvest rate is calculated using ARI Test Method 810-2008. Ambient temperature 90°F; water inlet temperature 70°F; water inlet pressure .30 ± 3 psig [2] H = Harvest Rate [1] Condenser water use is applicable to water-cooled ice machine

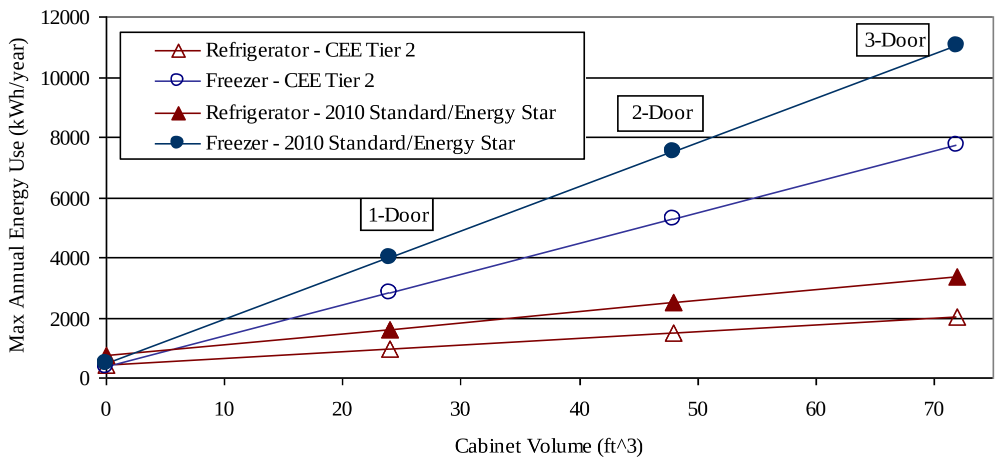


Figure 6-1: Efficiency Standards for Solid Door Reach-Ins

The Energy Independence and Security Act of 2007(EISA) includes prescriptive standards for walk-in coolers and freezers manufactured on or after January 1, 2009. The list below includes the requirements for walk-in coolers and freezers:

- Automatic door closers that firmly close all walk-in doors that have been closed to within 1 inch of full closure, except for doors wider than 3 feet 9 inches or taller than 7 feet.
- Strip doors, spring hinged doors, or other method of minimizing infiltration when doors are open.
- Wall, ceiling, and door insulation of at least R-25 for coolers and R-32 for freezers, except for glazed portions of doors and structural members.
- Floor insulation of at least R-28 for freezers.
- Evaporator fan motors under 1 hp and less than 460 volts must use electronically commutated motors or 3-phase motors
- Condenser fan motors under 1 hp must use electronically commutated motors, permanent split capacitor-type motors, or 3-phase motors.
- Interior lights that use light sources with an efficacy of 40 lumens per watt or more, including ballast losses if any. For light sources with an efficacy of 40 lumens per watt or less, a timer or device can be used that turns off the lights within 15 minutes of when the walk-in cooler or walk-in freezer is not occupied by people.
- Transparent reach-in doors for walk-in freezers and windows in walk-in freezer doors shall be of triple pane glass with either heat-reflective treated glass or gas fill.

- Transparent reach-in doors for walk-in coolers and windows in walk-in cooler doors shall be double-pane glass with heat-reflective treated glass and gas fill, or triple-pane glass with either heat-reflective treated glass or gas fill.
- Walk-ins with an anti-sweat heater and no anti-sweat heat controls shall have a total door rail, glass, and frame heater power draw of not more than 7.1 Watts per square foot of door opening (for freezers) and 3.0 Watts per square foot of door opening (for coolers).
- For walk-ins with an anti-sweat heater, anti-sweat heat controls, a total door rail, glass, and frame heater power draw of more than 7.1 Watts per square foot of door opening (for freezers), and 3.0 Watts per square foot of door opening (for coolers), the anti-sweat heat controls shall reduce the energy use of the anti-sweat heater in a quantity corresponding to the relative humidity in the air outside the door or to the condensation on the inner glass pane.

EISA also mandates that DOE set performance-based standards for walk-ins no later than January 1, 2012 with an effective date of January 1, 2015. The DOE rulemaking process for this new standard is underway.

Most recently, the U.S. Department of Energy published energy conservation standards for commercial refrigeration equipment on January 9, 2009 that apply to equipment sold in the U.S. on or after January 1, 2012. Covered equipment types include commercial ice-cream freezers; self-contained commercial refrigerators, freezers, and refrigerator-freezers without doors; and remote condensing commercial refrigerators, freezers, and refrigerator-freezers.   Table 6-4 shows the standards for the specified types of commercial refrigeration equipment.

Table 6-4: Federal Energy Conservation Standards for Commercial Refrigeration Equipment that Apply as of January 1, 2012)

Equipment Category	Condensing Unit Configuration	Equipment Family	Rating Temp. (°F)	Operating Temp. (°F)	Maximum Daily Energy Consumption (kWh/day)*
Remote Condensing Commercial Refrigerators and Commercial Freezers	Remote (RC)	Vertical Open (VOP)	38 (M)	≥ 32	0.82 x TDA + 4.07
			0 (L)	< 32	2.27 x TDA + 6.85
		Semivertical Open (SVO)	38 (M)	≥ 32	0.83 x TDA + 3.18
			0 (L)	< 32	2.27 x TDA + 6.85
		Horizontal Open (HZO)	38 (M)	≥ 32	0.35 x TDA + 2.88
			0 (L)	< 32	0.57 x TDA + 6.88
		Vert. Closed Transparent (VCT)	38 (M)	≥ 32	0.22 x TDA + 1.95

			0 (L)	< 32	0.56 x TDA + 2.61
		Horiz. Closed Transparent(HCT)	38 (M)	≥ 32	0.16 x TDA + 0.13
			0 (L)	< 32	0.34 x TDA + 0.26
		Vertical Closed Solid (VCS)	38 (M)	≥ 32	0.11 x V + 0.26
			0 (L)	< 32	0.23 x V + 0.54
		Horizontal Closed Solid (HCS)	38 (M)	≥ 32	0.11 x V + 0.26
			0 (L)	< 32	0.23 x V + 0.54
		Service Over Counter (SOC)	38 (M)	≥ 32	0.51 x TDA + 0.11
			0 (L)	< 32	1.08 x TDA + 0.22
Self-Contained Commercial Refrigerators & Commercial Freezers without Doors	Self-Contained (SC)	Vertical Open (VOP)	38 (M)	≥ 32	1.74 x TDA + 4.71
			0 (L)	< 32	4.37 x TDA + 11.82
		Semivertical Open (SVO)	38 (M)	≥ 32	1.73 x TDA + 4.59
			0 (L)	< 32	4.34 x TDA + 11.51
		Horizontal Open	38 (M)	≥ 32	0.77 x TDA + 5.55
			0 (L)	< 32	1.92 x TDA + 7.08
Commercial Ice-Cream Freezers	Remote (RC)	Vertical Open (VOP)	-15 (I)	≤ -5	2.89 x TDA + 8.7
		Semivertical Open (SVO)			2.89 x TDA + 8.7
		Horizontal Open (HZO)			0.72 x TDA + 8.74
		Vert. Closed Transparent (VCT)			0.66 x TDA + 3.05
		Horiz. Closed Transparent(HCT)			0.4 x TDA + 0.31
		Vertical Closed Solid (VCS)			0.27 x V + 0.63
		Horizontal Closed Solid (HCS)			0.27 x V + 0.63
		Service Over Counter (SVO)			1.26 x TDA + 0.26
	Self-Contained (SC)	Vertical Open (VOP)			5.55 x TDA + 15.02
		Semivertical Open (SVO)			5.52 x TDA + 14.63

		Horizontal Open (HZO)			2.44 x TDA + 9.
		Vert. Closed Transparent (VCT)			0.67 x TDA + 3.29
		Horiz. Closed Transparent(HCT)			0.56 x TDA + 0.43
		Vertical Closed Solid (VCS)			0.38 x V + 0.88
		Horizontal Closed Solid (HCS)			0.38 x V + 0.88
		Service Over Counter (SVO)			1.76 x TDA + 0.36
* k kWh = kilowatt-hours; TDA = total display area of the case; V = refrigerated volume of the case.					

Source: DOE 2009b

6.1.2 State Energy Conservation Standards

Nine states have adopted energy conservation standards for commercial refrigerators and freezers (see **Table 6-5**). Five of those states adopted standards that are already in effect. All existing state standards are equal to, and will be preempted by, the federal standards once they become effective on January 1, 2012.

Table 6-5: State Energy Conservation Standards for Commercial Refrigerators and Freezers

State	Effective Date of Standard(s)
Arizona	2010
California	2003, 2006
Connecticut	July 2007
Maryland	September 2005
New Jersey	2010
New York	2010
Oregon	2008
Rhode Island	2010
Washington	2007

Source: ASAP 2009

Five states and one district (California, Connecticut, District of Columbia, Maryland, Oregon and Rhode Island) have adopted state energy standards for walk-in coolers and freezers. These include a variety of prescriptive standards such as insulation levels, motor types, automatic door closers, etc. (Nadel 2006). California was the first state to institute standards, which became effective in January 2006. However, the federal standards that became effective on January 1, 2009, superseded all current state standards for walk-ins.

The States of Maryland, Washington, California, and Connecticut recently established standards, consistent with EPACT 2005, for commercial refrigeration energy use. However, California specifically excludes preparation tables from its proposed standards for commercial refrigeration. The states of Maryland, Washington and Connecticut define commercial refrigeration equipment very generally, leaving the question of which products are covered open to interpretation.

The States of Arizona, California, Oregon, and Washington currently regulate the energy and water use of automatic commercial ice machines manufactured on or after January 1, 2008, using the same standards as required by EPACT 2005 at the federal level (see **Table 6-3**. The Federal standard will preempt these state standards when it takes effect in January 2010. Rhode Island and New York have also adopted standards, but they will be preempted by the Federal standard, since they do not become effective until 2010.

The States of California, Connecticut, Maryland, Oregon, and Washington have set standards to regulate the energy use of solid- and glass-door refrigerators, freezers, and refrigerator-freezers. California was the first state to impose standards on reach-in units, which became effective in 2003. The Federal standard will supersede all current state standards when it takes effect in January 2010.

California Energy Commission

The California Energy Commission (CEC) is responsible for setting the California state regulations on energy consumption for products which are not regulated by the DOE. In addition to the standards mentioned above, California requires that all packaged beverage vending machines manufactured after January 1, 2006 meet energy efficiency standards. **Table 6-6** shows the California energy efficiency standards for beverage vending machines.

Table 6-6 California Energy Conservation Standards for Packaged Beverage Vending Machines

Appliance	Maximum Daily Energy Consumption (kWh)
Refrigerated canned and bottled beverage vending machines	$0.55(8.66 + (0.009 \times C))$

Source: CEC 2007

C = rated capacity (number of 12 ounce cans)

6.2 Voluntary Programs

Several voluntary programs exist to raise awareness about energy efficiency and promote adoption of high-efficiency products on the market. This section describes the criteria for the:

- ENERGY STAR program
- Consortium for Energy Efficiency Commercial Kitchens Initiative (CEE)
- Federal Energy Management Program (FEMP).

6.2.1 ENERGY STAR

ENERGY STAR is a joint labeling program of the U.S. Environmental Protection Agency and the U.S. Department of Energy designed to identify and promote energy- and water-efficient products to reduce greenhouse gas emissions. ENERGY STAR has become a widely recognizable symbol for energy efficiency. ENERGY STAR seeks to recognize products and equipment that fall in the top 25% of energy efficiency among competing products/equipment while meeting consumer needs for life, reliability, and functionality.

The ENERGY STAR criteria for the energy use of solid-door commercial refrigerators and freezers (i.e., reach-ins) have been in effect since September 1, 2001. ENERGY STAR is in the process of revising its criteria in anticipation of the new DOE standards coming into effect in 2010, which will be equal to the current ENERGY STAR criteria. Table 6-7 presents the reach-in criteria for the ENERGY STAR program and Federal Energy Management Program (FEMP). See section 6.2.3 for more information on FEMP.

Some food service equipment may qualify under the reach-in criteria, specifically work-top tables that have unrefrigerated work-top surfaces. Prep tables and buffet tables are not included.

Table 6-7: ENERGY STAR and FEMP Efficiency Criteria for Reach-ins[1]

Product Type	Efficiency Recommendation (kWh/day) [2]	Typical Product Volume (ft^3)	Max. Annual Energy Consumption for Typical Volume (kWh/yr)
Refrigerator	$\leq 0.10V + 2.04$	48	2497
Refrigerator-Freezer	$\leq 0.27AV - 0.71$	48	4471
Freezer	$\leq 0.40V + 1.38$	24	4008
Ice Cream Freezer	$\leq 0.39V + 0.82$	24	3716

Source: ENERGY STAR 2001, FEMP 2002
[1] Federal Energy Management Program (FEMP); See section 6.2.3.
[2] V = cabinet volume, AV = adjusted cabinet volume as defined in the Code of Federal Regulations DOE Energy Test Procedure section (10 CFR 430). Adjusted volume takes into account the larger impact of freezer volume as compared to refrigerator volume in refrigerator-freezers.

The ENERGY STAR criteria for the energy and water use of commercial ice machines have been effective since January 1, 2008. These criteria only cover air-cooled ice cube machines. Table 6-8 shows the criteria by equipment type. ENERGY STAR plans to revisit the ice machine criteria once the revision processes for the industry test procedures by the Air-conditioning, Heating, and Refrigeration Institute (ARI 810-2007: Performance Rating of Automatic

Commercial Ice Makers)[17] and American Society for Heating, Refrigerating, and Air-conditioning Engineers (ASHRAE 29: Methods of Testing Automatic Ice Makers) are complete. At that time, performance requirements for flake and nugget ice machines will be considered and shared with industry stakeholders for review and comment.

Table 6-8: ENERGY STAR Efficiency Requirements (Air-Cooled Cubers Only)

Equipment Type	Harvest Rate, H [1] (lbs ice/day)	Maximum Energy Use[2] (kWh/100 lbs ice)	Maximum Potable Water Use (gal/100 lbs ice)
IMH	< 450	9.23 – 0.0077H	≤ 25
	≥ 450	6.20 – 0.0010H	≤ 25
RCU (without remote compressor)	< 1000	8.05 – 0.0035H	≤ 25
	≥ 1000	4.64	≤ 25
RCU (with remote compressor)	< 934	8.05 – 0.0035H	≤ 25
	≥ 934	4.82	≤ 25
SCU	< 175	16.7 – 0.0436H	≤ 35
	≥ 175	9.11	≤ 35

Source: ENERGY STAR 2008
[1] Harvest rate is calculated using ARI Test Method 810-2008. Ambient temperature 90°F; water inlet temperature 70°F; water inlet pressure .30 ± 3 psig
[2] H = Harvest Rate

Lastly, ENERGY STAR has a two-tiered set of criteria in place for refrigerated beverage machines (see **Table 6-9**). Tier 1 has been in effect for new machines since April 1, 2004, and for refurbished machines since April 31, 2006. The Tier 2 criteria went into effect on July 1, 2007 for all newly manufactured packaged beverage vending machines.

Table 6-9 ENERGY STAR and FEMP* Criteria for Refrigerated Beverage Vending Machines

Appliance	Energy Consumption (kWh)	
	Tier 1 – Effective April 1, 2004	Tier 2 – Effective January 1, 2007
Vending Machine	Y = 0.55 [8.66 + (0.009 x C)]	Y = 0.45 [8.66 + (0.009 x C)]

Source: ENERGY STAR 2008, FEMP 2008
Y = 24 hr energy consumption (kWh/day) after the machine has stabilized
C = rated capacity (number of 12 ounce cans)
*Federal Energy Management Program (FEMP); See section 6.2.3.

For more information regarding the ENERGY STAR criteria, visit the ENERGY STAR website (http://www.energystar.gov/).

[17] http://www.ahrinet.org/Content/FindaStandard_218.aspx

6.2.2 Consortium for Energy Efficiency Commercial Kitchens Initiative

The Consortium for Energy Efficiency (CEE) develops initiatives for its North American members to promote the manufacture and purchase of energy-efficient products and services. CEE members include utilities, statewide and regional market-transformation administrators, environmental groups, research organizations and state energy offices in the U.S. and Canada. The goal of CEE initiative is to provide clear and credible definitions in the marketplace as to what constitutes highly efficient energy and water performance in refrigeration equipment, as well as cooking and sanitation equipment, and then to help streamline the selection of products through a targeted market strategy.

CEE launched an initiative for solid-door reach-in refrigerators and freezers in December 2002. A glass-door reach-in refrigerator and freezer initiative followed in December 2003. The current specifications are shown in Table 6-10. Tier 1 is identical to ENERGY STAR specifications for solid door reach-ins. CEE plans to revise its reach-in specification in the near future in response to the new Federal standard for reach-ins (effective in January 2010).

Just as in the ENERGY STAR program, some food service equipment may qualify under the reach-in criteria, specifically work-top tables that have work-top surfaces that do not add to the total energy consumption of the unit. Prep tables and buffet tables are not included.

Table 6-10: CEE Efficiency Specifications for Reach-Ins

Product Type	Door Type	Typical Volume (ft3)	Efficiency Specifications (kWh/day)		Max. Annual Energy Use for Typical Size (kWh/year)	
			Tier 1	Tier 2	Tier 1	Tier 2
Freezer	Solid	24	.40V + 1.38 (ENERGY STAR)	.28V + .97 (ENERGY STAR + 30%)	2,497	1,497
Refrigerator	Solid	48	.10V + 2.04 (ENERGY STAR)	.06V + 1.22 (ENERGY STAR + 40%)	3,322	2,379
Refrigerator	Glass	48	.12V + 3.34 (25% of top-performing products)	.086V + 2.39 (28% more than Tier 1)	4,008	2,807

Source: CEE 2006a

The CEE issued efficiency specifications for certain commercial ice machines, effective January 1, 2006. The ice machines covered include air-cooled ice cube machines, shown in **Table 6-11**, and water-cooled ice cube machines using a closed loop system or a system with a remote evaporative condenser, (i.e., cooling tower), shown in **Table 6-12**. Once-through or pass-through cooling systems are not covered in this specification. Tier 1 aligns with FEMP Standards[18], and Tier 2 aligns with ENERGY STAR requirements.

CEE indicates that they plan to include flake and nugget ice machines once a) a test procedure is available, and b) a robust database is established that can be used to derive performance requirements.

Table 6-11 CEE Tier Requirements (Air-Cooled Cubers)

Equipment Type	Harvest Rate, H* (lbs ice/day)	Specification	Corresponding Base Specification	Maximum Energy Use (kWh/100 lbs ice)	Maximum Potable Water Use (gal/100 lbs ice)
IMH	< 450	CEE Tier 1	Approx. FEMP	10.26-0.0086H	-
		CEE Tier 2	10% below Tier 1/ ENERGY STAR	9.23-0.0077H	≤ 25
		CEE Tier 3	15% below Tier 1	8.72-0.0073H	≤ 20
	≥ 450	CEE Tier 1	Approx. FEMP	6.89-0.0011H	-
		CEE Tier 2	10% below Tier 1/ ENERGY STAR	6.20-0.0010H	≤ 25
		CEE Tier 3	15% below Tier 1	5.86-0.0009H	≤ 20
RCU (without remote compressor)	< 1000	CEE Tier 1	Approx. FEMP	8.85-0.0038H	-
		CEE Tier 2	9% below Tier 1/ ENERGY STAR	8.05-0.0035H	≤ 25
		CEE Tier 3	15% below Tier 1	7.52-0.0032H	≤ 20
	≥ 1000	CEE Tier 1	Approx. FEMP	5.1	-
		CEE Tier 2	9% below Tier 1/ ENERGY STAR	4.64	≤ 25
		CEE Tier 3	15% below Tier 1	4.34	≤ 20
RCU (with remote compressor)	< 934	CEE Tier 1	Approx. FEMP	8.85-0.0038H	-
		CEE Tier 2	9% below Tier 1/ ENERGY STAR	8.05-0.0035H	≤ 25
		CEE Tier 3	15% below Tier 1	7.52-0.0032H	≤ 20

[18] The Federal Energy Management Program is discussed below in section 6.2.3.

	≥ 934	CEE Tier 1	Approx. FEMP	5.3	-
		CEE Tier 2	9% below Tier 1/ ENERGY STAR	4.82	≤ 25
		CEE Tier 3	15% below Tier 1	4.51	≤ 20
SCU	< 175	CEE Tier 1	Approx. FEMP	18.0-0.0469H	-
		CEE Tier 2	7% below Tier 1/ ENERGY STAR	16.7-0.0436H	≤ 35
		CEE Tier 3	15% below Tier 1	15.3-0.0399H	≤ 30
	≥ 175	CEE Tier 1	Approx. FEMP	9.8	-
		CEE Tier 2	7% below Tier 1/ ENERGY STAR	9.11	≤ 35
		CEE Tier 3	15% below Tier 1	8.33	≤ 30
Source: CEE 2008 * Harvest rate is calculated using ARI Test Method 810-2008. Ambient temperature 90°F; water inlet temperature 70°F; water inlet pressure .30 ± 3 psig.					

Table 6-12 CEE Tier Requirements (Water-Cooled Cubers)

Equipment Type	Harvest Rate, H (lbs ice/day)	Specification	Corresponding Base Specification [1]	Maximum Energy Use (kWh/100 lbs ice) [2]	Maximum Potable Water Use (gal/100 lbs ice)	Maximum Condenser Water Use (gal/100 lbs ice)
IMH	< 500	CEE Tier 1	Approx. FEMP	7.80-0.0055H	-	200-0.022H
		CEE Tier 2	10% below Tier 1	7.02-0.0049H	<=25	**
		CEE Tier 3	15% below Tier 1	6.63-0.0047H	<=20	**
	≥ 450 and < 1436	CEE Tier 1	Approx. FEMP	5.58-0.0011H	-	200-0.022H
		CEE Tier 2	8% below Tier 1	5.13-0.0010H	<=25	**
		CEE Tier 3	15% below Tier 1	4.74-0.0009H	<=20	**
	≥ 1436	CEE Tier 1	Approx. FEMP	4.00	-	200-0.022H
		CEE Tier 2	8% below Tier 1	3.68	<=25	**

		CEE Tier 3	15% below Tier 1	3.40	<=20	**
SCU	< 200	CEE Tier 1	Approx. FEMP	11.4-0.0190H	-	191-0.0315H
		CEE Tier 2	7% below Tier 1	10.6-0.0177H	<=35	**
		CEE Tier 3	15% below Tier 1	9.69-0.0162H	<=30	**
	≥ 200	CEE Tier 1	Approx. FEMP	7.60	-	191-0.0315H
		CEE Tier 2	7% below Tier 1	7.07	<=35	**
		CEE Tier 3	15% below Tier 1	6.46	<=30	**

Source: CEE 2008
[1] Federal Energy Management Program is discussed in section 6.2.3.
[2] Harvest rate, H, is calculated using ARI Test Method 810-2008. Ambient temperature 90°F; water inlet temperature 70°F; water inlet pressure .30 ± 3 psig

For more information regarding the CEE criteria, visit the CEE website (http://www.cee1.org/).

6.2.3 Federal Energy Management Program

DOE's Federal Energy Management Program (FEMP) works to reduce the cost and environmental impact of the Federal government by advancing energy efficiency and water conservation, promoting the use of distributed and renewable energy, and improving utility management decisions at Federal sites. Federal buyers are required by EPACT to purchase FEMP-designated equipment including reach-in refrigerators, freezers, and refrigerator-freezers, and commercial ice machines.

The FEMP adopted ENERGY STAR criteria as its efficiency recommendation for reach-ins in May 2002, shown above in **Table 6-7**.

The FEMP efficiency recommendations for commercial ice machines, which have been effective since November 2000, are shown in **Table 6-13**.

Table 6-13: FEMP Efficiency Recommendations for Ice Machines

Equipment Type	Cooling Type	Harvest Rate (lbs/day)	Recommended Energy Consumption (kWh / 100 lbs of Ice)
IMH	Air-Cooled	101-200	9.4 kWh or less
		201-300	8.5 kWh or less
		301-400	7.2 kWh or less
		401-500	6.1 kWh or less
		501-1000	5.8 kWh or less

		1001-1500	5.5 kWh or less
	Water-Cooled	201-300	6.7 kWh or less
		301-500	5.5 kWh or less
		501-1000	4.6 kWh or less
		1001-1500	4.3 kWh or less
		> 1500	4.0 kWh or less
SCU	Air-Cooled	101-200	10.7 kWh or less
	Water-Cooled	101-200	9.5 kWh or less
RCU	Air-Cooled	201-300	7.6 kWh or less
		301-400	8.1 kWh or less
		401-500	7.0 kWh or less
		501-1000	6.2 kWh or less
		1001-1500	5.1 kWh or less
		> 1500	5.3 kWh or less

Source: FEMP 2000

FEMP adopted ENERGY STAR criteria as its efficiency recommendation for vending machines in December 2008, shown above in **Table 6-9**.

For more information regarding FEMP, visit the FEMP website (http://www1.eere.energy.gov/femp/).

7 Recommendations

Implementation of currently available high efficiency technologies could generate substantial energy savings for users of commercial refrigeration equipment. Our analysis suggests that energy savings of nearly 35% relative to typical new equipment are possible. Research, development and demonstration (RD&D) leading to commercialization of advanced technologies that are not yet in use could produce additional savings. DOE can accelerate the adoption of existing energy saving technologies in several ways and can also support research and development of advanced technologies to facilitate commercialization. Approaches to facilitate market adoption of energy efficient technologies may be categorized as follows:

1) *Mandatory Efficiency Standards:* Mandatory minimum efficiency standards for many appliances and equipment including commercial refrigeration systems are developed by the DOE and are applicable nationwide. For products not covered by federal standards, some states, most notably California through Title 20 regulations, develop standards that are only applicable within their states
2) *Voluntary Efficiency Programs:* These include programs such as the Energy Star program managed by DOE and EPA, as well as programs run by energy efficiency advocacy organizations such as the Consortium for Energy Efficiency (CEE).
3) *Demonstration Programs:* These programs demonstrate the benefits, risks, and costs of emerging technologies, usually in a field setting where real world conditions can be used to rigorously evaluate the technology. The DOE supports such activities through its Technology Validation and Market Introduction (TVMI) Program and can also facilitate large volume group purchases to reduce the purchase price of low volume products. Utilities also support demonstrations of emerging technologies.
4) *Education, Outreach, and Training:* Examples include programs to train installation contractors to ensure quality installations, or educating engineers about the benefits of emerging technologies. DOE has supported such activities directly and through industry organizations and trade associations, as have other energy efficiency advocacy groups and utilities.
5) *Research and Development:* The DOE has historically played a leading role in sponsoring R&D of new energy efficient technologies for many types of building equipment, including commercial refrigeration systems. DOE funding ensures support for high risk, long term R&D which is unlikely to be funded by the private sector. The DOE can also support R&D that can help overcome institutional barriers to new technologies. For example, the DOE played a leading role in funding pre-competitive research on alternatives to CFC and HCFC refrigerants and could also support research on next generation non-fluorocarbon refrigerants.

7.1 Mandatory Minimum Efficiency Standards

DOE should develop energy test procedures and energy efficiency standards for those types of commercial refrigeration equipment that are not currently covered, including compressor racks, supermarket condensers, and food service equipment. Additionally, for all established standards, DOE should continually reassess their suitability in light of technological advancements which may significantly affect the cost-effectiveness of higher efficiency standards.

Test procedures should also be examined and updated to ensure that they do not inhibit the application of energy saving technologies by not crediting the energy savings attributable to those technologies. For example, test procedures for vending machines and beverage merchandisers are steady state tests which do not account for savings that could be achieved using energy management systems that reduce lighting and compressor energy consumption during periods of low activity. Consequently, the test procedure and standards discourage adoption of these technologies, which can be highly cost effective and substantially reduce energy consumption. Manufacturers may instead choose design options which may be less efficient and less cost-effective in order to meet the energy efficiency standards. Similarly, because energy consumption tests for self-contained refrigerators such as reach-ins are single rating point tests, they discourage adoption of technologies such as variable speed compressors which save energy over a wide operating range but may not be more efficient at the rating point. This issue is addressed in residential unitary air conditioners by rating equipment based on performance at multiple conditions (defined by the Seasonal Energy Efficiency Ratio or SEER) rather than a single rating point. A similar approach could be used for refrigeration.

7.2 Voluntary Programs

There are a number of currently available technologies, discussed in section 4, that are used in some commercial refrigeration equipment today or could be implemented with modest effort. However, they still face significant barriers to widespread adoption. Awareness programs, technology validation, analysis and design tools, and utility incentives can significantly enhance the adoption of these high-efficiency technologies.

7.2.1 Expand ENERGY STAR Program

The DOE/EPA ENERGY STAR program has proven to be very effective in raising awareness of energy efficiency for covered products and is a widely recognized brand among consumers. This brand recognition can be leveraged by manufacturers to credibly demonstrate the benefits of energy efficient products to customers who may be unaware of energy efficient options. DOE should consider expanding this program to cover supermarket refrigeration systems, walk-ins, and refrigerated food service equipment.

7.2.2 Expand Consortium for Energy Efficiency (CEE) Programs

CEE, a public benefits corporation, develops initiatives for its utility members to promote the purchase of energy efficient equipment. CEE specifications are used by many utilities as the basis for their incentive programs. CEE initiatives cover some commercial refrigeration equipment such as ice machines and self-contained refrigerators and freezers. Expanding the program to cover supermarket refrigeration systems and walk-in coolers and freezers would help spur adoption of high efficiency options for these products. The DOE could work with CEE to support establishment of such programs.

7.2.3 Leverage Technology Validation and Market Introduction and Retailer Energy Alliance Programs

The DOE Building Technologies TVMI program can help increase the acceptance of energy efficient commercial refrigeration technologies through field demonstrations to document costs and benefits of energy efficient equipment. Customers are often skeptical of energy savings claims made by manufacturers. Rigorous, independent field evaluations sponsored by the DOE can help allay concerns about the objectivity and validity of energy savings claims. Such field evaluations can also serve as a basis for utility incentive programs to promote energy efficient technologies. Case studies also educate customers and build awareness of energy efficient options. The Retailer Energy Alliance (REA) can also facilitate large volume group purchases to reduce the purchase price of such options. **Table 7-1** shows some technology options that could benefit from further technology validation or group purchases.

Table 7-1: Key Technology Options for Technology Validation and Market Introduction

Improvement	Targeted Applications	Technology Options
Improved Case Lighting Efficiency	• Supermarket display cases	• High-efficiency fluorescent or LED lighting • Placement of lamps and ballasts outside of refrigerated cabinet
Energy Management Systems	• Beverage merchandisers • Refrigerated vending machines	• Proximity sensors and energy management systems
Reduced Cabinet Thermal Losses	• Supermarket display cases • Beverage merchandisers • Refrigerated vending machines • Reach-ins • Walk-ins	• Reducing thermal short circuits • Glass doors with double or triple glazing using inert gas fill • Improved insulation
Reduced Cabinet Heat Generation	• Supermarket display cases • Beverage merchandisers	• Hot-gas defrost • Hot-gas anti-sweat heaters • On-demand defrost • On-demand anti-sweat

	• Refrigerated vending machines • Reach-ins • Walk-ins	heaters • High-efficiency evaporator fan motors
Advanced compressors	• Supermarket Refrigeration Systems • Walk-ins • Reach-ins • Vending Machines	• Modulating compressors • Vapor injection economizer
Improved System Control s	• Supermarket refrigeration systems	• Electronic expansion valves and case controllers

7.2.4 Modeling & Design Tools for Manufacturers

More accurate modeling tools are needed for manufacturers to optimize designs and maximize the benefits of high efficiency technologies both in self-contained equipment and in large split-systems. DOE could sponsor development of more advanced software for refrigeration system design. There is also a need for published literature on the up-to-date best practices for refrigeration system design and field implementation that could be used by manufacturers, system designers, and operators.

7.3 Support Research and Development of Advanced Technologies

Continued R&D support by DOE is essential to ensuring that advanced energy efficient technologies and new innovations are developed and implemented in future commercial refrigeration systems. Chapter 5 describes selected energy savings technologies currently under development. The technologies span the range of development status from small scale technology demonstrations to full scale prototypes and products. They cover a range of solutions, including several alternatives to conventional air-to-air vapor-compression refrigeration cycles. Other technologies such as fiber optic lighting and vacuum insulated panels aim to address potential energy savings from components such as lighting and compressors, and cabinets.

In **Figure 7-1**, we depict the technology and market maturity as well as the technical savings potential for each of the presented technologies. The precise position of each technology is uncertain and not critical to our overall recommendations. We recommend focusing attention and funding on technologies with high technology potential (on the right side of the graph). Technologies whose vertical position is in the central or lower region of the graph may be most worthy of DOE R&D support. For these technologies, significant technical barriers still exist, and further research is needed to overcome them. Those technologies in the upper region may be more suitable for demonstration and validation. For those more mature technologies, the technical barriers are lower but other obstacles such as high prices, low reliability, and some technical uncertainties need to be overcome.

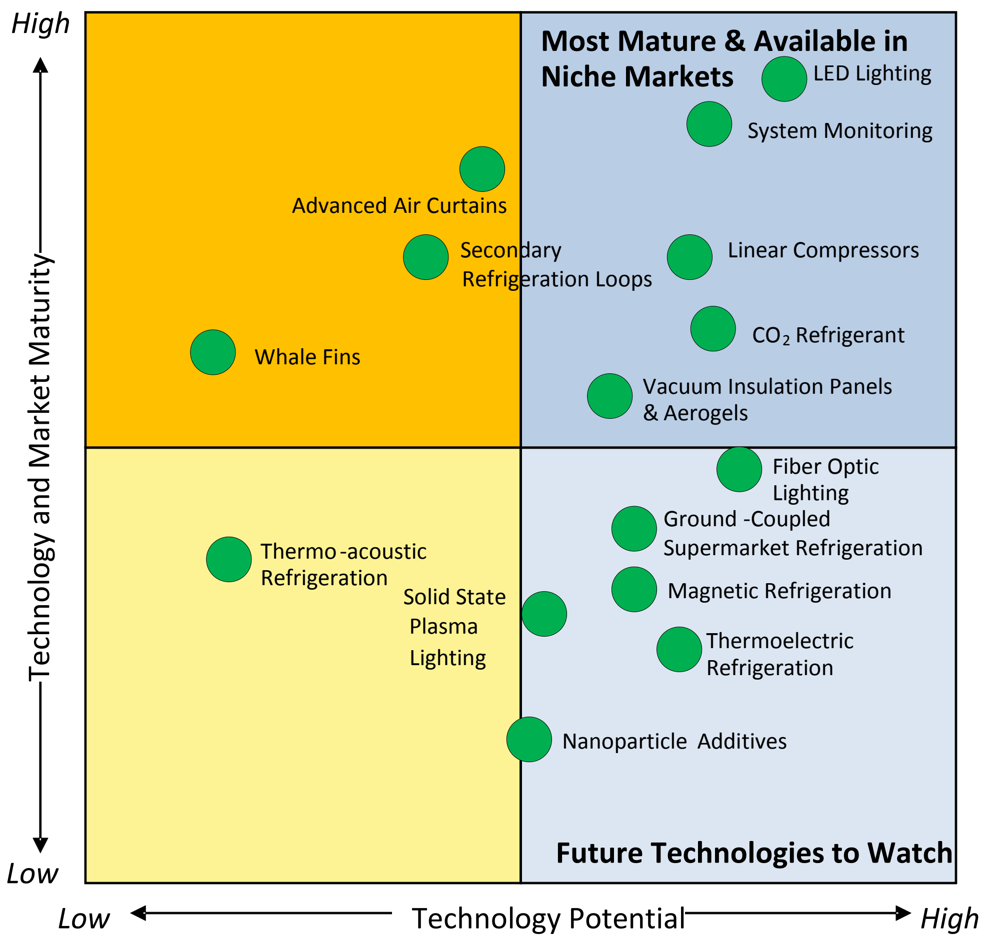


Figure 7-1: Advanced CRE Technologies Classified by Technology Maturity and Potential

References

ACR News 2008. "Nanotechnology Could Boost Chiller Efficiency", ACR News, September 2008.

ADL 1996. "Energy Potential Report for Commercial Refrigeration Equipment", Final Report, prepared by Arthur D. Little, Inc. for U.S. Department of Energy, Office of Building Technologies, June 1996.

Al-Dadah, 1992. R. K. Al-Dadah, T. G. Karayiannis, R. W. James and P. H. G. Allen, "EHD enhanced heat transfer: effect on the performance of a refrigeration system", *Inst. Chem. Engrs Symp*, Series 2, 1229-1236 (1992).

Appliance Magazine 2005. The Share-of-Market Picture for 2004, Appliance Magazine, September 2005. (Refrigerated display cases)

Appliance Magazine 2007. 54th Annual Report: Statistical Review 1998-2006, Appliance Magazine, May 2007. (Ice machines)

Appliance Magazine 2008. 55th Annual Report: Statistical Review 1998-2007, Appliance Magazine, May 2008. (Reach-ins)

AHRI 2007. ARI 810-2007: Performance Rating of Automatic Commercial Ice Makers, Air-Conditioning, Heating, and Refrigeration Institute, 2007. http://www.ahrinet.org/ARI/util/showdoc.aspx?doc=455

AHRI 2008. Directory of Certified Product Performance, Air-Conditioning, Heating, and Refrigeration Institute, 2008. http://www.ahridirectory.org/ahridirectory/pages/home.aspx

Antonijevic 2008. D.L Antonijevic. "Carbon Dioxide as the Replacement for Synthetic Refrigerants in Mobile Air Conditioning", Thermal Science, 12, 55-65 (2008).

ASAP 2009. Energy Efficiency Standards Adopted and Pending by State. Appliance Standards Awareness Project and Northeast Energy Efficiency Partnerships, January 2009. http://www.standardsasap.org/documents/StatestandardsstatusgridJanuary2009update.pdf

ASHRAE 2007. "Magnetic Refrigeration", Dieckmann, John, Dr. Roth, Kurt, and Brodrick, James, ASHRAE Journal. August, 2007.

Automatic Merchandiser 2007. Automatic Merchandiser Magazine, August 2007. http://www.amonline.com/

Bally 2008. Personal communication with Guy Senter, Bally Refrigerated Boxes, April 14, 2008.

Bell 2008. Lon E. Bell *et al*. "Cooling, Heating, Generating Power, and Recovering Waste Heat with Thermoelectric Systems". *Science*, **321**, 1457 (2008).

Beverage-Air 2008a. Personal communication with Jeff Hogston, Beverage-Air, July 2008.

Beverage-Air 2008b. Catalog, Beverage-Air Corporation, 2008. http://www.beverage-air.com/foodservice.html

Bohn 2007. Air-Cooled Condensers Technical Guide for Monarch™ and Ambassador™ Series (page 17). Heatcraft Refrigeration Products, LLC. February 2007.

Brown 2002. J.S. Brown, Y. Kim, and P.A. Domanski. "Evaluation of Carbon Dioxide as R-22 Substitute for Residential Air-Conditioning", ASHRAE Transactions, Vol. 108, No. 2, 954-964 (2002).

Bruk 2003. "Magnetic Refrigeration at Room-Temperature", Bruk, E., Encyclopedia of Materials: Science and Technology, Published by Elsevier Science Ltd, 2003.

Caroll Coolers 2008. Personal Communication with David Teske, Caroll Coolers, April 25, 2008.

CBECS 2003. Commercial Buildings Energy Consumption Survey 2003, Energy Information Administration, released September 2008. http://www.eia.doe.gov/emeu/cbecs/contents.html

Chen 2003. Chen, G., Dresselhaus, M. S., Dresselhaus, G., Fleurial, J. P. & Caillat, T. "Recent developments in thermoelectric materials". *Int. Mater. Rev.* **48,** 45–66 (2003).

CEC 2004. "Investigation of Secondary Refrigeration Loop Supermarket Refrigeration Systems", California Energy Commission, March 2004.

CEC 2008, 2009. Appliances Database, California Energy Commission, March 2008 (Food Service Equipment, Ice Machines), May 2008 (Reach-ins), May 2009 (Refrigerated Vending Machines). http://www.energy.ca.gov/appliances/database/

CEC 2007. 2007 Appliance Efficiency Regulations, California Energy Commission, December 2007. http://www.energy.ca.gov/appliances/index.html

CEE 2006a. High Efficiency Specifications for Solid Door Refrigerators and Freezers and Glass Door Refrigerators, Consortium for Energy Efficiency, January 2006. http://www.cee1.org/com/com-kit/files/RefrigerationSpecificationFINAL20080617.pdf

CEE 2006b. High Efficiency Specifications for Ice Machines, Consortium for Energy Efficiency, January 2006. http://www.cee1.org/com/com-kit/files/Ice%20MakersSpecificationFINAL20080620.pdf

CSG 2008. Personal communication with Joe Bina of Chain Store Guide (CSGIS), May 16, 2008.

Craig Industries 2008. Personal communication with Ellis Craig, Craig Industries, May 16, 2008.

CrownTonka 2008. Personal communication with Mike Kahler, CrownTonka, April 28, 2008.

Dieckmann 2007. Dieckmann, John, Dr. Roth, Kurt, and Brodrick, James. "Magnetic Refrigeration". *ASHRAE Journal.* August, 2007.

DoC 2007. Current Industrial Report, U.S. Department of Commerce, July 2007.

DOE 2000. "Electrocaloric Refrigeration for Room Temperature Refrigeration", *EERE: Inventions and Innovation*, Department of Energy, November 2000.

DOE 2006. 2004 Commercial Energy End-Use Splits, Buildings Energy Data Book, Office of Energy Efficiency and Renewable Energy, U.S. Department of Energy, September 2006. http://buildingsdatabook.eere.energy.gov/ChapterView.aspx?chap=3#1

DOE 2008a. 2006 Commercial Energy End-Use Splits, Buildings Energy Data Book, Office of Energy Efficiency and Renewable Energy, U.S. Department of Energy, September 2008. http://buildingsdatabook.eere.energy.gov/ChapterView.aspx?chap=3#1

DOE 2008b. Energy Conservation Standards for Beverage Vending Machines: ANOPR Technical Support Document, U.S. Department of Energy, June 2008. http://www1.eere.energy.gov/buildings/appliance_standards/commercial/beverage_machines_anopr_spreadsheets.html

DOE 2009a. Energy Conservation Standards for Commercial Refrigeration Equipment: Technical Support Document, U.S. Department of Energy, January 2009, http://www1.eere.energy.gov/buildings/appliance_standards/commercial/refrig_equip_final_rule.html

DOE 2009b. Energy Conservation Standards for Commercial and Industrial Equipment: Commercial Refrigeration Equipment, 10 CFR Part 431, Department of Energy, January 2009. http://www1.eere.energy.gov/buildings/appliance_standards/commercial/refrig_equip_final_rule.html

DOE 1997. Technical Support Document for Energy Conservation Standards for Room Air Conditioners, U.S. Department of Energy, September 1997.

Dresselhaus 2007. Dresselhaus, M. S. *et al.* "New directions for low-dimensional thermoelectric materials". *Adv. Mater.* **19,** 1043–1053 (2007).

Eames 1997. Eames, I. W., and Sabir, H. M. Potential benefits of electrohydrodynamic enhancement of two-phase heat transfer in the design of refrigeration systems". *Applied Thermal Engineering.* **17,** 79-92 (1997).

EIA 2008. "Technology Forecast Updates – Residential and Commercial Building Technologies - Reference Case", Energy Information Administration, September 2008.

EIA 2009. "Average Retail Price of Electricity to Ultimate Customers by End-Use Sector, by State, Year-to-Date through December 2008", Table 5.6.B, Electric Power Monthly, March 2009 http://www.eia.doe.gov/cneaf/electricity/epm/epm_sum.html.

Elstat 2008. Personal communication with Crystal Burdette, Elstat Electronics, Ltd., June 12, 2009. http://www.elstatgroup.com/company/energy.aspx.

Emerson 2009. Personal communication with Hung Pham, Emerson Climate Technologies, Emerson Electric Company, May-June 2009.

ENERGY STAR 2001. ENERGY STAR Program Requirements for Commercial Solid Door Refrigerators and Freezers, Environmental Protection Agency & Department of Energy, September 2001.
http://www.energystar.gov/index.cfm?fuseaction=products_for_partners.showRefrigComm

ENERGY STAR 2008. ENERGY STAR Program Requirements for Commercial Ice Machines, Environmental Protection Agency & Dept of Energy, January 2008.
http://www.energystar.gov/index.cfm?fuseaction=products_for_partners.showIceMachinesComm

ENERGY STAR 2009. Commercial Ice Machines Qualified Model List, Environmental Protection Agency & Dept of Energy, March 2009.
http://www.energystar.gov/index.cfm?c=comm_ice_machines.pr_comm_ice_machines

EPA 2007. Supermarket Energy Use Profile, Environmental Protection Agency, 2007.

EPACT 2005. Energy Policy Act of 2005, Section 136(c), July 2005.

FEMP 2000. How to Buy an Energy-Efficient Commercial Ice-Cube Machine, Federal Energy Management Program, November 2000.
http://www1.eere.energy.gov/femp/procurement/eep_ice_makers.html#efficiency

FEMP 2002. How to Buy Energy-Efficient Commercial Refrigerators and Freezers, Federal Energy Management Program, May 2002.
http://www1.eere.energy.gov/femp/procurement/eep_crf.html

FEMP 2008. How to Buy Energy-Efficient Refrigerated Beverage Vending Machines, Federal Energy Management Program, December 2008. http://www1.eere.energy.gov/femp/procurement/eep_beverage_vending_machine.html

Fisher-Nickel 2003. Zabrowski, D., and Cowen, D., "Refrigerated Prep Table Performance Testing". Food Service Technology Center, Fisher-Nickel, Inc., 2003.

FMI 2007. Facts and Figures: Top Food Retailers, Food Marketing Institute, April 2007. http://www.fmi.org/docs/facts_figs/top_retailers.pdf

FMI 2008a. Marketing Costs, Food Marketing Institute, August 2008. http://www.fmi.org/docs/facts_figs/MarketingCosts.pdf

FMI 2008b. Industry Overview 2008, Food Marketing Institute, accessed May 2009. http://www.fmi.org/facts_figs/?fuseaction=superfact

Foster Miller 1990. "Guide for the Selection of Supermarket Refrigeration Systems", Walker, D., Tsaros, T., and Deming, G., Foster-Miller, Inc. for EPRI, March 1990.

Freedonia 2004. Energy Savings in Refrigerated Walk-in Boxes, DOE/EE-0170, 2004.

FRPERC 2003. James, Stephen J. "Developments in domestic refrigeration and consumer attitudes," *Food Refrigeration and Process Engineering Research Centre (FRPERC)*, 2003. University of Bristol, UK. Bulletin of the IIR - No 2003-5. http://www.iifiir.org/en/doc/1051.pdf

Glacier Bay 2009a. Barrier Ultra-Rtm High-Performance Insulation, Glacier Bay, Inc, accessed January 2009. http://www.glacierbay.com/ultra-r.asp

Glacier Bay 2009b. "Vacuum Insulation Panels (VIPs) Principles, Performance and Lifespan". Glacier Bay, Inc., accessed January 2009. http://www.glacierbay.com/vacpanelinfo.asp

Gschneidner 2005. K.A. Gschneidner Jr., V.K. Pecharsky, and A.O. Tsokol "Recent developments in magnetocaloric materials", *Rep. Prog. Phys*. **68** (2005) 1479–1539.

Gschneidner 2008. K.A. Gschneidner, Jr.*, V.K. Pecharsky. "Thirty years of near room temperature magnetic cooling: Where we are today and future prospects." *International Journal of Refrigeration.* September, 2008.

GSET Research Journal 2006. "Thermoacoustic Refrigeration", Newman, J., Cariste, B., Queiruga, A., GSET Research Journal, 2006

Hamilton 2008. Hamilton, T. "Whale of an Idea Has Lots of Fans", Toronto Star, March 3, 2008. http://www.thestar.com/Business/CleanBreak/article/308728

Heinecke 2006. "Single-phase Electric Motors", Heinecke, K. LEESON Electric Corporation, November 2006. http://www.iprocessmart.com/leeson/leeson_singlephase_article.htm

Hirano 2003. Hirano, N. 2003 Paper No K7.002, *Am. Phys. Soc. Meeting*, March 4, 2003,Austin, TX.

ICOPT 1997. "Tests of Supermarket Display Cases when Operating with Secondary Refrigerants", International Conference on Ozone Protection Technologies, Baltimore MD, November 1997.

IEA 2003. IEA Annex 26: Advanced Supermarket Refrigeration/Heat Recovery Systems, Final Report Volume 1 – Executive Summary. Compiled by Van D. Baxter, Oak Ridge National Laboratory, April 2003.

International Journal of Refrigeration 2008a. "Thirty Years of Near Room Temperature Magnetic Cooling: Where we are today and future prospects", Gschneidner, Jr, K.A., Pecharsky, V.K., International Journal of Refrigeration, September, 2008.

International Journal of Refrigeration 2008b. "Experimental and Numerical Study on Microchannel and Round-Tube Condensers in a R410A Residential Air-Conditioning System", International Journal of Refrigeration, August 2008.

Kedzierski 2007. M.A. Kedzierski and M. Gong, "Effect of CuO Nanolubricant on R134a Pool Boiling Heat Transfer with Extensive Measurement and Analysis Details", NISTIR 7454, September 2007.

Krack 2009. 2009 Commercial Price List. Krack Corporation, Ingersoll Rand Climate Control Technologies. 2009.

Kyoto Protocol 1997. The Kyoto Protocol to the United Nations Framework Convention on Climate Change, 1997.

Lazzarin 2008. Lazzarin, R., Nardotto, D., and Noro, M. "Energy savings and economic benefits of using electronic expansion valves in supermarket display cabinets". International Journal of Low Carbon Technologies, Volume 3, Number 3, July 2008.

LIFI 2009. "Frequently Asked Questions for General Lighting", Luxim Corporation, accessed January 2009. http://www.lifi.com/dynamic/display.php/31

Nadel, S. 2002. "Packaged commercial refrigeration equipment: A briefing report for program planners and implementers", Report Number A022, American Council for an Energy-Efficient Economy, Washington, D.C.

Nadel 2004. "Increasing Appliance Energy Savings by Looking Beyond the Current Energy Star", American Council for an Energy-Efficient Economy, Energy Star Appliance Partner Meeting, 2004.

Nadel, S. 2006. "Leading the Way: Continued Opportunities for New State Appliance and Equipment Efficiency Standards", Nadel, S., deLaski, A., Eldrige, M., Kleish, J., ASAP-6/ACEEE-A062, March 2006. http://www.standardsasap.org/documents/a062.pdf

Neese 2008. Bret Neese, Baojin Chu, Sheng-Guo Lu, Yong Wang, E. Furman, and Q. M. Zhang, "Large Electrocaloric Effect in Ferroelectric Polymers Near Room Temperature", *Science.* August 2008.

Nekså . P. Nekså, J. Pettersen, and G. Skaugen. "CO_2 Refrigeration, Air Conditioning and Heat Pump Technology".

Newman 2006. Newman, J., Cariste, R., Queiruga, A. "Thermoacoustic Refrigeration". *GSET Research Journal.* 2006.

ORNL 1997. "Energy and Global Warming Impacts of HFC Refrigerants and Emerging Technologies", Oak Ridge National Laboratory, sponsored by Alternative Fluorocarbons Environmental Acceptability Study (AFEAS), U.S. Department of Energy, 1997.

ORNL 2008. D. H. Walker, R. T. Faramarzi, H. Navaz, and D. Dabiri "Air Curtain Stability and Effectiveness in Open Vertical Refrigerated Display Cases: Project Final Report", December 2008, US Department of Energy.

PepsiCo 2008. Personal Communication with PepsiCo, May 2008.

Patel-Predd 2008. Prachi Patel-Predd. "A Plastic that Chills", *Technology Review,* August 2008

Pearson . S. Forbes Pearson. "Refrigerants Past, Present and Future", International Institute of Refrigeration.

PG&E 2004. "Analysis of Standards Options for Walk-in Coolers (Refrigerators) and Freezers", Codes and Standards Enhancement Initiative For PY2004: Title 20 Standards Development, prepared for Pacific Gas and Electric Company, Davis Energy Group, May 2004.

PNL 2008. Heat Exchanger Development, Pacific Northwest National Laboratory, December 2002. http://www.pnl.gov/microcats/aboutus/capabilities/heatexchdev.html

Progressive Grocer 2008. Annual Report of the Grocery Industry, Progressive Grocer, April 2008.

Provenzano 2004. V. Provenzano, A. J. Shapiro, and R.D. Shull. "Reduction of hysteresis losses in the magnetic refrigerant $Gd_5Ge_2Si_2$ by the addition of iron", *Nature*, **429**, 853 (2004).

PSU 2009. "Frequently Asked Questions about Thermoacoustics", Penn State University Thermoacoustics Department, January 2009. http://www.acs.psu.edu/thermoacoustics/refrigeration/faqs.htm#2

Raghavan 2002. Ramesh Raghavan and Nadarajah Narendran, "Refrigerated Display Case Lighting with LEDs", Lighting Research Center (2002)

Rossing 2007. "Springer Handbook of Acoustics", Thomass D. Rossing, Springer 2007

RPI 2002. "Refrigerated Display Case Lighting with LEDs", Raghavan, R. and Narendran, N., Lighting Research Center, Rensselaer Polytechnic Institute, 2002.

SCE 2006. Scott Mitchell, Henry Lau, and Ramin Faramarzi, "Fiber Optic Lighting in Low Temperature Refrigerated Display Cases", Southern California Edison, 2006.

SCE 2009. "T-8s, LED and Fiber Optic Lighting Compared", Refrigeration & Thermal Test Center, Southern California Edison, accessed February 2009. http://www.sce.com/b-sb/design-services/RTTC/ResearchProjects/supermarket/t8_led_fiberoptic.htm

Science 2008. "Large Electrocaloric Effect in Ferroelectric Polymers Near Room Temperature", Neese, B., Chu, B., Lu, S., Wang, Y., Furman, E., and Zhang, Q., *Science*, August 2008.

Scotsman 2008. Personal Conversation with Scotsman, April 2008.

Shaws 2002. "Shaw's Supermarkets: A Comprehensive Environmental Commitment", Clean Air-Cool Planet, June 2002.

Snyder 2008. G.J. Snyder, and E.S. Toberer, "Complex Thermoelectric Materials", *Nature Materials*, **7**, 105 (2008).

Sunpower 1999. Unger, R. "Development and Testing of a Linear Compressor Sized for the European Market," *Proceedings of the International Appliance Technology Conference*, May, 1999. Sunpower, Inc., Athens, OH. http://www.sunpower.com/lib/sitefiles/pdf/publications/Doc0074.pdf

Sustainable Business 2008. "GE Submits Hydrocarbon Refrigerant for EPA Approval", SustainableBusiness.com, October 2008.

Technology Review 2008. "A Plastic that Chills", Patel-Predd P., *Technology Review*, August 2008.

Tecumseh 2008. Compressor Data, Tecumseh Compressor Company, accessed May 2008. http://202.56.127.90/nacg/pro_comp.html

TIAX 2005. Zogg, R., and Roth, K. "Assessment of Combined Heat and Power in Supermarket Applications", Final Report, prepared by TIAX, LLC for U.S. Department of Energy, Distributed Energy Program, November 2005.

Topping 2004. Dick Topping, "Carbon Dioxide Refrigerant Makes a Comeback", Refrigerant Magazine, September 2004.

True Manufacturing 2008. Product Literature, Accessed May 2008. http://www.truemfg.com/catalog/servlet/com.apsiva.servlet.HomeServlet

U.S. Census 2008. "Estimates of the Resident Population by Selected Age Groups for the United States", U.S. Census, July 1, 2008. http://www.census.gov/popest/states/asrh/SC-EST2008-01.html

Vending Times 2007. Census of the Industry, Vending Times, 2007.

Vending Times 2008a. Census of the Industry, Vending Times, 2008.

Vending Times 2008b. Personal communication between Matthew Millard of NCI and Vending Times on April 22, 2008.

WSJ 2008. "Chew This Over: Munchable Ice Sells Like Hotcakes", Brat, I., Wall Street Journal, January 30, 2008.

Zimm 1998. Zimm C, Jastrab A, Sternberg A, Pecharsky V K, Gschneidner K Jr, Osborne M and Anderson I, *Adv. Cryog. Eng*, **43** (1998)**,** 1759.

Zimm 2003. Zimm, C. 2003 Paper No K7.003, *Am. Phys. Soc. Meeting*, March 4, 2003, Austin, TX.

Image References

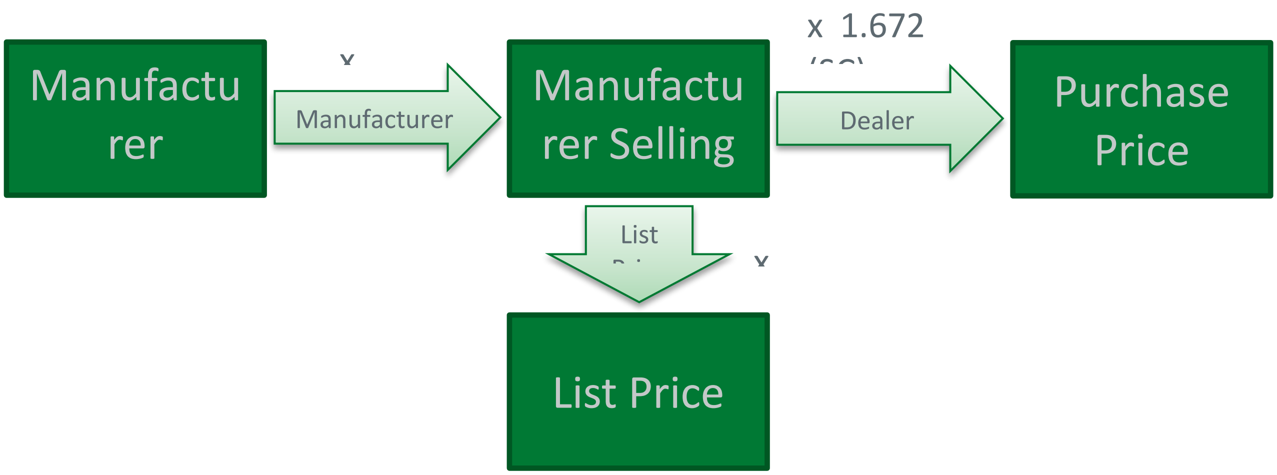
AWRCO 2008. http://www.awrco.com/
Beverage-Air 2008. http://www.beverage-air.com/
Delfield 2008.
http://www.delfield.com/?xhtml=xhtml/del/us/en/general/productcatalog.html&xsl=productcatalog.xsl&category=0151
Dixie-Narco 2008. http://www.dixienarco.com/comSite/
Electrolux 2008.
http://tools.professional.electrolux.com/Mirror/Doc/MAD/ELECTROLUX/English/FAC010.pdf
Food Service Warehouse 2008.
http://www.foodservicewarehouse.com/img/cat/qg_iceNugget.jpg
Hoshizaki America 2008. http://www.hoshizakiamerica.com/flaker.asp
Hussman 2008. http://www.hussmann.com/Supermarkets/Pages/Supermarket.aspx
Ice-O-Matic 2008.
http://www.iceomatic.com/nafemproducts.asp?poc=ice&category=0142&article=ICE_1405_Modular_Cube_ICE_Maker.xml
McCall 2008. McCall Refrigeration,
http://www.4mccall.com/Store/OnlineCatalog.asp#ss_photos
Nor-Lake 2008. http://www.norlake.com/
Sanden Vendo 2008. http://www.vendoco.com/index.php
Traulsen 2008. http://www.traulsen.com/products/refrigeration/refrigeration_refrigerators-freezers/refrigerators-freezers_dual-temps
Tyler 2008. http://www.tylerrefrigeration.com/
Zero Zone 2009. http://www.zero-zone.com/default.asp (from Emerson Climate Technologies PowerPoint called, “High Efficiency Scroll for Walk-in Refrigeration”)

Appendix A Explanation of Markups and Inflation Factors

In this analysis, we report on various costs associated with CRE including the total installed cost for the baseline equipment and the end-user cost premium for the high efficiency technology options. This appendix discusses the markups and inflation rates we assumed when calculating such prices.

Markups

We assumed three markups throughout the report, based on the information in the DOE CRE Rulemaking Final Rule (DOE 2009a). Figure A-1 shows the relationships between the various cost types and the markups used in this report.

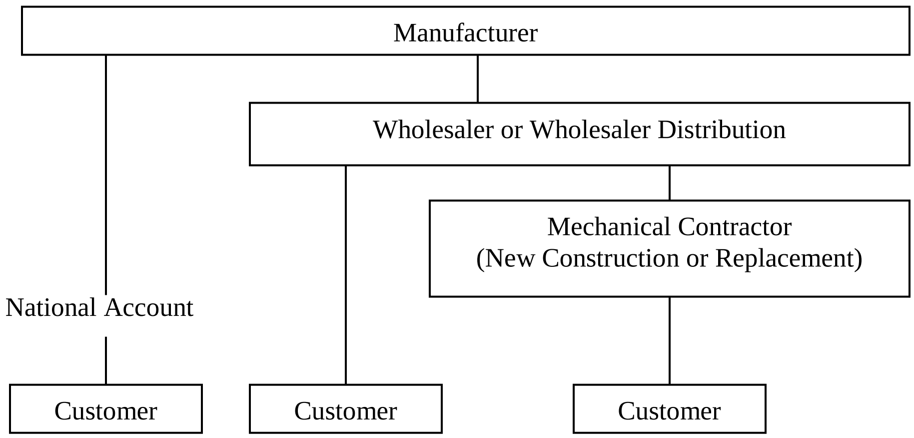


Source: DOE 2009a

Figure A-0-1: Markup Relationships

The manufacturer markup of 1.32 is used to convert manufacturer production cost (MPC) to manufacturer selling price (MSP).

The dealer markup includes both the markup applied by the dealer and a weighted average sales tax multiplier to convert MSP to purchase price. The dealer markup differs for remote-condensing (RC) and self-contained (SC) equipment because of the differing breakdowns by distribution channel as shown in Figure A-2. In the first distribution channel, the manufacturer sells the equipment directly to the customer through a national account. In the second and third distribution channel, the manufacturer sells the equipment to a wholesaler, who in turn may sell it directly to the customer or through a mechanical contractor. The wholesaler in this case can be a refrigeration wholesaler focusing on refrigeration equipment, or a grocery warehouser (supply chain distributor) who sells food and store equipment to the retailer.



Source: DOE 2009a

Figure A-0-2: Distribution Channels for Refrigerated Display Cases

Table A-1 gives the distribution channel shares (in percentage of total sales) through each of the three distribution channels for remote-condensing and self-contained equipment. The distribution channel shares are based on estimates provided by a major manufacturer to DOE during the DOE energy conservation standards rulemaking for commercial refrigeration equipment.

Table A-0-1: Distribution Channel Shares for New and Replacement Construction

	National Account	Distributor	Contractor
Remote Condensing Equipment	70%	15%	15%
Self-Contained Equipment	30%	35%	35%
Source: DOE 2009a			

The list price is typically greater than the purchase price. We assumed a list price markup of 2 between the MSP and the list price in cases where we used list prices to estimate purchase prices in chapter 3.

Inflation

All cost data in this report is updated to 2008 dollars, either by direct reference to a 2008 source or by adaptation of an older source using an assumed inflation rate. To bring older cost data up to 2008 dollars, we used the producer price index (PPI) for the industry segment titled "commercial refrigerators and related equipment" (NAICS code 3334153). The PPI gives an indication of how prices have changed over time for specific sectors of the economy. The raw data from the PPI is shown in table A-2. The rightmost column is an adjusted version of the raw data to show inflation relative to 1996, the year in which the previous Arthur D. Little report on CRE energy savings potential (ADL 1996) was published. When necessary, we updated costs from that report using a multiplier of 1.254 as indicated in the table.

Table A-0-2: Producer Price Index for CRE

Year	Inflation relative to 1982	Inflation relative to 1996 (calculated)
1996	1.505	1.000
1997	1.514	1.006
1998	1.546	1.027
1999	1.584	1.052
2000	1.608	1.068
2001	1.636	1.087
2002	1.626	1.080
2003	1.633	1.085
2004	1.659	1.102
2005	1.713	1.138
2006	1.744	1.159
2007	1.822	1.211
2008	1.887	**1.254**
Source: PPI 2009		

Appendix B Electricity Rate Data

The operating cost data used to calculate payback periods for each of the energy-saving technologies is based on the electricity data presented in this appendix. For an explanation of how the three scenarios were calculated refer to section 4.

State [1]	Price 2008 ($/kWh)	Population 2008	Cumulative Population	% Total Population
High Rate States				
HI	0.2972	1,288,198	1,288,198	0.4%
NY	0.1645	19,490,297	20,778,495	6.8%
MA	0.1605	6,497,967	27,276,462	9.0%
CT	0.1593	3,501,252	30,777,714	10.1%
RI	0.1549	1,050,788	31,828,502	10.5%
NJ	0.1472	8,682,661	40,511,163	13.3%
NH	0.1428	1,315,809	41,826,972	13.8%
DC	0.1364	591,833	42,418,805	14.0%
AK	0.1333	686,293	43,105,098	14.2%
CA	0.1305	36,756,666	79,861,764	26.3%
Medium Rate States				
ME	0.1295	1,316,456	81,178,220	26.7%
MD	0.1282	5,633,597	86,811,817	28.6%
VT	0.1251	621,270	87,433,087	28.8%
DE	0.1204	873,092	88,306,179	29.0%
TX	0.1065	24,326,974	112,633,153	37.0%
FL	0.102	18,328,340	130,961,493	43.1%
NV	0.1015	2,600,167	133,561,660	43.9%
LA	0.101	4,410,796	137,972,456	45.4%
MS	0.0998	2,938,618	140,911,074	46.3%
AL	0.0985	4,661,900	145,572,974	47.9%
MI	0.0943	10,003,422	155,576,396	51.2%
PA	0.0942	12,448,279	168,024,675	55.3%
WI	0.0928	5,627,967	173,652,642	57.1%
OH	0.0926	11,485,910	185,138,552	60.9%
GA	0.0919	9,685,744	194,824,296	64.1%
TN	0.0902	6,214,888	201,039,184	66.1%
AZ	0.0886	2,855,390	203,894,574	67.1%
NM	0.0856	1,984,356	205,878,930	67.7%
CO	0.0856	4,939,456	210,818,386	69.3%
MT	0.0854	967,440	211,785,826	69.7%
IL	0.0853	12,901,563	224,687,389	73.9%
SC	0.0853	4,479,800	229,167,189	75.4%

Low Rate States				
OK	0.0804	3,642,361	232,809,550	76.6%
MN	0.0786	5,220,393	238,029,943	78.3%
IN	0.0781	6,376,792	244,406,735	80.4%
AR	0.0776	6,500,180	250,906,915	82.5%
NC	0.0767	9,222,414	260,129,329	85.6%
OR	0.0759	3,790,060	263,919,389	86.8%
KS	0.0755	2,802,134	266,721,523	87.7%
VA	0.0738	7,769,089	274,490,612	90.3%
IA	0.0728	3,002,555	277,493,167	91.3%
KY	0.0725	4,269,245	281,762,412	92.7%
SD	0.0683	804,194	282,566,606	92.9%
ND	0.068	641,481	283,208,087	93.1%
WA	0.0679	6,549,224	289,757,311	95.3%
UT	0.0673	2,736,424	292,493,735	96.2%
WY	0.067	532,668	293,026,403	96.4%
NE	0.0664	1,783,432	294,809,835	97.0%
MO	0.0659	5,911,605	300,721,440	98.9%
WV	0.0606	1,814,468	302,535,908	99.5%
ID	0.0572	1,523,816	304,059,724	100.0%
Total		304,059,724		
Sources: EIA 2009 for electricity prices, U.S. Census 2008 for U.S. population. [1] Washington, DC, is included as a state.				

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