Energy Savings Potential for Commercial Refrigeration Equipment

Final Report
Prepared by
Arthur D. Little, Inc.

For
Building Equipment Division
Office of Building Technologies
U.S. Department of Energy

June 1996

Prepared by:
Detlef Westphalen
Robert A. Zogg
Anthony F. Varone
Matthew A. Foran

DOE Program Managers:
Dr. James R. Brodrick
Terry DuBois

Arthur D. Little, Inc.
Acorn Park
Cambridge, Massachusetts
02140-2390

Reference 46230-00

Table of Contents

Table of Contents (continued)

List of Figures

1. Executive Summary

Primary energy usage in the commercial refrigeration sector is estimated as 990 trillion Btu.[1] The contribution to this sum of the different commercial refrigeration sectors is shown in Figure 1-1 below.

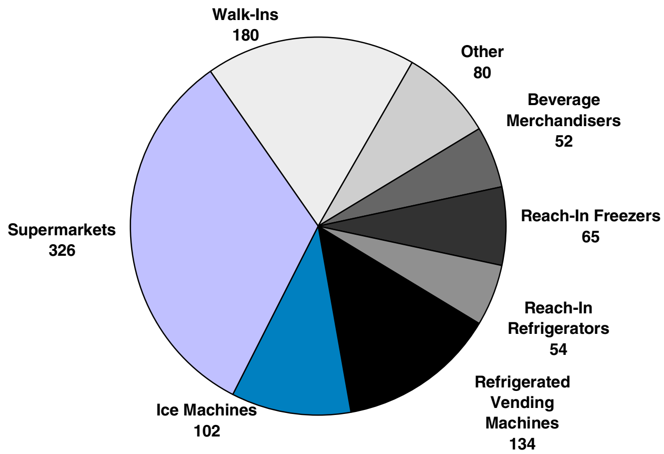


Figure 1-1: Primary Energy Usage in Commercial Refrigeration (trillion Btu)

Note: "Others" includes Roll-Ins, Under-counter, non-beverage self-contained merchandisers and display cases, and single-compressor remote systems serving display cases in small grocery applications.

The analyses described in this report indicate that there are large opportunities for savings in the commercial refrigeration sector. Primary energy savings of about 266 trillion Btu (29% for the equipment types examined) were identified based on improvements in current technology assuming implementation of the most economically attractive technologies for all equipment in the installed based. The split of these savings amongst the studied equipment types is shown in Figure 1-2 below.

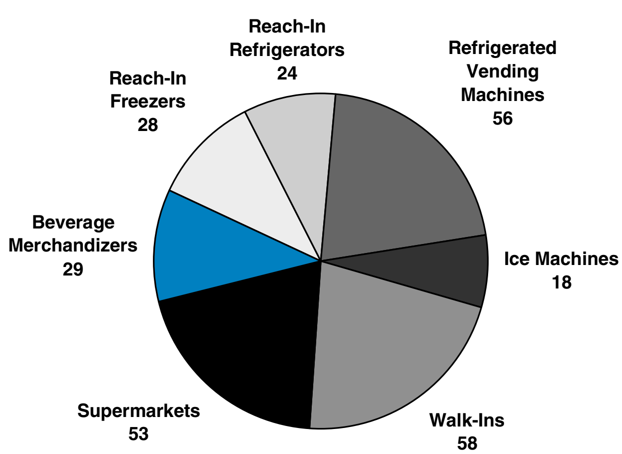


Figure 1-2: Commercial Refrigeration Savings Potential (trillion Btu)

[1] Primary energy use is calculated based on a heat rate of 10,867 Btu/kWh, which takes into account power production, distribution, and transmission losses.

Much of this savings potential is associated with high efficiency fan motors and high efficiency compressors, technologies with paybacks of less than 2 years when used for new equipment. High efficiency fan motors can also be implemented on a retrofit basis (with longer payback if the existing motors are still functional).

Additional savings are associated with hot gas defrost, use of hot gas or liquid for antisweat heating, and defrost control. Implementation of these technologies is projected to have payback periods typically within 5 years.

Table 1-1 below shows the identified energy savings distributed among technologies.

Table 1-1: Identified Energy Savings Potential

	Savings Potential (Trillion Btu)	Payback Range (Years)
Evaporator Fan ECM Motor	85	0.5 - 3
ECM/Variable Speed Compressor	48	2 - 5
High-Efficiency Compressors	39	0.5 - 2
High-Efficiency Fan Blades	30	0.1 - 1
Condenser Fan ECM Motor	25	0.5 - 8
Floating Head Pressure	25	0.3 - 3
Electronic Ballasts	24	1 - 2.5
Non-Electric Antisweat	20	1 - 1.5
Thicker Insulation	20	1 - 1.5
Ambient Subcooling	12	2 - 11
Hot Gas Defrost	10	1.5 - 3
Liquid-Suction Heat Exchangers	10	4 - 14
Evaporative Condensers	10	*
Antisweat Heater Controls	10	2 - 6
Other Ice Machine Process Improvements	9	1 - 6
Evaporator Fan Shutdown	7	1 - 2
External Heat Rejection	6	7
Economizer Cooling	6	20
Heat Reclaim	3	2 - 5
Defrost Control	3	3
Mechanical Subcooling	2	5

*No payback in most locations due to non-energy costs

The use of alternate refrigeration cycles which are not now commonly used in refrigeration equipment does not, at this point, appear to have significant potential for reducing primary energy use. Examined alternative cycles include absorption and chemisorption. The use of waste heat to drive these cycles does have the potential to reduce primary energy usage, however.

The phaseout of CFC refrigerants and the future phaseout of HCFC refrigerants requires that equipment be redesigned and that much existing refrigeration equipment be replaced. This provides an opportunity to accelerate the implementation of efficiency-improving technologies. However, the theoretical efficiency of possible alternative refrigerants is not significantly higher than that of currently-used refrigerants. The supporting analysis takes into account the secondary refrigerant loop which would be required for implementation of ammonia and flammable refrigerants in commercial refrigeration.

The use of gas-fired equipment will not result in significant primary energy savings. Table 1-2 below shows the comparison of electric and gas technologies for low temperature supermarket applications. Gas-fired technologies are more economical in some areas with high-priced electricity, and they provide advantages in peak load reduction, which have associated infrastructure cost benefits.

Table 1-2: Comparison of Electric and Gas Refrigeration Options

	Electric Use[1] (kW/ton)	Equipment Gas Use (mBtu per ton-hr)	Primary Energy Use[2] (mBtu/ton-hr)	Primary Energy COP
Electric	2.3	0	25	0.48
Gas-Engine	0.25	22	25	0.48
Chemisorption[4]	0.56	42[3]	48	0.25
Advanced Absorption	0.46	21[3]	26	0.46

Supermarket Low Temperature: -20°F; 110°F Condenser
[1] Includes parasitics for heat rejection, burner fan, solution pumps
[2]Calculated based on 10,867 Btu/kWh and zero distribution losses for gas
[3]Assuming 82% burner efficiency.
[4]Assuming use of a single-stage cycle. Two-stage cycles may have primary energy COP's ranging from 0.4 to 0.46.

1.1 Supermarket Refrigeration

Supermarket refrigeration accounts for 326 trillion Btu of primary energy usage annually.

Supermarket refrigeration is divided into two distinct segments which have different technology and which are governed by different issues. The more visible part of these systems are the display cases which hold food for the self-service shopping style of supermarkets. The display cases have their own electric loads, and they must be cooled by the store's refrigeration system. Display case selection is merchandising-based. The mechanical equipment, including compressors, condensers, and associated controls, is engineering-based.

The potential for energy consumption reductions associated with machine room equipment is limited to about 5% of overall supermarket refrigeration energy usage (see Table 1-3 corresponds to about 13 trillion Btu. The limited savings opportunities reflect the sensitivity to energy efficiency in machine room equipment selection decisions.

Reduction of 1% of overall usage with a two year payback[2] is possible with increased use of evaporative condensers, a technology which currently has little market penetration. This technology should not have a cost premium with respect to air-cooled condensers, but the savings will be overshadowed by water and maintenance costs for about two thirds of US locations. Additional reductions of 2.5% with paybacks of less than five years could be achieved by further use of floating head pressure, mechanical subcooling, and heat reclaim, technologies which currently have varying degrees of market penetration.

Table 1-3: Supermarket Energy Savings: Machine Room Technologies

	Energy Savings			Economics 45,000 sq. ft. Supermarket (New Construction) Medium Energy Costs $0.053/kWh; $5.04/kW; $5.60/MMBtu Gas		
	Refrigeration Electricity Savings (%)	Store Electricity Savings (%)	U.S. Primary Energy Savings (10^{12}Btu)	Cost Premium ($)	Savings ($)	Payback Period (years) (Raleigh, NC)
Evap. Condenser	3.1	1.5	10	($7,100)	($560)	N/A
Floating Head Pressure	3.1	1.5	4	$8,000	$3,200	2.5
Heat Reclaim	N/A	N/A	3	$13,700	$5,400	2.5
Mechanical Subcooling	1.4	0.7	2	$8,000	$1,600	4.9
Ambient Subcooling	0.5	0.25	1	$6,100	$560	11
Total < 5 year payback	4.5	2	9	$30,000	$10,000	3

Energy savings potential in the display case area are summarized in Table 1-4 below. Savings of 14% (45 trillion Btu[3]) of total supermarket refrigeration primary energy use are possible with improvements in this area. Savings of 9% can be achieved with less than 2 year payback with high-efficiency evaporator fan motors and hot gas defrost. Additional savings of 4% can be achieved with less than 5 year payback with liquid-suction heat exchangers, antisweat control, and defrost control. Insulation thickness increases are ineffective because wall losses are not the dominant case load and because of the associated volume decrease.

---

[2] Payback periods presented in this report are based on new construction or purchase of new products. For retrofit, the payback period will generally be longer.

[3] Savings potential assuming 100% penetration in existing base. The savings potentials reported in this section are based on this assumption.

Table 1-4: Supermarket Energy Savings: Display Case Technologies

	Energy Savings			Economics 45,000 sq. ft. Supermarket (New Construction) Medium Energy Costs $0.053/kWh; $5.04/kW; $5.60/MMBtu Gas		
	Refrigeration Electricity Savings (%)	Store Electricity Savings (%)*	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium	Savings ($)	Payback Period (years)
Hot Gas Defrost	3.1	1.5	3	$3,800	$2,600	1.4
Antisweat Ht. Control	5.7	2.8	5	$7,500	$4,800	1.6
Evap. Fan ECM Motor	8.2	4.1	26	$12,600	$7,700	1.6
Defrost Control	1.3	0.6	2	$3,300	$1,100	3.0
LSHX* Low Temp.	2.4	1.2	4	$10,000	$2,400	4.1
LSHX* Med. Temp.	1.8	0.9	4	$25,000	$1,800	14
Insulation Improvement	0.3	0.15	1	$11,000	$315	35
Total < 2 years payback	17	8.5	35	$24,000	$15,100	1.6
Total < 5 year payback	21	10	40	$37,000	$18,600	2.0

*LSHX = Liquid Suction Heat Exchangers

Reference No.

1.2 Beverage Merchandisers

Primary energy usage reductions of about 45%, representing 17 trillion Btu, are possible with beverage merchandisers (see Table 1-5 below). Reductions of 41% are possible within a two-year payback with the use of ECM motors for evaporator fans and high efficiency compressors. Additional reductions of 4% can be achieved within a five year payback with ECM motors for condenser fans. Long paybacks are associated with increased R-value insulation or increased insulation thickness.
Supermarket Energy Savings: Display Case Technologies

Table 1-5: Beverage Merchandiser Energy Savings

	Energy Savings		Economics Medium Energy Costs $0.0782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium	Savings ($)	Payback Period (years)
High-Efficiency Compressors	9	5	$16	$26	0.6
Electronic Ballasts	10	5	$30	$30	1.0
Evap. Fan ECM	29	15	$120	$85	1.4
ECM/Var. Spd. Compressors	14	7	$150	$42	3.6
Cond Fan ECM M otor	4.5	2	$60	$14	4.4
Thicker Insulation*	3.0	2	$56	$9	6.2
Total < 2 years payback	44	23	$166	$134	1.2
Total < 5 year payback (with ECM/Var. Spd. Compressor)	55	29	$376	$168	2.2

*Increase from 1 1/2” to 2 1/2”

1.3 Reach-In Freezers

Primary energy usage reductions of about 40%, representing 29 trillion Btu, are possible with reach-in freezers (see Table 1-6 below). Reductions of 30% are possible within a two-year payback with the use of high efficiency compressors and non-electric antisweat heating. Additional reductions of 10% can be achieved within a five year payback with ECM motors for evaporator fans, hot gas defrost, and defrost controls. Long paybacks are associated with increased R-value insulation or increased insulation thickness. Additional energy reductions with impractical payback periods could be achieved with the use of liquid-suction heat exchangers.

Table 1-6: Reach-In Freezer Energy Savings

	Energy Savings		Economics Medium Energy Costs $0.782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium ($)	Savings ($)	Payback Period (years)
High-Efficiency Compressors	16	10	$24	$65	0.4
Non-Electric Antisweat	14	9	$67	$58	1.2
ECM/Var. Spd. Compressors	19	12	$160	$77	2.1
Cond. Fan ECM Motor	2.7	2	$24	$11	2.2
Evap. Fan ECM Motor	2.3	1.5	$24	$9	2.6
Hot Gas Defrost	6.3	4	$83	$26	3.2
Thicker insulation**	3.8	2.5	$84	$15	5.5
LSHX*	3.4	2	$75	$14	5.5
Total < 2 years payback	30	20	$91	$123	0.7
Total < 5 year payback (with ECM/Var. Spd. Compressor)	44	28	$382	$178	2.1

*Liquid-Suction Heat-Exchanger
**Increase from 2 1/4” to 3 1/4”

1.4 Reach-In Refrigerators

Primary energy usage reductions of about 50%, representing 27 trillion Btu, are possible with reach-in refrigerators (see Table 1-7 below). Reductions of 47% are possible within a two-year payback with the use of ECM motors for evaporator fans, high efficiency compressors, and non-electric antisweat heating. Additional reductions of 3% can be achieved within a five year payback with ECM motors for condenser fans. Long paybacks are associated with increased R-value insulation or increased insulation thickness.

Table 1-7: Reach-In Refrigerator Energy Savings

	Energy Savings		Economics Medium Energy Costs $0.782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium ($)	Savings ($)	Payback Period (years)
High-Efficiency Compressors	12	6	$16	$40	0.4
Non-Electric Antisweat	20	11	$93	$67	1.4
Cond Fan ECM Motor	3.3	2	$22	$11	2.0
Evap. Fan ECM Motor	7	4	$48	$23	2.1
ECM/Var. Speed Compressor	16	9	$150	$54	2.8
Thicker insulation**	2	1	$100	$8	13
Total < 2 years payback	35	19	$131	$118	1.1
Total < 5 year payback (with ECM/Var. Spd. Compressor)	45	24	$313	$152	2.1

*Increase from 2 1/4” to 3 1/4”

1.5 Ice Machines

Primary energy usage reductions of about 20%, representing 13 trillion Btu, are possible with ice machines (see Table 1-8 below). Reductions of 15% are possible within a two-year payback with the use of ECM motors for condenser fans, high efficiency compressors, and reduced evaporator thermal cycling. Additional reductions of 5% can be achieved within a five year payback with thicker insulation and mechanical harvest assist to reduce ice meltage during harvest.

Table 1-8: Ice Machine Energy Savings

	Energy Savings		Economics Medium Energy Costs $0.0782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium ($)	Savings ($)	Payback Period (years)
Reduced Evap. Thermal Cycling	4.2	4	$20	$16	1.2
High-Efficiency Compressors	5.6	6	$40	$22	1.8
Cond Fan ECM Motor	5.4	6	$46	$21	2.2
Thicker insulation**	3.0	3	$40	$12	3.4
Reduced Meltage During Harvest	4.6	5	$100	$18	5.6
Total < 2 years payback	10	10	$60	$38	1.6
Total < 5 year payback (with ECM/Var. Spd. Compressor)	18	18	$146	$71	2.1

*Increase from 1/2” to 1”

1.6 Refrigerated Vending Machines

Primary energy usage reductions of about 32%, representing 45 trillion Btu, are possible with refrigerated vending machines (see Table 1-9 below). Reductions of 30% are possible within a two-year payback with the use of ECM motors for evaporator fans, and high efficiency compressors. Additional reductions of 1% can be achieved within a five year payback with high efficiency condenser fan motors.

Table 1-9: Refrigerated Vending Machine Energy Savings

	Energy Savings		Economics Medium Energy Costs $0.0782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium ($)	Savings ($)	Payback Period (years)
High-Efficiency Compressors	9	12	16	$20	0.8
Electronic Ballasts	9	12	$30	$20	1.5
Evap. Fan ECM Motor	14	19	$56	$31	1.8
Improved Insulation	5.4	7	$54	$12	4.5
ECM/Var Speed Compressors	15	20	$150	$32	4.6
Cond Fan ECM Motor	3	4	$56	$7	8
Total < 2 years payback	32	43	$102	$70	1.5
Total < 5 year payback	42	56	$290	$91	3.2

2. Introduction

In the commercial sector, energy conservation programs currently put an emphasis on lighting and HVAC equipment, since this equipment accounts for 65% of 1993 primary energy use in the commercial sector. Significant energy savings may also be achieved for other commercial end-uses. The majority of the non-HVAC/lighting primary energy use is from office equipment, water heating, and refrigeration. Commercial refrigeration equipment represents about 20% of this load (see Figure 2-1 below). This study evaluates the energy savings potential for commercial refrigeration equipment. The baseline usage estimates are compared with previous estimates in Section 4.

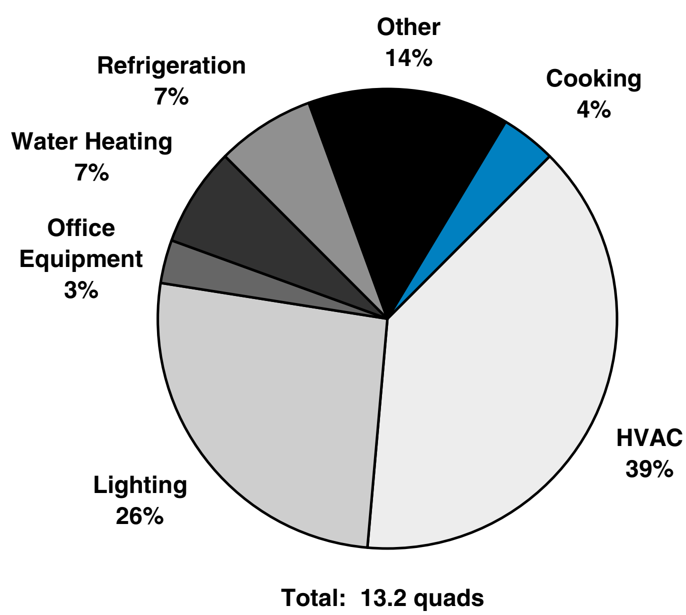


Figure 2-1: Commercial Sector Primary Energy Usage - 1993

Source: EIA, "Energy Outlook 1995;" ADL., "Characterization of Commercial Appliances," 1993; ADL estimates

The purpose of this study was:

- To select several types of self-contained and engineered equipment to represent the general categories of commercial refrigeration equipment;
- To identify the physical characteristics and typical energy consumption for each equipment type (broken down by component);
- To identify typical costs, life and reliability characteristics, and major manufacturers and end-users for each equipment type;

- To estimate the first cost premiums and simple payback periods resulting from the use of various energy-saving technologies for each equipment type;

- To identify barriers to the use of each energy-saving technology investigated; and

- To identify programmatic options to stimulate the use of those technologies that appear most attractive.

Section 3 of this report describes the scope of work for this project. The general approach to the study is discussed, outlining the various equipment types reviewed and energy saving technologies considered.

Section 4 contains the prototypical descriptions of each type of commercial refrigeration equipment considered in this study. For each equipment type, a general overview of the equipment and its current commercial sector energy use is followed by a detailed description of the prototypical equipment. This description includes:

- physical characteristics and illustrations
- refrigeration component characteristics
- refrigeration loads and case loads
- energy consumption breakdown

The prototypical equipment description is followed by a discussion on equipment life, reliability and maintenance characteristics. Major manufacturers and end-users are also identified.

Section 5 reviews the energy saving technologies for each equipment type. A brief description of each technology is followed by an economic analysis with estimated cost premiums and simple paybacks. Possible barriers to the implementation of these technologies are identified in addition to recommendations for DOE to stimulate the use of technologies which would have the most impact.

3. Scope of Work

Commercial refrigeration includes the following equipment and system types:

- Supermarket refrigeration systems, consisting of display cases and walk-in refrigerators and freezers utilizing remote parallel compressors and condensing equipment;
- Self-contained systems (such as upright and horizontal merchandisers, beverage merchandisers, deli cases, reach-in and roll-in refrigerator/freezers, and undercounter refrigerator/freezers);
- Walk-in coolers and freezers;
- Refrigerated vending machines;
- Water coolers; and
- Ice machines.

Six specific equipment types were examined in detail to determine energy savings potential for commercial refrigeration. These six equipment types represent the general categories of commercial refrigeration. Prototypical descriptions for each of the following equipment types were developed:

- Supermarket refrigeration system (including both low and medium temperature);
- Beverage Merchandiser (single glass door);
- Reach-in freezer (single door);
- Reach-in refrigerator (two-door);
- Ice machine;
- Canned beverage vending machine; and
- Walk-in freezers and coolers

Water coolers are not covered in the report because they are small systems which do not represent large aggregate energy usage.

Each prototypical equipment type was characterized by:

- Efficiency, power consumption (broken down by component), and duty cycle;
- Illustration showing typical overall dimensions and layout, including location of major components;
- Typical manufacturing, purchase, and installation costs;
- Life and reliability characteristics; and
- Identification of major manufacturers and end users.

After prototypical equipment descriptions were established, energy savings potentials, first cost premiums, and payback periods were estimated for various improved-efficiency technologies. Estimating first-cost premiums required estimating the incremental manufacturing cost for incorporating the improved-efficiency technology.

Examined technologies have been classified according to their current status as follows:

- ***Current technologies*** that are available in the marketplace but may not be in widespread use due to a variety of cost/or industry structure reasons
- ***New technologies*** that are available but not yet utilized in commercially available equipment
- ***Advanced technologies*** that need research and development to establish technical and commercial viability.

The examined technologies are listed below.

For remotely located engineered equipment (supermarket), the technology options examined were:

Current Technologies:
- Head-pressure control;
- Evaporatively cooled condensers;
- Mechanical subcooling;
- Ambient Subcooling
- Heat Reclaim
- Hot Gas Defrost
- Antisweat control
- Liquid - Suction Heat Exchangers

New Technologies:
- High-Efficiency fan motors
- Insulation improvements
- Electronic Ballasts
- Engine-driven refrigeration

Advanced Technologies:
- Alternative refrigerants (propane or ammonia)
- Absorption refrigeration
- Demand defrost control

For self-contained equipment, the list of examined technology options is:

New Technologies:
- High-efficiency compressors;
- Improved insulation;
- Hot gas defrost;
- Liquid-suction heat exchangers;

- Ice machine process improvements.
- High-efficiency fan blades
- Electronic Ballasts
- High-efficiency motors (for fans and compressors)

Advanced Technologies:
- Variable speed compressors
- Non-electric antisweat heating
- Demand defrost control

We have identified barriers (technical, market, and institutional) to the use of each technology, such as safety concerns, reliability, R&D costs, manufacturing facility limitations, customer acceptance issues, installation requirements, emissions requirements, and service requirements.

We have formulated recommendations for various programmatic options to DOE designed to promote the increased use of high-efficiency commercial refrigeration equipment.

4. Prototypical Descriptions of Baseline Equipment

Primary energy usage in the commercial refrigeration sector is estimated as 990 trillion Btu. This estimate is based on the inventory and energy usage data compiled during this study. The estimates of this study are compared with previous ADL estimates (Characterization of Commercial Building Appliances, for DOE, June 1993) in Table 4-1 below.

Table 4-1: Estimates of Commercial Refrigeration Inventories and Energy Usage

Commercial Appliance Report (June 1993)			This Report		
Equipment Type	**Equipment Inventory (1000's)**	**Total Primary Energy* (TBtu)**	**Total Primary Energy** (TBtu)**	**Equipment Inventory (1000s)**	**Equipment Type**
Ice Makers	1200	105	102	1200	Ice Makers
Supermarkets	187[1]	320	326	30	Supermarkets
Centralized Systems	1186	170	206	900	Total
(excluding supermarkets)			180	880	Walk-Ins
			26	20	Small Grocery[3]
Vending Machines	2270	72	134	4100	Vending machines
Self-Contained	2099	149	225	4050	Total
			54	1300	Reach-In Refrigerators
			65	800	Reach-In Freezers
			52	800	Beverage Merchandisers
			54	1150	Other[2]
Total		**744**	**993**		**Total**

* Based on 11,200 Btu/kWh
**Based on 10,867 Btu/kWh

1 Assuming 30,000 buildings, 5 systems per building, and adjustment for total shipments
2 Roll-Ins, Under-counter, Over-Counter, Non-beverage Merchandisers
3 These systems consist of supermarket-style display cases with single remotely located condensing units (compressor configuration is not parallel, as in supermarkets).

Estimates of the two studies for inventories and energy usage of most of the equipment types are fairly consistent. The exceptions are self-contained equipment and vending machines, for which inventory estimates of this study are higher. The current estimates are based on a more detailed examination of available data, and are based on information of trade and industry representatives as well as raw shipment data. The use of equipment classifications more consistent with industry standards has also improved the estimates. The overall sector energy usage estimate is about 30% higher.

4.1 Supermarket Refrigeration Systems

There are approximately 30,000 supermarkets in the USA (Progressive Grocer, April 1995, p.9). Food sales from supermarkets represents roughly $300 billion in retail value, or 75% of the overall food sales market (Progressive Grocer, April 1995, p.9).

Supermarkets are distinguished by Progressive Grocer Magazine from smaller grocery stores primarily by their revenue, which according to their definition exceeds two million dollars per year.

Table 4-2 shows recent trends in numbers and per store revenues for the food sales sector. The trend for supermarkets has been towards smaller numbers of larger stores which sell more food. The convenience store category represents establishments which have less than one million dollars annual revenue.

Table 4-2: Store Numbers and Average Sales in the Food Sales Sector

	Supermarkets		Small Grocery		Convenience Stores	
	Number of Stores (1,000's)	Annual Sales per Store ($ million)	Number of Stores (1000's)	Sales per Store ($ Million)	Number of Stores (1000's)	Sales per Store ($ Million)
1990	30.7	8.84	11.6	1.88	102.7	0.508
1991	30.7	9.14	12.1	1.83	97.2	0.541
1992	30.4	9.42	12.7	1.84	94.9	0.565
1993	29.8	9.80	13.3	1.74	95.1	0.592
1994	29.7	10.1	13.9	N/A	87.4	N/A

Source: Progressive Grocer, Annual Reports of the Grocery Industry, April 1991-1995
Supermarket Business, Consumer Expenditures Studies, September 1990-1994

Supermarket complexity has increased, as supermarkets have added 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; they currently represent one of the largest growing food service sectors.

4.1.1 Equipment Description and Illustrations

General Description

The purpose of the refrigeration systems of supermarkets is for food storage and for display of food for self-service sales. Food is stored prior to transfer to the store area in walk-in storage areas.

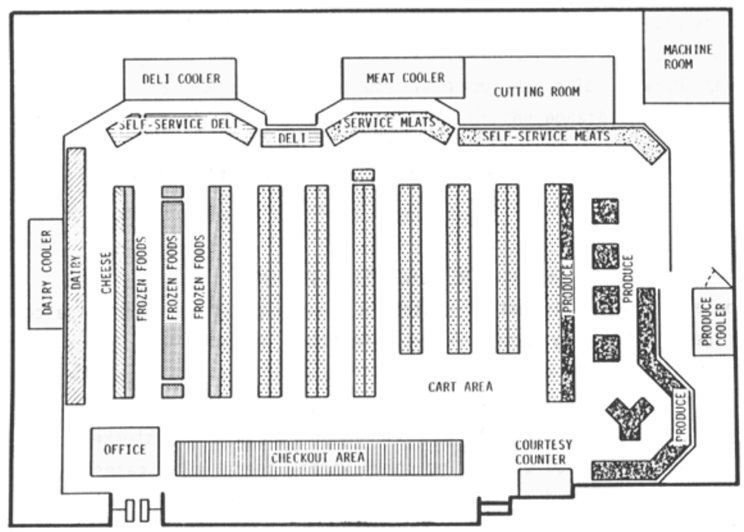


Figure 2-1. Layout of a Typical Supermarket

Figure 4-1 shows a typical arrangement of display cases in a supermarket. The cases are generally located at the periphery of the store near their associated walk-ins. Display cases of a variety of configurations are used in the sales area.

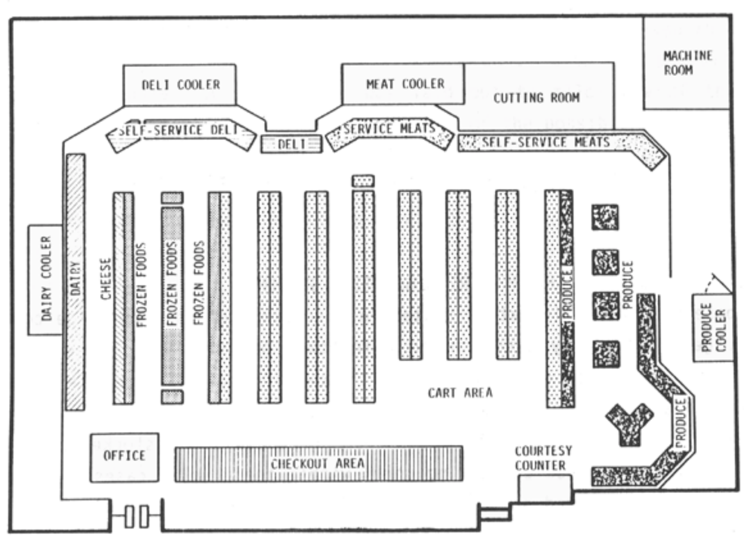


Figure 2-1. Layout of a Typical Supermarket

Figure 4-1: Typical Supermarket Layout (Foster Miller, 1990)

Storage temperature depends on the product. Classification of temperatures is primarily low and medium. Additional temperature ranges are very low and high. The evaporator temperature ranges and their associated applications are as follows:

- 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

Display Cases

The purpose of supermarket cases is to display food for the self-service style of supermarket shopping. Hence they are primarily evaluated considering two criteria: food preservation and sales enhancement. Both glass-door and open display cases are used. The most common case types are:

- Glass-Door Reach-Ins
- Open Multi-Deck
- Coffin/Open-Tub Freezers (Single-Level)
- Seafood/Deli Display Case

A typical supermarket will have from 60 to 80 or more display cases. About half of these will be low temperature or very low temperature cases. Figure 4-2, Figure 4-3, Figure 4-4, and Figure 4-5 show some of the most common supermarket cases.

The case contains an expansion valve and one or more evaporators for case cooling. Evaporator fans circulate case air. The air flow in open cases is blown over the open section of the case, creating an air curtain which separates food from the warmer store air. Multiple fans are required for most cases. Low temperature evaporators and some medium temperature evaporators require periodic defrosting to remove frost which condenses and/or freezes on the evaporator surface. This can be done with electric defrost, or hot gas defrost. The former involves electric resistive heating with a defrost coil which is integrated into the evaporator coil. Hot gas defrost involves piping and valves which send hot gas from the compressor discharge into the evaporator. Some medium-temperature cases can also use off-cycle defrost. More discussion on defrost appears in Section 5.1.1.

The case insulation is typically 1-1/2 to 2 inches thick, with insulating valves from R-11 to R-15. Glass doors are typically fitted with at least two panes. Anti-sweat heaters and glass heaters are used to prevent condensation formation. Today's more efficient doors have three glass layers enclosing insulating gases and don't require glass heaters. Cases are generally fitted with lighting to illuminate the products. High pressure liquid and suction refrigerant piping must be connected to the case. Additional connections are electrical power and condensate drain lines.

Walk-Ins

In addition to the display cases, the supermarket refrigeration system has walk-in areas for temporary storage of delivered products. The low temperature variety are called walk-in freezers, and the medium temperature variety are called walk-in coolers. Walk-ins and display cases serving the same products are generally located in close proximity. Walk-ins have 3 or more inches of insulation and good door sealing systems — they typically require no anti-sweat heaters.

Figure 4-2: Single-Level Wide Aisle Open Display Case

Source: Zero-Zone Refrigerator Manufacturing Company, Inc.

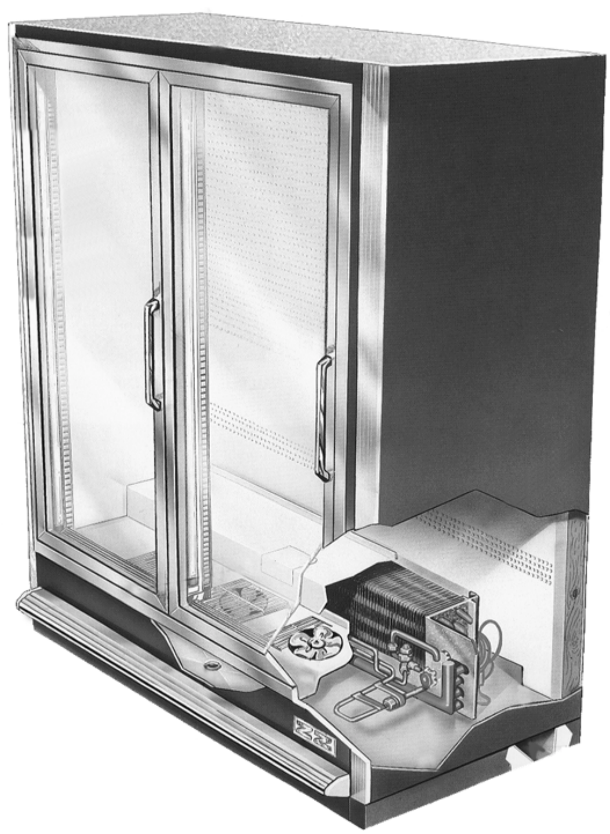


Figure 4-3: Glass Door Reach-In Display Case

Source: Zero-Zone Refrigerator Manufacturing Company, Inc.

Figure 4-4: Multi-Level Open Dairy Case

Source: Kysor-Warren

Figure 4-5: Multi-Level Open Meat Case

Source: Kysor-Warren

Refrigeration Equipment

The heart of the supermarket refrigeration system are the compressors. In most modern supermarkets, the compressors are configured as compressor racks, which consist of a number of parallel-connected compressors located in a separate machine room. Each rack may have from 3 to 5 compressors serving a series of loads with nearly identical evaporator temperature. A typical store will have 10 to 20 compressors in the 3-hp to 15-hp size range. Most compressor racks are “uneven parallel”, meaning that the capacities of compressors in a rack are not equal. This improves the ability for the system to handle part-load conditions efficiently. A typical compressor rack is shown in Figure 4-6.

Figure 4-6:   Supermarket Refrigeration System Compressor Rack

Source: Tyler Refrigeration Corporation

Semi-hermetic reciprocating compressors are used in most supermarket refrigeration systems. A new development is the use of scroll and screw compressors, which are capturing increasing market share. Screw compressors represented less than two percent of unit sales in supermarkets in 1994,[1] but have increased sales since then. Scroll compressors currently also have a significant market share (see discussion about the Hussmann PROTOCOL System below). Hermetic scroll compressors and semi-hermetic screw compressors are physically smaller than the traditional reciprocating compressors. Screw compressors also allow liquid refrigerant subcooling (or economized operation) without the use of a dedicated subcooling compressor. (Mechanical subcooling with the use of a subcooling compressor is shown in Figure 5-3 in Section 5.1.1)

The scroll design has also allowed the development by Hussmann of the PROTOCOL System which Hussmann claims had 6% market share in 1995. This is a distributed refrigeration system in which cabinets containing parallel scroll compressor racks are placed in the store next to their associated display cases. The system is water cooled with compact plate heat exchangers. The reduced size and noise level of the scroll compressor makes the concept possible. The major system benefits are claimed to be reduced refrigerant charge and refrigerant piping, which minimizes refrigerant leakage, and the elimination of the machine room floor space requirement. Potential disadvantages are the added maintenance requirements of water-cooled systems and that compressor rack service will occur in the sales area rather than in a hidden mechanical room.

A simplified supermarket refrigeration circuit is shown in Figure 4-7. The figure shows only one compressor rack. The dashed lines in the figure show separation between rack piping and field piping. Rack integrators generally supply a packaged compressor rack for which much of the necessary piping, components, and controls are pre-assembled. The field piping involves connection of heat recovery heat exchangers, condensers, and display cases. Each rack typically has a dedicated condenser or a separate circuit of a single common condenser. Separate circuits of the heat recovery heat exchanger would also be used for separate racks. Air-cooled, evaporative, and water-cooled condensers are used--air cooling is the most common for supermarkets. The balance of the system is generally made up of piping, insulation, valves, and controls. The system in the figure does not have hot gas defrost—this would require an additional manifold for the hot gas and additional control valves. The system also does not have mechanical subcooling. The system might be used for medium or high-temperature cases for which defrost is not necessary and mechanical subcooling is not economical.

Refrigeration circuits are fed in parallel from the liquid manifold, which is at the refrigerant condensing pressure. Refrigerant flows through each circuit to display cases or walk-ins, through one or more parallel sets of expansion valves and evaporators. The low pressure side of each circuit leads to the suction manifold and to the compressors.

---

[1] Based on U.S. Department of Commerce data, discussions with representatives from Carlyle, Carrier, Delta Heat Transfer; A.D. Little estimates.

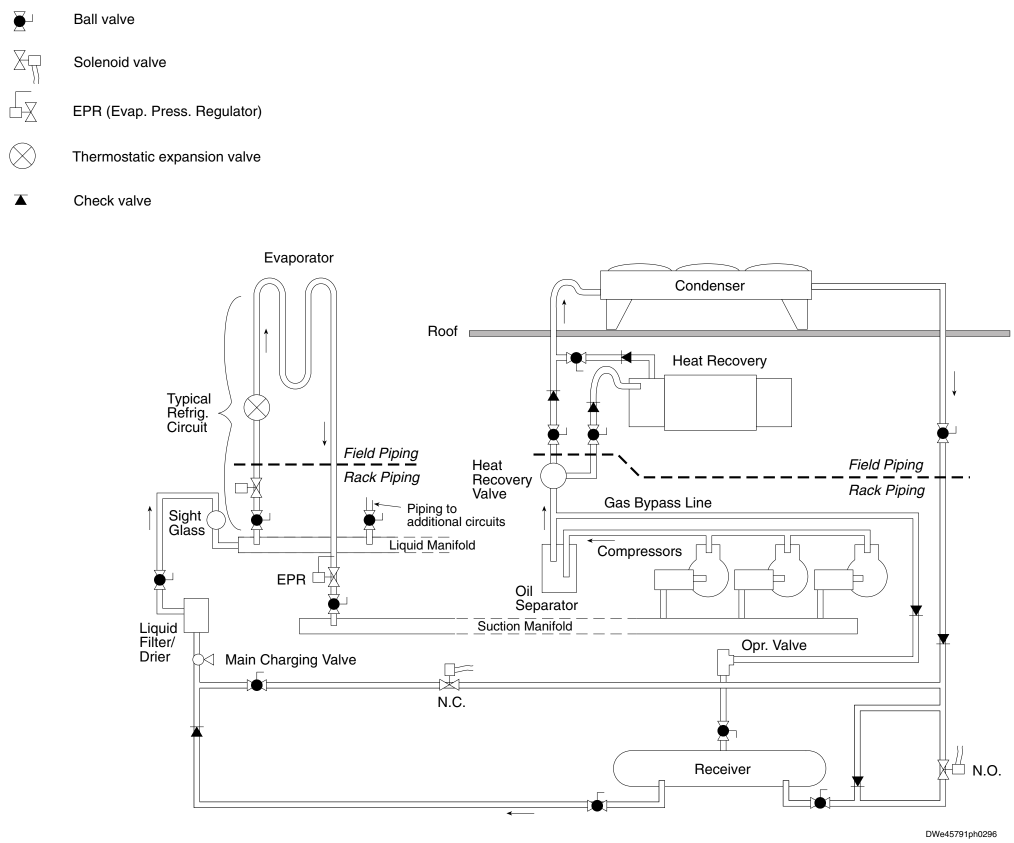


Figure 4-7: Supermarket Refrigeration System

Refrigerant leaves the compressors at condensing pressure, passing through the oil separator before moving to the heat recovery coil, which is located in the supermarket HVAC system. The heat recovery valve allows bypassing of this coil when heat is not needed. The refrigerant then is condensed, typically in a roof-mounted condenser. Condensed refrigerant is collected in a receiver, which feeds the liquid manifold.

4.1.2 Energy Consumption: Prototypical Supermarket Refrigeration System

Supermarkets range in size from less than 10,000 sq. ft. to greater than 70,000 sq. ft. total selling area.[2] The average size is about 27,000 sq. ft.[3] Supermarkets typically use on the order of 2,000,000 kWh of electricity per year.[4] The typical breakdown of this usage amongst building systems is shown below in Figure 4-8. The focus of this study is on the refrigeration system: the compressors, the condenser, and the cases.

---

[2] Progressive Grocer, April 1995, p. 52; Energy International, Inc., "Evaluation of Alternative Designs for Engine-Driven Supermarkets", 1991, p. 52.

[3] Progressive Grocer, April 1995, p.53.

[4] Foster Miller, "Guide for the Selection of Supermarket Refrigeraiton Systems:, 1990.

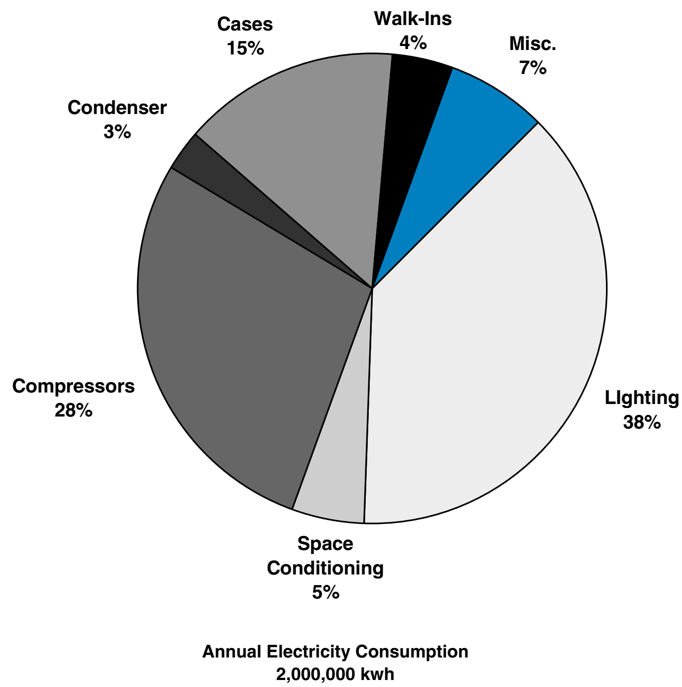


Figure 4-8: Supermarket Electricity Consumption Breakdown (27,000 sq. ft. Supermarket*)

*Average size according to Progressive Grocer, 62nd Annual Report of the Grocery Industry, April 1995.
Source: Foster Miller, 1990; estimates of this study

A supermarket with 45,000 sq. ft. sales area and 24 hour-per-day operation is chosen as the baseline for the energy consumption and savings calculations of Sections 4.1 and 5.1. This is larger than the average size of existing supermarkets, (about 27,000) but is about average for new supermarkets, according to representatives of Kysor Warren and Hussmann. Most large supermarkets average between 18 and 24 hours per day. Establishing a 24-hour store as the baseline results in some overestimation of usage and savings as compared with stores with less operational hours, but will not change the basic conclusions. The prototypical description of the supermarket was developed with the use of design information for example supermarkets provided by two manufacturers. These so-called "legends" list all supermarket refrigeration system components with associated design operating conditions. The prototypical supermarket has two medium temperature and two low temperature refrigeration systems with about 200 hp total connected compressor power.

The electricity consumption of the components of the prototypical supermarket refrigeration system is summarized in Table 4-3. Total annual electricity consumption is 1.6 million kWh, 55 percent of which is attributable to the compressors. An additional 7 percent of the usage is attributable to the condenser fans. The remaining usage is attributable to components in the display cases and walk-ins. Additional detail regarding the electricity usage is discussed below. The baseline demand is 242 kW, assuming that the defrost loads do not contribute to demand.

Table 4-3: Supermarket Refrigeration System--Electricity Consumption Summary (prototypical 45,000 sq. ft. supermarket)

Component	Power Consumption (kW)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)
2 Medium Temp Racks	88[2]	63[4]	485,700	31
2 Low Temp Racks	68[2]	63[4]	375,300	24
Condenser(s)	18[3]	63[4]	99,300	7
Medium Temp Display Cases[1]	31	6-100	129,600	8
Low Temp Display Cases[1]	198	2-100	348,200	22
Medium Temp Walk-Ins[1]	19	4-100	83,800	5
Low Temp Walk-Ins[1]	34	4-100	50,000	3
TOTAL			1,600,000	100

1 Fans, lights, defrost, and antisweat heaters - see Tables 4-7 through 4-10
2 Table 4-4
3 Fan power, based on the heat rejection requirements (Table 4.4) and Hussmann condenser performance data
4 Based on data of Reference 3 (see text)
Note: Percentages may not add to 100 due to round-off error

The prototypical supermarket has two medium temperature racks and two low temperature racks. The evaporator temperatures are assumed to be -25°F for the low temperature racks and 15°F for the medium temperature racks. In some situations different evaporator temperatures are selected. For instance, the medium-temperature display cases may be divided up to be served by two racks operating at 15°F and 20°F evaporator temperature. For this study, however, two evaporator temperatures for the entire store are considered. The refrigeration loads of the racks are summarized in Table 4-4.

Table 4-4: Compressor Rack Loads

	Saturated Suction Temp (F)	Saturated Discharge Temp (F)	System COP	Evaporator Load (mBtu/hr)	Compressor Power (kW)	Heat Rejection (mBtu/hr)
Medium Temp (2 Racks)	15	115	2.5[1]	750[2]	88	1050
Low Temp (2 Racks)	-25	110	1.3[1]	300[2]	68	531
Totals				1050	156	1581

1 Copeland Discus performance data (HCFC-22)
2 Table 4-5

The condenser temperatures of the two rack systems are often different. The medium temperature rack is likely to operate with a 115°F saturated discharge temperature. The saturated discharge temperature (SDT) is the saturation temperature associated with the compressor discharge pressure. The medium temperature, therefore, has a 15 degree temperature difference with the representative 100°F ambient design condition. For the low temperature rack, a 10 degree difference and 110°F saturated discharge temperature are common. The trade-offs between condenser capital cost, and life cycle compressor power costs allow lower condenser temperature difference (larger condenser size) for the low temperature system.

The coefficient of performance (COP) values for the racks are based on performance data for the Copeland Discus compressor. This semihermetic reciprocating compressor is used in a large percentage of supermarket refrigeration installations (see market share discussion in Section 4.1.6). The COP for the Discus series is fairly constant for the range of compressor sizes (3 to 15-hp) typically used in supermarket racks. This is true for both the medium and low temperature conditions. COP values for scroll and/or screw compressors have traditionally been lower than for reciprocating compressors of the same size. However, the performance gap has essentially been closed for the latest scroll compressors now available.

The compressor power in Table 4-4 is determined by dividing the evaporator load by the COP and converting to kW. The heat rejection load is the sum of the evaporator and compressor loads. The duty cycle for the compressors and condensers is usually in the range 60% to 70%. A representative value of 63% has been chosen based on simulation data presented by Foster Miller (1990).

The design evaporator loads are itemized in Table 4-5. These loads consist of display cases and walk-in coolers and freezers. Two display case types are assumed for each temperature level.[5] Two medium temperature walk-in types are considered. Note that display case amounts are reported in units of linear feet of display case. Size of walk-ins is on a floor area basis.

The store's refrigeration load includes 750 mBh of medium temperature cases and walk-ins and 300 mBh for the low temperature loads. These numbers include added loads due to lighting and electric defrost and contributions from anti-condensate heaters. The case loads per lineal foot are higher for the medium-temperature cases because they are large open multideck display cases.

The display case loads are broken down in Table 4-6 below according to contributions from electrically powered case components and other loads. The estimates for antisweat heaters assumes that half of the heat contributes to the case load.

---

[5] Although these prototypical cases do not represent the entire range of display case types, energy usage characteristics aare well represented by two types at each temperature level.

Table 4-5: **Refrigeration Load Summary**

Temperature Designation	Item	Quantity per Store	Load[1] per ft or per ft^2	Total Load[1] (mBtu/hr)
Medium	Multideck Meat Cases	120 ft	1500 Btu/hr/ft	180
	Other Multideck Cases	260 ft	1500 Btu/hr/ft	390
	Meat Walk-In Coolers	400 ft^2	60 Btu/hr ft^2	26
	Other Walk-In Coolers	2,600 ft^2	60 Btu/hr/ft^2	154
			Total	750
Low	Reach-In Cases	268 ft	560	150
	Single-Level Open Cases	128 ft	550	70
	Walk-In Freezers	1,000 ft^2	80 Btu/hr/ft^2	80
			Total	300

[1]Total refrigeration load and caseloads based on data from Hussmann and Kysor-Warren

Table 4-6: **Case Load Breakdown**

Case Type	Total Load (Btu/hr/ft)[1]	Lights[2] (Btu/hr/ft)	Evap. Fans[2] (Btu/hr/ft)	Defrost [2] (Btu/hr/ft)	Anti-Sweat[2,3] Heaters (Btu/hr/ft)	Other Loads[4] (Btu/hr/ft)
Multideck Meat	1500	40	91	26	17	1326
Multideck Other (MT)	1500	62	42	0	0	1396
Reach-In (LtT)	560	113	65	27	116	239
Single Level Open (LT)	550	0	34	29	41	446

[1] Caseloads based on data from Hussmann and Kysor-Warren

[2] See Tables 4-7 through 4-10

[3] The listed loads assume that half the anti-sweat heating contributes to the case load

[4] Includes product load, wall losses, infiltration

Electricity usage of the display case and walk-in components is itemized in Table 4-7, Table 4-8, Table 4-9, and Table 4-10. The itemized component loads are for the evaporator fans, antisweat heaters, lights, and defrost heaters. The relative importance to electricity usage of each of these components depends on the case type and temperature. For instance, the lighting usage ranges from 0 percent of the case electric load for single-level open freezer cases to 59 percent for non-meat multideck medium temperature cases. Defrost and antisweat heaters become less important for higher-temperature cases. The non-meat medium-temperature cases do not require such heaters.

Evaporator fans represent most of the walk-in electricity usage besides compressor power. Case lighting is much brighter than walk-in lighting because well-lit displays enhance sales. The lights in walk-ins are also shut off when food is not being transferred to or from them. The defrost and antisweat loads are less in walk-ins because of better insulation and gasketing.

Table 4-7: Medium Temperature Cases--Electricity Consumption

Case Type	Component	Power Consumption per foot (W)	Total Power Consumption (kW)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)
Multideck Meat (120 ft)	Evaporator Fans	26.7	3.2	100	28,000	48
	Antisweat Heaters	20	2.4	50	10,500	18
	Electric Defrost	135	16.2	5.6[1]	7,900	13
	Lights	11.8	1.4	100	12,300	21
	Total				58,700	100
Other Multideck (260 ft)	Evaporator Fans	12.5	3.3	100	28,900	41
	Lights	18.3	4.8	100	42,000	59
	Total				70,900	100
				TOTAL:	129,600	

[1] 20 minutes every 6 hours
Source: Hussmann Corporation

Table 4-8: Low Temperature Cases--Electricity Consumption

Case Type	Component	Power Consumption per foot (W)	Total Power Consumption (kW)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)
Reach-In (268 ft)	Evaporator Fans	20	5.36	96[1]	45,100	15
	Antisweat Heaters	71	19	96[1]	159,800	53
	Lights	33	8.8	100	77,100	26
	Electric Defrost	400	107	2[2]	18,700	6
	Total				300,700	100
Single-Level Open (128 ft)	Evaporator Fans	10	1.28	100	11,200	24
	Antisweat Heaters	24	3.07	100	26,900	56
	Lights	0	0		0	0
	Electric Defrost	420	53.8	2[2]	9,400	20
	Total				47,500	100
				TOTAL:	348,200	

[1] 23 hours per day
[2] 30 minutes per day
Source: Hussmann Corporation

Table 4-9: Medium Temperature Walk-Ins--Electricity Consumption

Type	Component	Power Consumption per sq ft (W)	Total Power Consumption (kW)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)
Meat Coolers (400 sqft)	Evaporator Fans	3.7	1.48	100	13,000	74
	Electric Defrost[3]	19.5	7.8	4.2[2]	2,900	16
	Lights[1]	1	0.4	50	1,800	10
	Total				**17,700**	**100**
Other Coolers (2600 sqft)	Evaporator Fans	2.4	6.24	100	54,700	83
	Lights[1]	1	2.6	50	11,400	17
	Total				**66,100**	**100**
				TOTAL:	**83,800**	

[1] Incandescent Lighting
[2] 20 Minutes every 8 hours
[3] Heated condensate pan included
Source: Hussmann Corporation

Table 4-10: Low Temperature Walk-Ins--Electricity Consumption

Component	Power Consumption per sqft (W)	Total Power Consumption (kW)	Duty Cycle (%)	Energy Consumption (kWh/yr)	Energy Consumption (%)
Evaporator Fans	4.0	4.0	100	35,000	70
Electric Defrost[3]	29	29	4.2[2]	10,600	21
Lights[1]	1.0	1.0	50	4,400	9
Total				**50,000**	**100**

[1] Incandescent Lighting
[2] 60 Minutes every 24 hours
[3] Heated condensate pan included
Source: Hussmann Corporation

4.1.3 Manufacturing, Purchase, and Installation Costs

The total installed cost of a 100 ton supermarket refrigeration system is between 1 and 1.1 million dollars. See Figure 4-9.

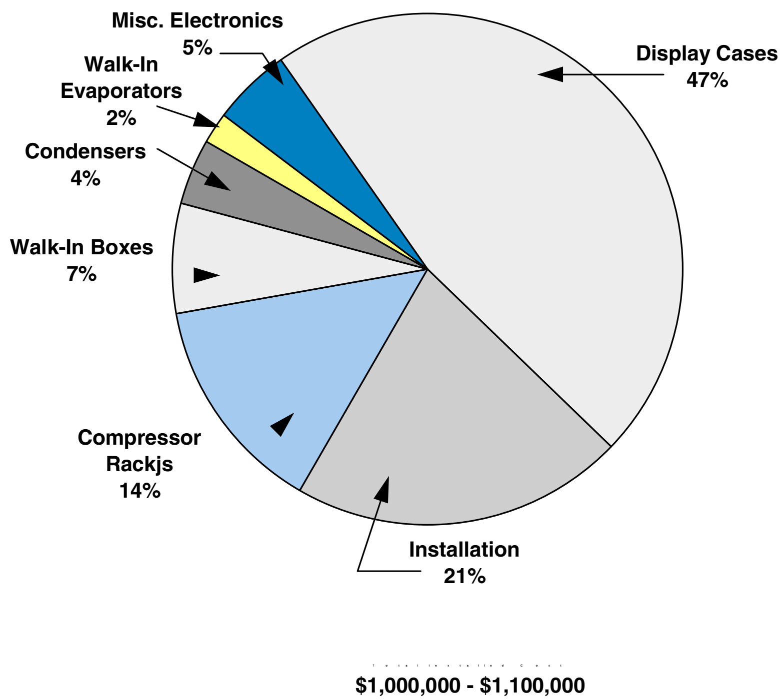


Figure 4-9: Installed Cost Breakdown for a 100 Ton Supermarket Refrigeration System

Source: Personal communication with supermarket industry representatives

4.1.4 Life, Reliability, and Maintenance Characteristics

Supermarkets system compressors have a 10-year expected lifetime. The typical lifetime of air-cooled condensers is at most 10 years. Refrigerated display cases are usually replaced for cosmetic reasons prior to the end of their useful life. Replacement occurs at 5-15 years, depending on store policies. 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.

Costs for refrigeration system maintenance are roughly 0.25% of supermarket revenues (Progressive Grocer, April 1995). The maintenance cost for a parallel refrigeration system is about $75 per 100 sq ft of store sales area (Adams, 1990). This gives a cost of about $20,000 annually for the average 27,000 sq ft supermarket.

4.1.5 Major Manufacturers

Figure 4-10 and Figure: 4-11 show estimates of the supermarket refrigeration equipment market and its breakdown among major manufacturers. A large percentage

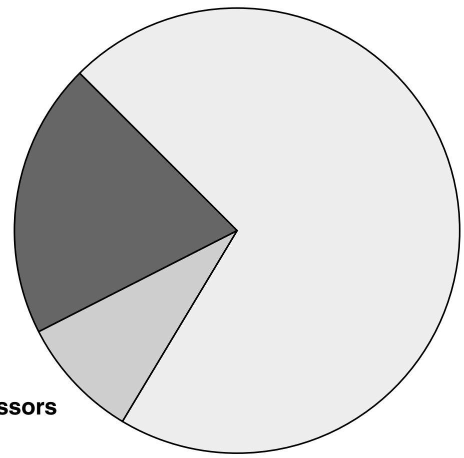


Figure 4-10: Estimated Supermarket Refrigeration Sales (1992)

Source: U.S. Department of Commerce; ACHR News

Error! Not a valid link.

Figure: 4-11: Supermarket Refrigeration System Market Share (based on sales values)

Source: Manufacturer Interviews and ADL Estimates

of this market is represented by the display cases. Four manufacturers dominate the market: Hussmann, Hill-Phoenix, Kysor-Warren, and Tyler. Figure 4-12 shows the distribution chain for supermarket refrigeration systems. Larger supermarket chains typically involve central engineering staff in design, selection, and installation of equipment. The manufacturer and/or contractor has a larger role in equipment selection for independent operators.

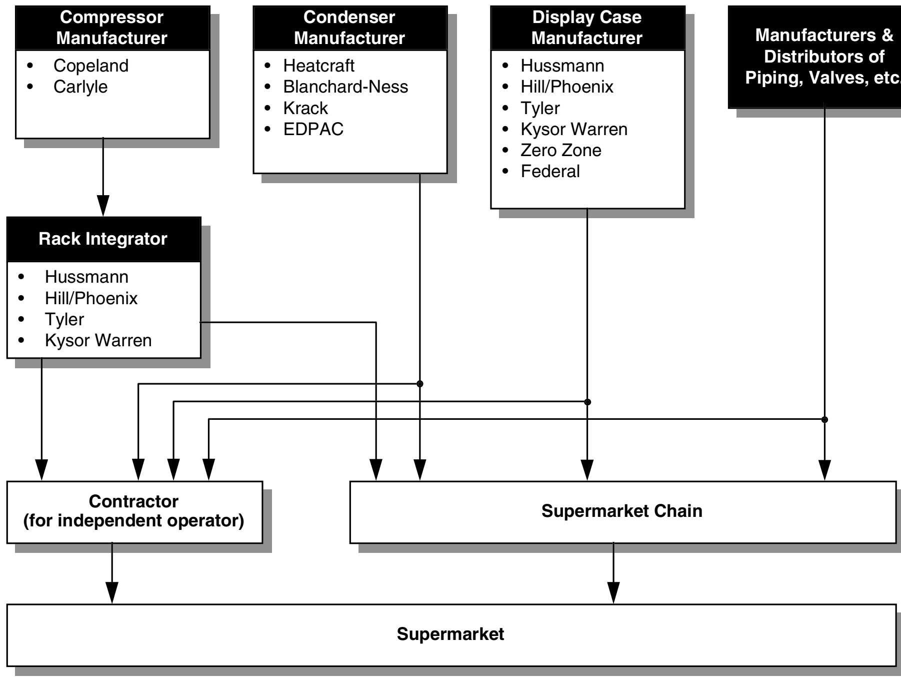


Figure 4-12: Supermarket Refrigeration Distribution Chain

Source: Manufacturer Interviews

Compressors are delivered to the supermarket already integrated into a 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. Additional major components are the condensers, the cases, and piping.

The major compressor manufacturers serving the supermarket sector are Copeland and Carlyle (a division of Carrier). Copeland leads in this market with about 65% of sales (see Figure 4-13). The total market for compressors used in supermarket refrigeration systems is about 55,000 units annually (ADL, “Energy Efficiency Alternatives to Chlorofluorocarbons”, for DOE, 1993).

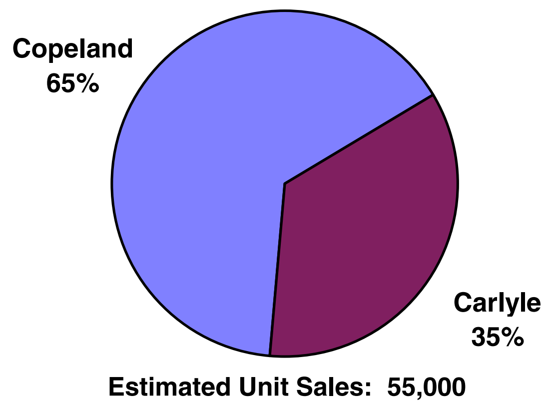


Figure 4-13: Supermarket Refrigeration Compressor Market

Source: Manufacturer Interviews and ADL Estimates

4.1.6 Major End-Users

The major end-users of supermarket refrigeration systems are the large supermarket chains. Table 4-11 gives a breakdown of the major chains.

Table 4-11: Top 10 Supermarket Chains

Sales Rank	Company	Number of Stores	% Market Share of Food Sales
1	Kroger	2,225	5.7
2	American Stores	904	4.8
3	Safeway	1,075	3.9
4	Albertson's	678	2.9
5	Winn-Dixie Stores	1,152	2.8
6	A & P	1,158	2.7
7	Food Lion	992	2.0
8	Publix Super Markets	447	1.9
9	Vons Companies	345	1.3
10	H.E. Butt Grocery Company	223	1.2
Totals		**9,199**	**29.1**

Source: Reference 8

The other major group of supermarket end-users, the independent operators, are subdivided into voluntaries and cooperatives. Voluntaries are groups of retailers which 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.

Market shares for the groups of supermarket types are 53% for the chains, 33% for the voluntaries, and 14% for the cooperatives. (8).

4.2 Beverage Merchandisers

Beverage merchandisers are upright, refrigerated display cases with glass doors and bright lighting whose purpose is to hold and display cold beverages. These cases are commonly used in convenience stores, aisle locations in supermarkets, and some retail stores and small foodservice establishments. The entire refrigeration system is built into the merchandiser and heat is rejected from the refrigeration cycle to the building interior air.

Beverage merchandisers are owned by bottling companies and vending companies. The bottling companies place the merchandisers in retail locations such as convenience stores. They 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.

Even though supermarkets and convenience stores use centralized systems to refrigerate most of their display cases, beverage merchandisers are still set up to maximize the overall sales area. Because these units are self-contained, they are easy to install in locations away from other refrigeration equipment and easy to relocate within a store. For example, in supermarkets beverage merchandisers are put near locations such as the checkout lane to lure customers into an "impulse buy".

Before 1993, estimated beverage merchandiser annual sales were about 60,000 units. By 1994, annual sales grew to about 120,000 units. In 1995 sales dropped again to historical levels of 60,000 units annually. The increased sales level was due to intensified marketing efforts by some bottlers as well as some consolidations.[6] Based on one manufacturer's estimates, there is an estimated installed base of between 750,000 and 1,000,000 beverage merchandisers. Approximately 50% are one-door units. Figure 4-14 shows the beverage merchandiser inventory breakdown for the commercial sector.

---

[6] Source: Beverage-Air

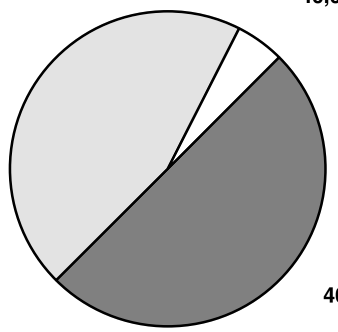


Figure 4-14: Beverage Merchandisers - 1994 Equipment Inventory

Source: Beverage-Air, 5/31/94

Based on an estimated inventory of 800,000 units, beverage merchandisers consume roughly 4.7 TWh (see Table 4-12).

Table 4-12: Commercial Sector Overview - Beverage Merchandiser Consumption

Unit Type	Estimated Inventory	Unit Energy Consumption, kWh/yr	Total Energy Consumption, TWh/yr	Total Energy Consumption, %
One-Door	400,000	3,900	1.56	33
Two-Door	360,000	7,600	2.74	58
Three-Door (or more)	40,000	11,200	0.45	9
Total	**800,000**	**-**	**4.74**	**100**

Source: ADL estimates; Beverage-Air, 5/31/94

4.2.1 Equipment Description and Illustration

The purpose of a beverage merchandiser is to 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 their products through these merchandisers provide refrigeration system performance specifications (pull-down time, holding temperature, etc.) and merchandiser aesthetics specifications to the manufacturer.

Figure 4-15 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 is used internationally.

Figure 4-15: Beverage Merchandiser - Equipment Illustration

Source: Dixie-Narco

Most beverage merchandisers are equipped with T-12 fluorescent lighting (1 1/2”-diameter tubes) to illuminate the beverages and the logo. A 20-watt bulb is usually used for the logo. Either a 20-watt or a 40-watt is used for product illumination. Both configurations can be designed with a single ballast.

Table 4-13 summarizes the physical characteristics of the prototypical beverage merchandiser.

Table 4-13: Beverage Merchandisers - Refrigerated Cabinet Description

Overall Exterior Dimensions			Overall Interior Dimensions			Insulation		Shelves		Door
W (in.)	D (in.)	H (in.)	W (in.)	D (in.)	H (in.)	Thick-ness (in.)	R-Value per inch (ft^2.°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: **Product literature from Beverage-Air and True Manufacturing, personal communication with Beverage-Air**

***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. A bottom-mounted system also provides easy access for maintenance and servicing.

The prototypical refrigeration system components consist of a 1/3 hp hermetic compressor, two evaporator fans and one condenser fan. Refrigerant flow is governed by a capillary flow restrictor. All fans are equipped with shaded-pole motors. In the past, CFC-12 was the refrigerant used by most manufacturers. Today, nearly all units are manufactured using HFC-134a.

The refrigeration circuit for a typical self-contained system is shown in Figure 4-16.

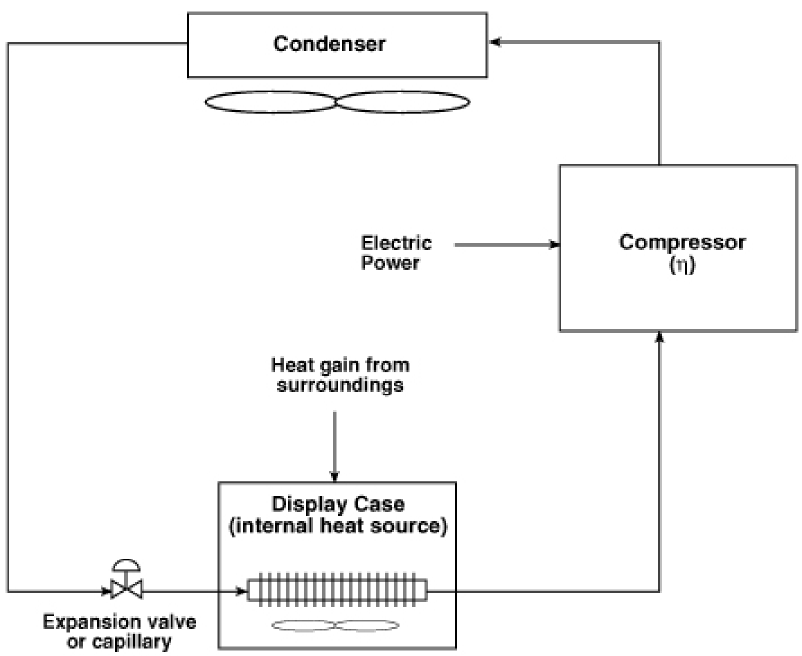


Figure 4-16: Typical Self-Contained Refrigeration Circuit

4.2.2 Energy Consumption

To characterize the energy consumption breakdown for a typical beverage merchandiser, the one-door unit was chosen since it is the most common unit currently used. Table 4-14 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 listed condition is 48%. This compares with efficiencies in the mid 50's, which are achieved with good residential refrigerator compressors.

Table 4-14: Beverage Merchandiser - Compressor Data

Compressor				Temperatures			
HP	Type	Capacity (Btuh)	Power Draw (W)	Cabinet (°F)	Evaporator (°F)	Ambient (°F)	Condensing (°F)
1/3	Reciprocating Hermetic	2,500	425	35	20	100	120

Source: Beverage-Air and Tecumseh compressor data

An estimate of the refrigeration load breakdown for the unit to show the relative importance of the different loads is shown in Table 4-15 below. Note that total average load is significantly lower than the compressor capacity. The compressor is sized for quick temperature pull-down.

Table 4-15: Beverage Merchandiser Load Breakdown

	Load (Btu/hr)
Evaporator Fans	362
Lighting	427
Infiltration	125
Wall Losses	204
Door Losses	50
Total	1168
Compressor Capacity	2500

Table 4-16 shows the energy usage breakdown for the beverage merchandiser.

Table 4-16: Energy Consumption Breakdown - Beverage Merchandiser (One-Door)

Component	Power Consumption, W[1]	Duty Cycle, %	Energy Consumption, kWh/yr	Energy Consumption, %
Compressor	425[2]	45[3]	1,675	43
Evaporator Fans (2)	106 (53x2)	100	928	24
Condenser Fan	57	45[3,4]	225	6
Lighting	125	100	1,095	27
Total	-	-	**3,923**	**100**

[1]Based on typical amp and power factor information provided by manufacturers
[2]1/3 hp compressor nominal power draw. Actual compressor power draw varies.
[3]Estimated duty cycle based on a 70°F ambient temperature: 35%. Additional 10% for pulldown
[4]Condenser fan cycles with the compressor

4.2.3 Manufacturing, Purchase, and Installation Costs

Table 4-17 shows list prices for two models of beverage merchandisers from Beverage-Air and True Manufacturing. The typical list price for a merchandiser in the 21 to 27 ft^3 range is between $2,300 and $2,400.

Table 4-17: Beverage Merchandisers - Manufacturer List Prices

Manufacturer	Model Number	Volume (cu ft)	Overall Exterior Dimensions			List Price, $
			W (in.)	D (in.)	H (in.)	
Beverage-Air	MT21	21	27	27.5*	78	2,283
Beverage-Air	MT27	27	30	32*	78	2,409
True Manufacturing	GDM-23	23	27	29.5	78.25	2,290
True Manufacturing	GDM-26	26	30.25	29.5	78.25	2,380

*Not including door handle

The average end-user purchase price is about 60% of the manufacturer list price. Therefore, a 27 ft^3 merchandiser should cost about $1,400.

4.2.4 Life, Reliability, and Maintenance Characteristics

The typical life of a beverage merchandiser is 7 to 10 years.[7]

Since almost all units are trade-identified, there is no significant used equipment market for beverage merchandisers. Bottling companies do not want their brand identity (i.e. the logo) to be misused. After a unit's 7 to 10 year life, 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.

4.2.5 Major Manufactures

The beverage merchandiser equipment market is dominated by two manufacturers: Beverage-Air and True Manufacturing (see Figure 4-17).

[7] source: Beverage-Air, 5/31/94

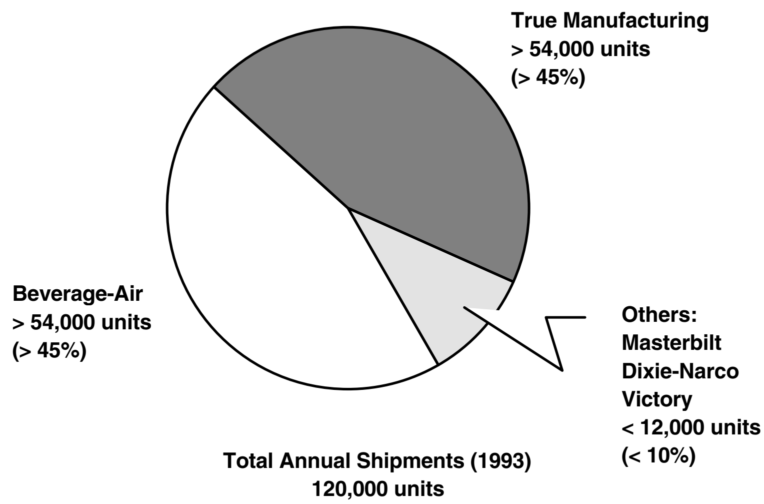


Figure 4-17: Beverage Merchandisers - Estimated Market Share

Source: Beverage-Air, 5/31/94

4.2.6 Major End-Users

Nearly all beverage merchandisers are purchased direct 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 sales. Smaller bottlers such as Snapple and Clearly Canadian account for less than 10%.

4.3 Reach-In Freezer

Reach-in freezers are upright, refrigerated cases with solid doors whose purpose is to hold frozen food products. These cases are commonly used in commercial and institutional foodservice establishments. The entire refrigeration system is built into the reach-in and heat is rejected from the refrigeration cycle to the building interior air.

Annual sales are estimated to be about 80,000 units. Based on inventory estimates from North American Food Manufacturers (NAFEM) and Food Management, there is an estimated installed base of 800,000 reach-in freezers. Approximately 55% are one-door units (Beverage-Air, 5/18/94). Figure 4-18 shows the reach-in freezer inventory breakdown for the commercial sector.

Error! Not a valid link.

Figure 4-18: Reach-in Freezers - 1994 Equipment Inventory

Sources: NAFEM, Food Management, ADL estimates

Based on an estimated inventory of 800,000 units, reach-in freezers consume roughly 6.0 TWh (see Table 4-18).

Table 4-18: Commercial Sector Overview - Reach-in Freezer Consumption

Unit Type	Estimated Inventory	Unit Energy Consumption, kWh/yr	Total Energy Consumption, TWh/yr	Total Energy Consumption, %
One-Door	440,000	5,200	2.29	38
Two-Door	320,000	9,800	3.14	52
Three-Door (or more)	40,000	14,400	0.58	10
Total	**800,000**	**-**	**6.00**	**100**

Sources: ADL Estimates, NAFEM, Food Management

4.3.1 Equipment Description and Illustration

The purpose of a reach-in freezer is to store frozen food products for commercial and institutional foodservice establishments.

Figure 4-19 shows a typical one-door reach-in freezer. Its 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 stands on four 6-inch legs. The cabinet and the doors are usually stainless steel. Antisweat heaters located along the door perimeter are used to prevent condensation and frosting on the gasket. There is one incandescent light (typically 25W) inside the freezer which operates when the freezer door is open. The evaporator coil has an electric defrost heater with about 600 W capacity that utilizes 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.

Figure 4-19: Reach-In Freezer - Equipment Illustration

Source: Continental Refrigerator Co.

Table 4-19 summarizes the physical characteristics of a prototypical reach-in freezer.

Table 4-19: Reach-in Freezers - Refrigerated Cabinet Description

Overall Exterior Dimensions			Overall Interior Dimensions			Insulation		Shelves		Door
W (in.)	D (in.)	H (in.)	W (in.)	D (in.)	H (in.)	Thick-ness (in.)	R-Value per inch ($ft^2 \cdot {}^\circ F$/ Btuh)	#	Total Shelf Space (ft^2)	Type
30	32	83	25	27	58	2-2.5	6.5-7	3	14-15	stainless steel

Source: Traulsen and Delfield product literature and personal communication

The refrigeration system is located at the top of the unit. This keeps refrigeration components away from spills and other debris unique to foodservice establishments, and reduces accumulation of dust on the condenser, while also keeping those components readily accessible for maintenance and servicing.

The refrigeration system components of the prototypical reach-in freezer consist of a 1/2 hp hermetic compressor, one evaporator fan and one condenser fan. Refrigerant flow is governed by a thermostatic expansion valve. Most units manufactured today use permanent-split-capacitor (PSC) fan motors. In the past, R-502 was the refrigerant used in most low temperature self-contained equipment. With the CFC phaseout, new reach-in freezers are being manufactured with R22 and R-404A (HP62).

Some models are available with hot gas defrost systems as an alternative to electric defrost.[8] 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 antisweat heaters.

The reach-in freezer refrigeration circuit is well represented by the typical self-contained system refrigeration circuit shown in Figure 4-16 in Section 4.2.1.

4.3.2 Energy Consumption

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

Table 4-20 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 over ambient. The compressor efficiency at the listed condition is 56%. This is comparable to the efficiencies achieved by good residential refrigerator/freezer compressors.

---

[8] With this defrost method, hot gas from the compressor discharge is piped through the evaporator during the defrost cycle, thus saving electricity. Additional piping and valves are required for this function.

Table 4-20: Reach-in Freezers - Refrigeration Component Description

Compressor				Typical Temperatures			
HP	Type	Capacity (Btuh)	Power Draw (W)	Cabinet (°F)	Evaporator (°F)	Ambient (°F)	Condensing (°F)
1/2	Hermetic	2,200	530	0	-20	90	110

Sources: Personal communication with Delfield and Traulsen

Table 4-21 below shows the typical load breakdown for the reach-in freezer.

Table 4-21: Reach-In Freezer Load Breakdown

	Load (Btu/hr)
Evaporator Fans	68
Lighting	3
Infiltration	41
Wall Losses	329
Defrost	128
Antisweat Heating	145[1]
Total	714
Compressor Capacity	2200

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

The energy consumption of the freezer is shown in Table 4-22 below. The compressor duty cycle is somewhat higher than a comparison of load and compressor capacity (Table 4-21) would suggest. Additional load is due to frequent door openings and some pull-down of food placed in the unit.

Table 4-22: Energy Consumption Breakdown - Reach-in Freezer (one-door)

Component	Power Consumption, W	Duty Cycle, %	Energy Consumption, kWh/yr	Energy Consumption, %
Compressor	530[1]	75[2]	3,482	67.0
Evaporator Fan	20	100	175	3.4
Condenser Fan	70	75[2, 3]	460	8.9
Anti-sweat Heaters	85[4]	100	745	14.3
Electric Defrost	600	6.25[5]	329	6.3
Lighting	25	3.125[6]	7	0.1
Total	**-**	**-**	**5,198**	**100.0**

1 1/2 hp compressor nominal power draw. Actual compressor power draw varies.
2 Duty cycle at 70°F ambient temperature based on manufacturer estimates
3 Condenser fan cycles with the compressor.
4 There are 6 W of anti-sweat heaters per linear foot of door perimeter (27” x 58”).
5 Defrost cycles 3 times per day, 0.5 hours per cycle
6 One incandescent 25 W light operates when freezer door is open (0.5 to 1 hour per day).

4.3.3 Manufacturing, Purchase, and Installation Costs

Table 4-23 shows list prices for one-door reach-in freezers of comparable size from four manufacturers. The list price for a single door reach-in freezer is between $3,500 and $4,000.

Table 4-23: Reach-in Freezers (One-Door) - Manufacturer List Prices

Manufacturer	Model Number	Volume (cuft)	Overall Exterior Dimensions W(in.)	D(in.)	H(in.)	List Price, $
Delfield	6125-S	20.0	25.5	34.75	79.5	3,789
Traulsen	G12010(11)	24.2	29.88	34.94	83.25	3,995
Continental Refrigerator	1F	19.0	26	35	83.25	3,448
Nor-Lake	GF1W	22.5	27.5	35.5	82.75	3,476

Note: Depth includes door handles and other hardware.

The average end-user purchase price is about 60% of the manufacturer list price. Therefore, a 24 ft^3 reach-in freezer should have an average cost of about $2,200.

4.3.4 Life, Reliability, and Maintenance Characteristics

The typical life of a reach-in freezer is 8 to 10 years.[9] 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 done if there is a problem.

4.3.5 Major Manufacturers

Figure 4-20 and Figure 4-21 are market share figures for reach-in refrigerators & freezers, "standard-line" and "specification-line", respectively. "Standard-Line" refrigerators and freezers, representing about 70% of 200,000 annual sales, are sold primarily to commercial food establishments. "Specification -line," representing 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.

4.3.6 Major End-Users

Reach-in freezers 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 such as McDonalds. The market for reach-in refrigerators and freezers is very fragmented due to the large variety and number of restaurants and other users.

[9] Source: personal communication with Tom Yingst of Traulsen

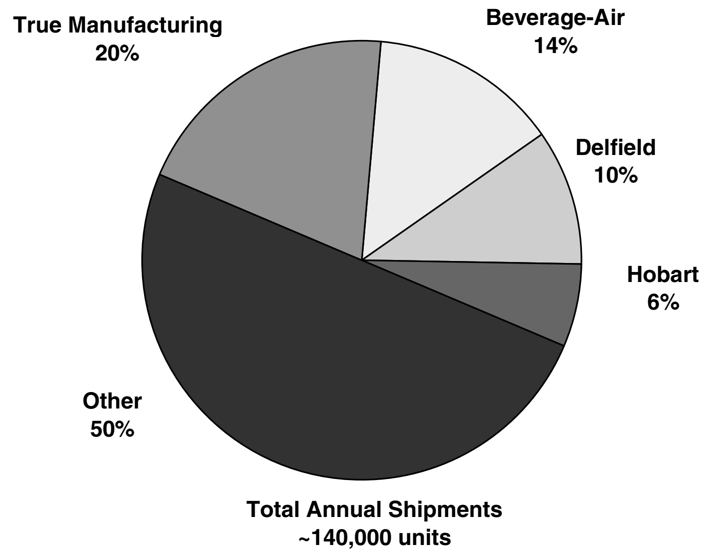


Figure 4-20: Reach-In Refrigerators & Freezers - "Standard-Line" Estimated market Share

Source: Easton Consultants, 1993; Manufacturer Estimates; ADL Estimates

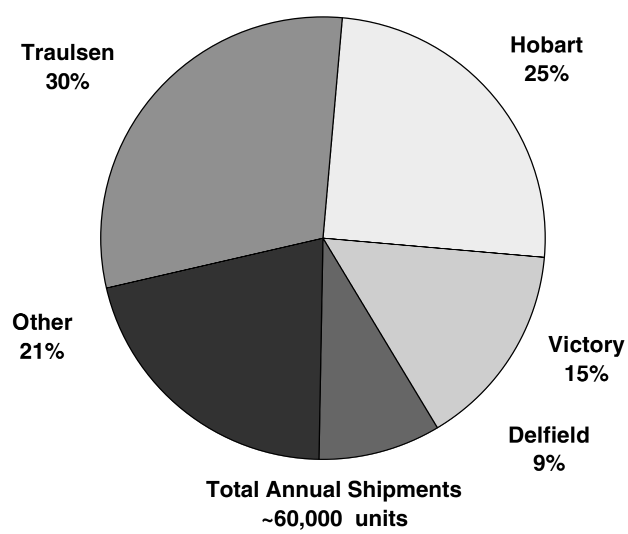


Figure 4-21: Reach-In Refrigerators & Freezers - "Specification-Line" Estimated Market Share

Source: Easton Consultants,1993; manufacturer estimates; ADL estimates

4.4 Reach-In Refrigerator

Reach-in refrigerators are upright, refrigerated cases with solid doors whose purpose is to hold refrigerated food products. These cases are commonly used in commercial and institutional foodservice establishments. The entire refrigeration system is built into the reach-in and heat is rejected from the refrigeration cycle to the building interior air.

Annual sales are estimated to be about 120,000 units. Based on inventory estimates from NAFEM and Food Management, there is an estimated installed base of 1,300,000 reach-in refrigerators. Approximately 65% are two-door units. Figure 4-22 shows the reach-in refrigerator inventory breakdown for the commercial sector.

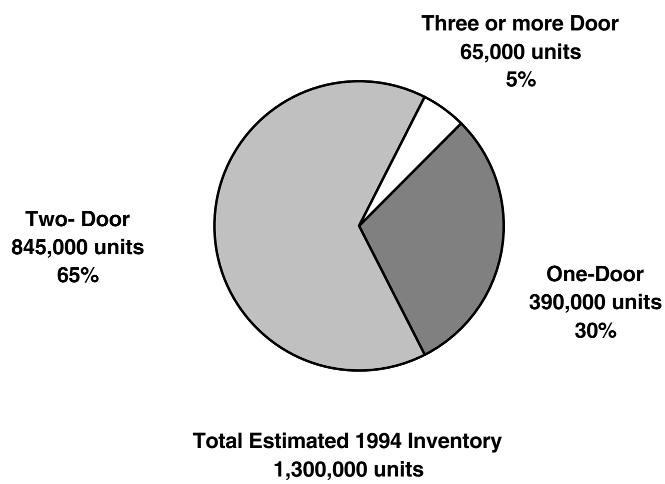


Figure 4-22: Reach-In Refrigerators - 1994 Equipment Inventory

Sources: NAFEM, Food Management; ADL Estimates

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 1,300,000 units, reach-in refrigerators consume roughly 4.9 TWh (see Table 4-24).

Table 4-24: Commercial Sector Overview - Reach-in Refrigerator Consumption

Unit Type	Estimated Inventory	Unit Energy Consumption, kWh/yr	Total Energy Consumption, TWh/yr	Total Energy Consumption, %
One-Door	390,000	2,300	0.90	18
Two-Door	845,000	4,300	3.63	74
Three-Door (or more)	65,000	6,300	0.41	8
Total	**1,300,000**	**-**	**4.94**	**100**

Sources: NAFEM, Food Management, ADL estimates.

4.4.1 Equipment Description and Illustration

The purpose of a reach-in refrigerator is to hold refrigerated food products for commercial and institutional foodservice establishments.

Figure 4-23 shows a typical 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 stands on four 6-inch legs. Cabinets and doors are usually stainless steel. Antisweat heaters are installed along the door perimeter to prevent condensation on the door gasket. There are two incandescent lights (usually 25W each) inside the refrigerator which operate when either refrigerator door is open

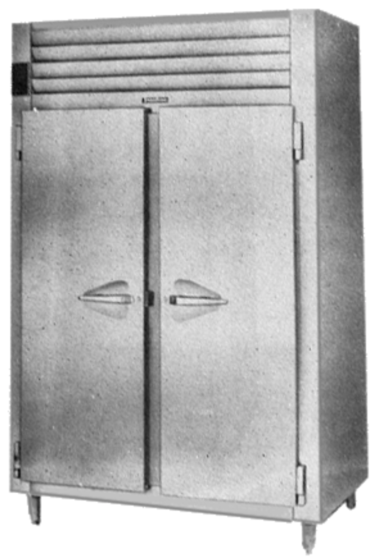


Figure 4-23: Reach-In Refrigerator - Equipment Illustration

Table 4-25 summarizes the physical characteristics of the reach-in refrigerator. The refrigeration system is located at the top of the unit. This keeps refrigeration components away from spills and other debris unique to foodservice establishments and reduces accumulation of dust on the condenser while also keeping those components readily accessible for maintenance and servicing.

Table 4-25: Reach-in Refrigerators - Refrigerated Cabinet Description

Overall Exterior Dimensions			Overall Interior Dimensions			Insulation		Shelves		Door
W (in.)	D (in.)	H (in.)	W (in.)	D (in.)	H (in.)	Thick-ness (in.)	R-Value per inch ($ft^2 \cdot °F$/ Btuh)	#	Total Shelf Space (ft^2)	Type
52	32	83	48	27	58	2-2.5	6.5-7	6	26-33	stainless steel

Sources: Traulsen and Delfield product literature and personal communication

The refrigeration system components consist of a 1/3 hp hermetic compressor, two evaporator fans and one condenser fan. Refrigerant flow is governed by a capillary flow restrictor. Most units manufactured today use permanent-split-capacitor (PSC) fan motors. In the past, R-12 was the refrigerant used in most medium temperature self-contained equipment. With the CFC phaseout, new reach-in refrigerators are being manufactured to use R-134a.

The self-contained refrigeration circuit shown in Figure 4-16 in Section 4.2.1 is typical for reach-in refrigerators.

4.4.2 Energy Consumption

To characterize the energy consumption breakdown for a typical reach-in refrigerator, the two-door unit was chosen since it is the most common unit currently used. Table 4-26 summarizes the characteristics of the compressor and the associated design temperatures. The evaporator temperature is usually about 20°F and the condenser temperature is usually about 20°F above ambient. 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.

Table 4-26: Reach-in Refrigerators - Refrigeration Component Description

Compressor				Typical Temperatures			
HP	Type	Capacity (Btuh)	Power Draw (W)	Cabinet (°F)	Evaporator (°F)	Ambient (°F)	Condensing (°F)
1/3	Hermetic	3,000	440	40	20	90	110

Sources: Personal communication with Delfield and Traulsen

Table 4-27 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 pulldown of warm food.

Table 4-27: Reach-In Refrigerator Load Breakdown

	Load (Btu/hr)
Evaporator Fans	164
Lighting	5
Infiltration	62
Wall Losses	265
Antisweat Heating	169[1]
Total	665
Compressor Capacity	3000

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

The energy consumption of the refrigerator is shown in Table 4-28 below.

Table 4-28: Energy Consumption Breakdown - Reach-in Refrigerator (two-door)

Component	Power Consumption, W	Duty Cycle, %	Energy Consumption, kWh/yr	Energy Consumption, %
Compressor	440[1]	65[2]	2,506	58.0
Evaporator Fans	48 (2x24)	100	420	9.7
Condenser Fan	90	65[2, 3]	512	11.9
Anti-sweat Heaters	99[4]	100	869	20.1
Lighting	50 (2x25)	3.125[5]	14	0.3
Total	**-**	**-**	**4,321**	**100.0**

1 1/3 hp compressor nominal power draw. Actual compressor power draw varies.
2 Duty cycle at 70°F ambient temperature based on manufacturer estimates
3 Condenser fan cycles with the compressor.
4 There are 3.5 W of anti-sweat heaters per linear foot of door perimeter (27" x 58").
5 Two incandescent 25 W lights operate when either refrigerator door is open (0.5 to 1 hour per day).

4.4.3 Manufacturing, Purchase, and Installation Costs

Table 4-29 shows list prices for one-door reach-in refrigerators of comparable size from four manufacturers. The list price for a two-door reach-in refrigerator is between $3,800 and $4,500.

Table 4-29: Reach-in Refrigerators (Two-Door) - Manufacturer List Prices

Manufacturer	Model Number	Volume (cu ft)	Overall Exterior Dimensions			List Price, $
			W (in.)	D (in.)	H (in.)	
Delfield	6051-S	43.5	51	34.75	79.5	4,468
Traulsen	G20010(11)	46.0	52.13	34.94	83.25	4,435
Continental Refrigerator	2R	48.0	52	35	83.25	3,787
Nor-Lake	GR482	48.8	55	35.5	82.75	4,045

The average end-user purchase price is about 60% of the manufacturer list price. Therefore, a 48 ft^3 reach-in refrigerator should have an average cost of about $2,500. According to NAFEM, roughly 50% of units purchased by restaurants are of used equipment.

4.4.4 Life, Reliability, and Maintenance Characteristics

The typical life of a reach-in refrigerator is 8 to 10 years.[10]

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

4.4.5 Major Manufacturers

Reach-in refrigerators and reach-in freezers are manufactured by the same companies. Refer to Section 4.3.5

[10] Source: personal communication with Tom Yingst of Traulsen

4.4.6 Major End-Users

Reach-in refrigerators and reach-in freezers are sold to the same end-users. Refer to Section 4.3.6

4.5 Ice Machines

Ice machines are used to produce a variety of ice types used in the food service, food preservation, hotel, and hospital industries. The types of ice produced include:

- cube ice -distinct portions of clear, regularly shaped ice ranging from 1/6 - 1/2 oz. in weight containing minimal quantities of liquid water with the ice
- 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
- crushed ice -small, irregular portions of ice created by crushing larger portions used primarily for keeping drinks cool
- nugget ice - small cloudy, nugget-shaped portions of ice created by extruding and freezing the slushy ice/water mixture of flake ice into a nugget; used primarily for keeping drinks cool

Ice machine sales in 1993 (according to Department of Commerce Data) are summarized in Table 4-30 below.

Table 4-30: Approximate 1993 Ice Machine Unit Sales

Equipment Description	1993 Unit Sales
Self-contained cubers 200 lb/day and under	72,508
Self-contained cubers over 200 lb/day	48,224
Not self-contained cubers	41,753
Self-contained flake machines 300 lb/day and under	5,860
Self-contained flake machines over 300 lb/day	3,913
Combination Ice Machines and ice/drink dispensers	15,321
Total	**187,579**

The term self-contained in Table 4-31 refers to equipment in which the ice making equipment and ice storage compartment are contained in an integral cabinet. Note that all ice machine types discussed in this chapter have self-contained refrigeration systems. Combination Ice Machines and ice/drink dispensers include equipment in which the ice making equipment, ice dispenser, and possibly a drink dispenser are contained in an integral unit. The ice type produced in the combination unit may be cube, flake, crushed, or nugget ice.

As can be seen in Table 4-31, ice cubers account for over 80 percent of the ice machine sales. The discussion in the following material will focus on ice cubers, herein referred to as ice machines.

Primary applications for cube ice include the fast food and food service industries, hotel vending rooms, and bag ice sold at convenience stores. Machines are referred to by their nominal capacity, defined as the weight of ice produced per 24 hour period. Manufacturers produce machines of similar capacities which typically are about 250, 400, 500, 650, 800, 1,000, 1,200, and 1,400 lb/24 hours (these nominal capacities typically refer to operation in ambient temperatures of 70°F and inlet water temperature of 50°F) with the very largest machines producing up to several tons per day. The largest machines are used in airline and airport food service industries, very large restaurants, and industrial processes.

Ice machines are typically located indoors (e.g., kitchen area, hotel vending rooms) and are sometimes located outdoors (e.g., walkways 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 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.

4.5.1 Equipment Description and Illustration

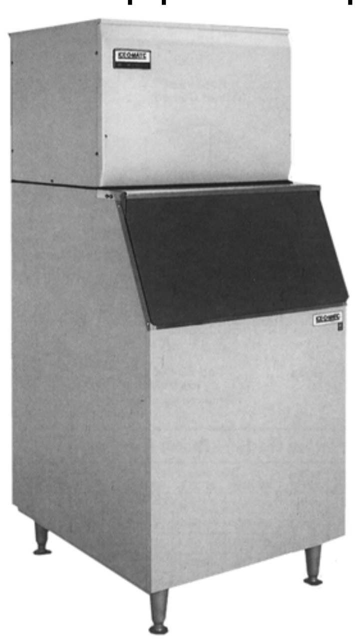


Figure 4-24 illustrates a typical ice cuber mounted on an insulated ice storage bin.

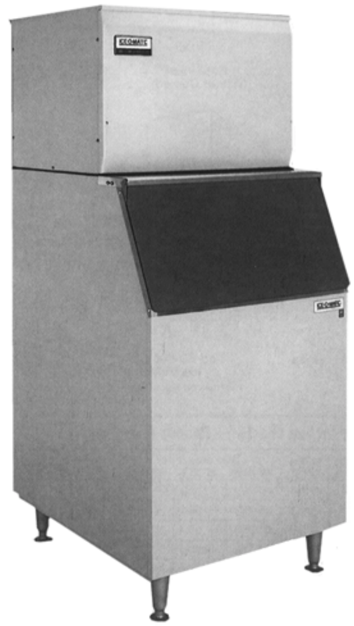


Figure 4-24: Ice Machine Equipment Illustration

Ice machines are generally integrated with an insulated ice storage bin or are mounted atop a separate storage bin. The bins generally have a full-width door for user access to the ice. Expansion of ice production capacity for non-self-contained machines is achieved 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.

About 80 percent of ice machines have integral air-cooled condensers. Other condenser configurations include integral water-cooled and remote air-cooled.

Because impurities in the water supply are practically insoluble in ice, deposits already present in the water supply are rejected from the water as it freezes, which concentrates impurities in the liquid water. Over time, lime and scale adheres to the water system and evaporator surfaces requiring the periodic cleaning of the machine using cleaners such as acetic and phosphoric acid. To lower the concentration of impurities and reduce the level of lime and scale buildup, most ice machines circulate within the machine and or flush the machine with more water than required for the production of a given quantity of ice. Typical potable water consumption ranges from 12.5 - 36 gallons/100 lb of ice produced, compared to the minimum of 12 gallons/100 lb.

An ice machine consists of two major subsystems: the refrigeration system and water supply/ circulation/purge system. All ice machines use vapor compression refrigeration to produce the refrigeration needed for ice production. Primary refrigeration system components include:

- compressor - Typically, conventional reciprocating refrigeration or heat pump compressors are used, capacity 1/3 to 3 hp, depending on ice machine size.
- condenser - Conventional air-cooled fin-tube or water-cooled concentric tube. 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 interchanger
- refrigerant piping
- hot gas bypass line - This directs refrigerant directly from compressor to evaporator for harvesting the ice.
- hot-gas solenoid valve - This controls hot gas refrigerant flow to the condenser during ice production and to the evaporator during ice harvest.
- refrigerant - Traditionally, R-502 and R-22 have been used, now moving to R-404A (HP-62).
- 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

The basic ice making process is a batch process and is described as follows:

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 to the proper ice batch weight as 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 directly 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 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 free heat contained in 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 prechilling 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 is more of an art than a science and 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. In addition, the evaporator design and cube shape are used to distinguish manufacturers from each other.

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.

4.5.2 Energy Consumption

4.5.2.1 General Characteristics

Ice machine performance (capacity, energy consumption, and water consumption) is usually presented for operating conditions prescribed by the Air-Conditioning and Refrigeration Institute (ARI), which are for a 90°F ambient temperature and a 70°F inlet water temperature. Figure 4-25 shows a plot of energy consumption versus capacity at these conditions for air-cooled machines summarized in the 1994 ARI Directory of Certified Automatic Commercial Ice-Cube Machines and Ice Storage Bins, effective September, 1, 1993 - February 28, 1994.

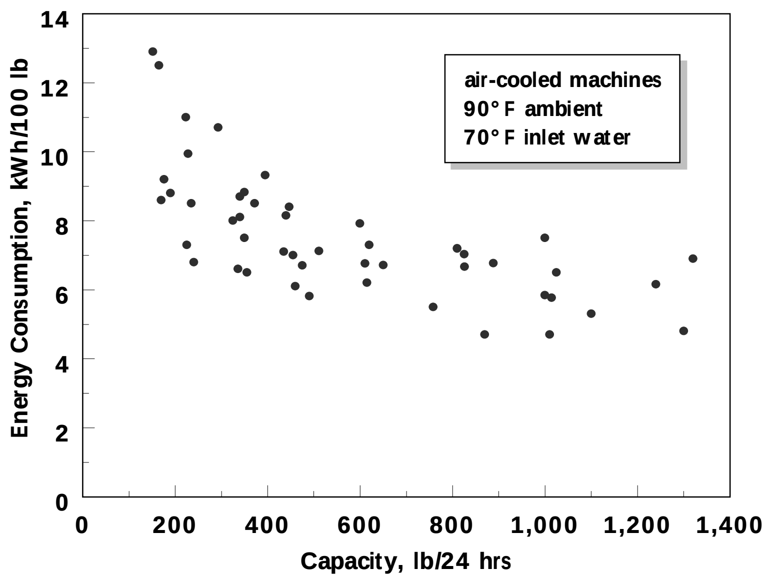


Figure 4-25: 1994 Ice Machine Energy Consumption Versus Capacity

Source: ARI

As can be seen in the Figure 4-25, energy consumption decreases with capacity, ranging from an average of about 10 kWh/100 lb in the 200 lb/day capacity range down to an average of about 6 kWh/100 lb in the 1,400 lb/day capacity range. The spread in the data for the small machines is quite large, ranging from about 7.5 - 13 kWh/100 lb.

Figure 4-26 shows ice machine water consumption versus capacity. In general, water consumption decreases with capacity, with the average water consumption for the small machines at about 25 gal/100 lb decreasing to about 17 gal/100 lb. The spread in water consumption for the small machines is huge, ranging from about 13 gal/100 lb up to 35 gal/100 lb.

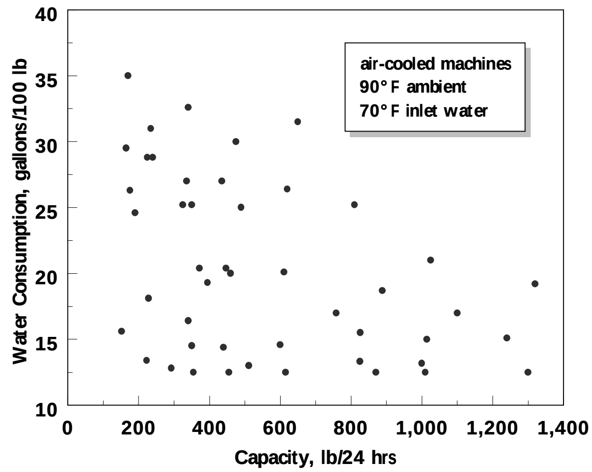


Figure 4-26: 1994 Ice Machine Water consumption Versus Capacity

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

- 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 desire 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.

In general, energy and water-use efficiency are not important market drivers. In fact, higher water consumption is thought to improve overall machine reliability by keeping components such as the water pump and sump water level detector clean from scale and by keeping the evaporator clean to promote ice detachment during harvest

4.5.2.2 Estimated Total Energy Consumption

Annual primary energy usage is summarized in Table 4-31 below. The table presents energy usage by building type. Average annual electricity usage per machine varies by application based on machine size and duty cycle.

Table 4-31: Ice Machine Primary Energy Usage

Building Type	Number of Buildings (1000's)	Ice machine Inventory (1000's)	Electricity Usage (TWh)	Primary Energy Usage* (Trillion Btu)
Office	614	33.6	0.4	4.3
Retail	1,287	70.4	0.8	9.1
Restaurant	201	164.9	1.3	14.1
Grocery	102	27.9	0.3	3.6
Schools	241	65.9	0.8	8.5
Hotel	137	524.5	2.1	22.5
Hospitals	52	312.8	3.7	40.2
Total	3,183	1,200	9.4	102.3

***Based on 10,867 Btu/kWh**
Source: ADL, "Characterization of Commercial Appliances," 1993

4.5.2.3 Energy Consumption Breakdown

The following discussion will use an air-cooled machine with a nominal capacity of 500 lb/day. Table 4-32 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 5,000 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.

Table 4-32: Estimated Annual Energy Consumption by Component

Component	Average Power Consumption, watts	Typical Duty Cycle, % (full capacity operation)	Estimated Annual Energy Consumption ** kWh (% total)
Compressor (during freezing)	1,000	90-95	4,050 (81)
Compressor (during harvest)	1,400	5-10	460 (9.2)
Condenser Fan	100	90-95	405 (8.1)
Water Pump	20	90-100	80 (1.6)
Hot-gas solenoid valve	15	5-10	5 (0.1)
Total	**---**	**---**	**5,000 (100)**

* For 500 lb/day air-cooled machine operating in a 70°F ambient with 50°F inlet water

** Assuming an overall annual duty cycle of 50%

Because the compressor accounts for about 90 percent of the total energy consumption, significant efficiency improvement must begin with reducing the compressor energy consumption.

Figure 4-27 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, subcooling of the ice, and heat input from the water pump motor accounts for 6 percent of the total compressor energy input.

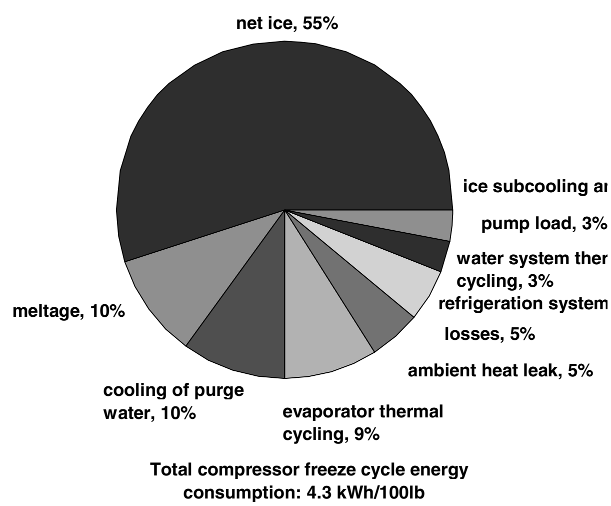


Figure 4-27: Compressor Freeze Cycle Energy Consumption by Load

Source: Calculations for a typical ice machine using the program FREEZE. Validation of the program was done by comparison with confidential test data of three manufacturers' machines.

Based on the results shown in Figure 4-27, 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. This will be discussed in the next section.

4.5.3 Manufacturing, Purchase, and Installation Costs

The factory cost of a 500-lb ice machine is about $1,000. The dealer markup is about 50%. The market is very competitive with many producers, resulting in little price increase over the past few years.

Installation for ice machines requires machine delivery and placement, and connection of electric power, water supply, and drainage piping. The cost is about $200. Machines with remote condensers involve additional cost of about $100 for installation of the condenser.

4.5.4 Life, Reliability, and Maintenance Characteristics

Reliability and first cost are the primary 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 it can be justified because of higher reliability.

Ice machine life is in the range of 7 - 10 years.

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, 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. Self-cleaning/sanitizing machines have been introduced into the market in the past few years. These machines eliminate many of the manual steps of the cleaning/sanitizing process and can be programmed to clean/sanitize at prescribed intervals.

4.5.5 Major Manufacturers

Major manufacturers and approximate market share are summarized in Table 4-33 below.

Table 4-33: Major Ice Machine Manufacturers and Approximate Market Share

Manufacturer	Market Share, Percent
Manitowoc	30
Scotsman/Crystal Tips	25
Hoshizaki	20
Ice-O-Matic	15
Serve-End, Icecraft, Vogt, Cornelius, others	10

4.6 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. These cases can be found almost anywhere. According to *Vending Times*, the most common locations are inside or outside factories, offices, health care institutions, schools, hotels, colleges and universities and other public locations (Vending Times, August 1994) The entire refrigeration system is built into the machine and heat is rejected from the refrigeration cycle to the surrounding air.

There is an estimated installed base of about 4,100,000 refrigerated vending machines. Approximately 59% are canned beverage vendors. The canned beverage vending machine was chosen for analysis in this report since it is the most common unit. It is expected that its energy consumption characteristics will be similar to those of other types of refrigerated vending machines. Figure 4-28 shows the refrigerated vending machine inventory breakdown for the commercial sector.

About 80% of all canned beverage vending machines are purchased directly from the manufacturer by bottling companies (i.e. Coca-Cola, Pepsi, etc.) (personal communication with Dixie-Narco and Coca-Cola: References 18, 19). Some of these units are supplied to independent vending operators on consignment while the remaining units are owned and operated by the bottlers themselves (see Figure 4-29).

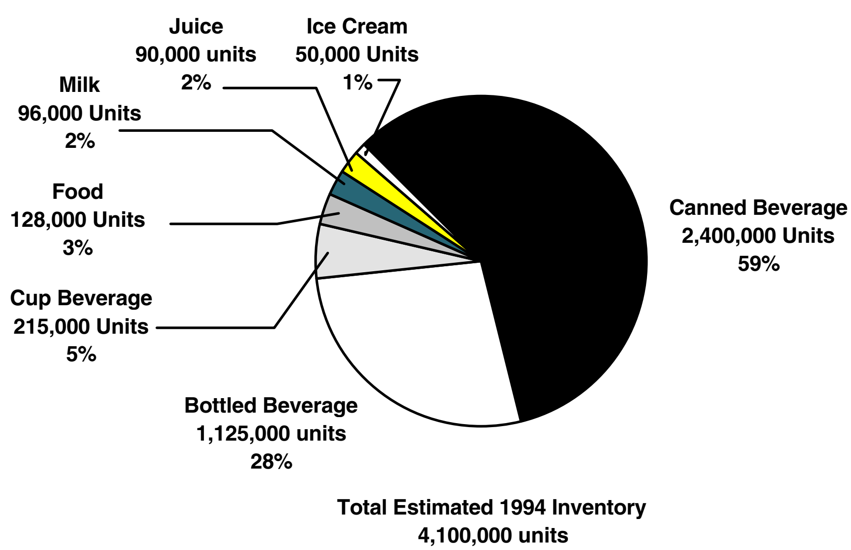


Figure 4-28: Refrigerated Vending Machines - 1994 Equipment Inventory

Sources: Vending Times, August 1994; discussion with a representative of Vending Times 6/2/94

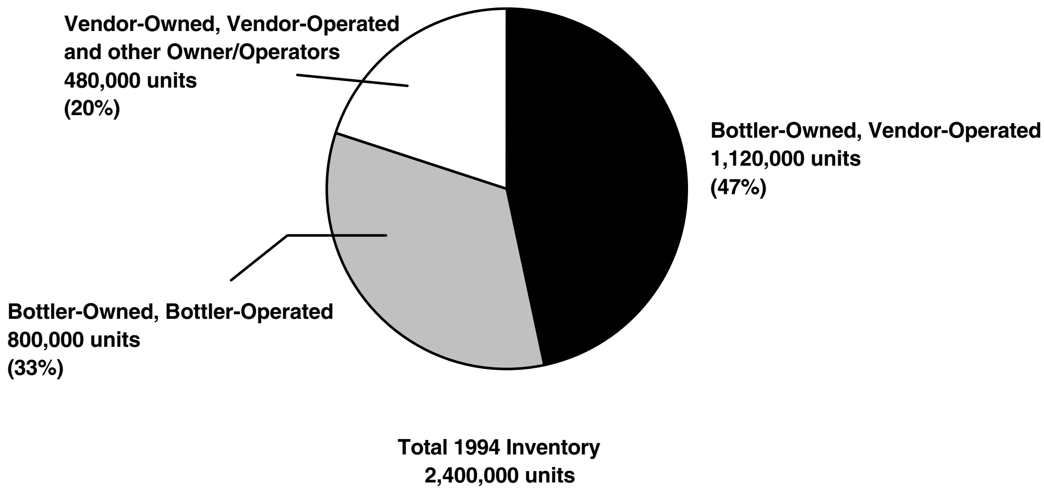


Figure 4-29: Refrigerated Vending Machines (Canned Beverage) - Owner/Operator Inventory Share

Sources: Vending Times, Dixie Narco, Coca-cola

The other 20% of canned beverage vending machines are purchased by owner/operators. These include “Ma & Pa” stores, canteens, foodservice operators and vending operators (i.e. American Vending).

There are about 7,100 canned beverage vending operators in the United States. Approximately 4,700 are independent operators and 2,400 are bottling company operators. 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.

Typically, a vending operator will roughly double the cost of a canned beverage when the beverage is sold through a machine. For example, a canned beverage purchased from a bottling company for $0.40 will be sold for $0.80 in a vending machine. The average vending machine dispenses about 192 cans per week (Vending Times, August 1994, p. 15), hence average weekly revenues total $154. About half of this revenue goes directly to the bottling company and the remainder is divided between the operator and the vending site. Figure 4-30 shows the breakdown of weekly sales.

Based on an estimated inventory of 2,400,000 units, canned beverage vending machines consume roughly 7 TWh annually (see Table 4-34). Assuming that other refrigerated vending machines have similar performance characteristics, all refrigerated vending machines together consume about 12 TWh annually.

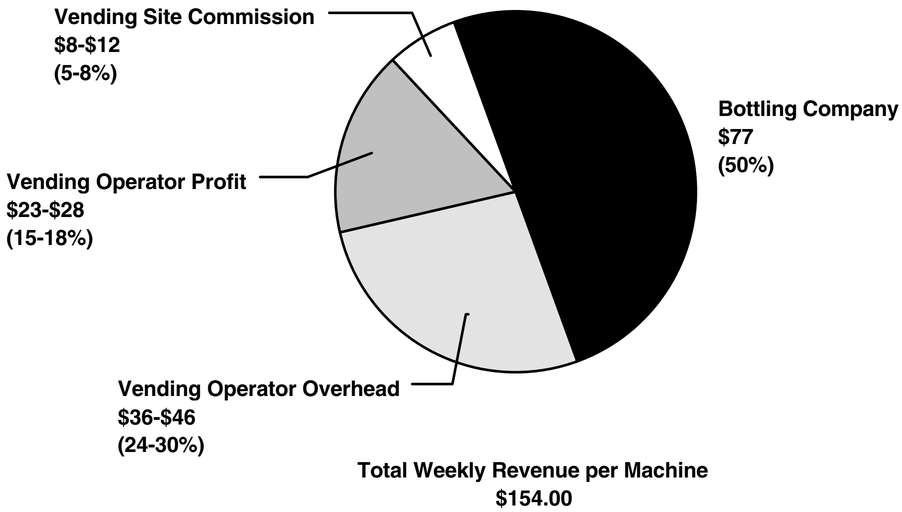


Figure 4-30: Refrigerated Vending Machines (Canned Beverage) - Weekly Sales Breakdown

Sources: Industry interviews, Vending Times, ADL estimates

Table 4-34: Commercial Sector Overview - Refrigerated Vending Machine Consumption

Unit Type	Estimated Inventory	Unit Energy Consumption, kWh/yr	Total Energy Consumption, TWh/yr	Total Energy Consumption, %
Canned Beverage	2,400,000	3,000	7	59
Bottled Beverage	1,125,000	3,000	3	27
Other	575,000	3,000	2	14
Total	**4,100,000**	**-**	**12**	**100**

Sources: Vending Times, August 1994; ADL estimates

4.6.1 Equipment Description and Illustration

The purpose of a canned beverage vending machine is to hold a large quantity of cold canned beverages and vend them in exchange for currency

Since vending machines are evaluated primarily on sales enhancement, they must:

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

Bottling companies (Coca-Cola, Pepsi, etc.) purchase vending machines directly from the manufacturer. They will often dictate 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.

Figure 4-31 shows a typical canned beverage vending machine. Its capacity is about 400 12 oz. cans. The case is typically insulated to R-10 with 1.25 inches of blown polyurethane foam.

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

The unit is equipped with two T-12 fluorescent lamps (1-1/2 inch diameter tubes)on one ballast to illuminate the logo. The most common lighting configuration is two 40-watt bulbs.

Figure 4-31: Refrigerated Vending Machine (Canned Beverage) - Equipment Illustration

Source: Dixie-Narco

Table 4-35 summarizes the physical characteristics of the canned beverage vendor.

Table 4-35: Refrigerated Vending Machine (Canned Beverage) - Refrigerated Cabinet Description

Overall Exterior Dimensions			Insulation		Capacity
W (in.)	D (in.)	H (in.)	Thick-ness (in.)	R-Value per inch (ft^2.°F/ Btuh)	Number of Cans
37	26	72	1.25	8	400

Source: Dixie-Narco

The refrigeration system is packaged as a modular unit and is located in the bottom rear section of the vending machine. This configuration has the following benefits:

- Aesthetic reasons. The vending machine has its logo prominently displayed to enhance sales. Unsightly access panels would lessen the overall merchandising capability of the machine.
- Overall sizing constraints.
- Zoned refrigeration. Most of the refrigeration is directed toward the soon-to-be-vended products in the lower section of the cabinet.

The refrigeration system components consist of a 1/3 hp hermetic compressor, one evaporator fan and one condenser fan. Refrigerant flow is governed by a capillary flow restrictor. All fans are equipped with shaded-pole motors. In the past, CFC-12 was the refrigerant used by most manufacturers. Today, most units are manufactured with HFC-134a.

4.6.2 Energy Consumption

Table 4-36 summarizes performance data for the compressor and the associated design temperature data of a typical refrigerated vending machine. The evaporator temperature is typically 20°F and the condenser temperature is usually a 20°F over ambient. The compressor efficiency at the listed condition is 48%. This compares with efficiencies in the mid 50's, which are achieved with good residential refrigerator compressors.

Table 4-36: Refrigerated Vending Machine (Canned Beverage) - Refrigeration Component Description

Compressor				Temperatures			
HP	Type	Capacity (Btuh)	Power Draw (W)	Cabinet (°F)	Evaporator (°F)	Ambient (°F)	Condensing (°F)
1/3	Hermetic	2,500	425	35	20	100	120

Sources: DIxie-Narco, Tecumseh compressor data

The self-contained equipment refrigeration circuit shown in Figure 4-16 in Section 4.2.1 is typical for refrigerated vending machines.

Table 4-37 shows the steady state thermal load breakdown for a typical 400-can beverage vending machine. The compressor capacity is much higher than the steady state load because of the need for quick pull-down of beverage temperatures.

Table 4-37: Vending Machine Refrigeration Load Breakdown

	Load (Btu/hr)
Evaporator Fans	126
Wall Losses	253
Total	487
Compressor Capacity	2,500

Table 4-38 shows the energy consumption breakdown for a typical 400-can beverage vending machine.

Table 4-38: Refrigerated Vending Machine (Canned Beverage) - Energy Consumption Breakdown

Component	Power Consumption, W	Duty Cycle, %	Energy Consumption, kWh/yr	Energy Consumption[6], %
Compressor	425[1]	35[2]	1303	47
Evaporator Fan	37	100	324	12
Condenser Fan	37	35[2, 3]	113	4
Lighting	117-163[4]	100	1,022-1,424	37
Dispensing Mechanism	120	~ 0[5]	1	-
Total	**-**	**-**	**2,763-3,165**	**100.0**

1 Nominal power draw (1/3 hp compressor). Actual compressor power draw varies.
2 Manufacturer estimated of duty cycle based on a 70°F ambient temperature plus 10% for pulldown.
3 Condenser fan cycles with the compressor.
4 Range based on machines with standard T12 lighting @ 2.8 kWh/day and machines with high-output lighting @ 3.9 kWh/day.
5 The dispensing mechanism operates about 2 seconds/vend. At an average of 190 vends/week, this translates to only 5.5 hours of total annual run time.
6 Assuming standard lighting wattages

4.6.3 Manufacturing, Purchase, and Installation Costs

The typical unit purchase price for a canned beverage vending machine is approximately $1,700 in a purchase quantity of 100 units or less (Dixie-Narco price lists, personal communication with Dixie-Narco).

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 past its typical 5-year warranty, a replacement system costs about $400 to the manufacturer (personal communication with Pepsico, Inc.).

4.6.4 Life, Reliability, and Maintenance Characteristics

The typical life of a canned beverage vending machine is 7 to 10 years. During this time, it is refurbished at least once but probably twice in a refurbishing center run by the bottling company. There is usually no refurbishing done to the refrigeration system since the machine must be in working order to be worthy of refurbishing. Most of the modifications made are for cosmetic purposes (new logo, lighting, etc.).

Most manufacturers provide a packaged refrigeration system. If a serviceman discovers a problem in the refrigeration loop, the old system is replaced with a new operating 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. Since the typical life of a vending machine is only 7 to 10 years, it is often not worth replacing its refrigeration system in case of failure.

Regular maintenance for refrigerated vending machines consists of keeping the condenser coil clean. Lamps are replaced when necessary.

4.6.5 Major Manufacturers
According to industry representatives, the canned beverage vending machine equipment market is dominated by three manufacturers, with Dixie-Narco holding approximately 40 - 45% market share (Figure 4-32).

There are approximately 270,000 units shipped per year (Pepsi Co., Dixie Narco).

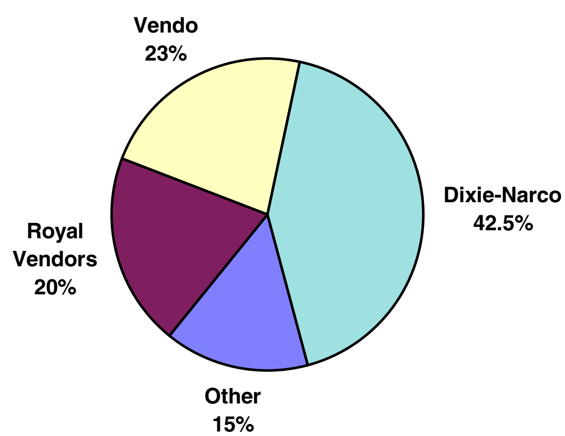


Figure 4-32: Refrigerated Vending Machines (Canned Beverage) - Estimated Market Share

Source: Pepsi Co., Dixie-Narco (Ref. 13 & 14)

4.6.6 Major End-Users
The major bottling companies that purchase canned beverage vending machines are:

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

4.7 Walk-In Coolers and Freezers

Walk-in coolers and freezers are room-sized insulated compartments which are refrigerated. As the name implies, walk-ins have an access door large enough for entry of people. They are used primarily for refrigerated storage of food and non-food items, but are also used to cool products which enter the walk-in at room temperature (or higher), and in some cases are also used to house small-scale food processing operations.

Walk-in coolers are also known as walk-in refrigerators. Both walk-in coolers and freezers are sometimes referred to as cool rooms.

Walk-ins are used in various segments of the food sales and food service markets, as well as a variety of non-food applications. A partial list of establishments which use walk-ins is as follows:

- Restaurants (fast food and sit-down; chain and independent)
- Convenience Stores
- Cafeterias (in schools, hospitals, prisons, factories, etc.)
- Food Wholesalers
- Produce and Fruit Farms
- Small Meat Packagers
- Small Ice Cream Companies
- Florists
- Research Laboratories
- Warehouses

Supermarkets also use walk-ins, for temporary storage of food prior to transfer to display cases. This market segment has been discussed in Section 4.1. In supermarkets, walk-ins are generally served by larger central refrigeration systems, which also serve display case circuits. In contrast, the walk-ins discussed in this section generally have dedicated refrigeration systems.

Some manufacturers of walk-ins also sell large refrigerated warehouses which are constructed of walk-in components. Such a warehouse is generally constructed of insulated panels supported by steel beam framing. The warehouse would be refrigerated by a large number of small self-contained systems mounted on the roof. Such warehouses represent roughly one-third of industry sales of walk-in refrigeration.[11] In spite of the importance of such business for the walk-in manufacturers involved in this

---

[11] Source: personal communication with Robert Gray, Jelason Products, 4/5/96

market, these structures represent a small percentage of the installed base of refrigerated warehouses, most of which are refrigerated with more complex central refrigeration systems, often using ammonia as the refrigerant. This section focuses on walk-in coolers and freezers, and does not provide an examination of warehouses constructed of similar components.

Walk-in coolers and freezers use either a packaged refrigeration system or a split refrigeration system. Packaged refrigeration systems consist of a manufactured package containing the condenser, compressor, evaporator, and controls. The system is mounted on the roof or wall of the walk-in such that the condenser has access to the outside of the compartment, and the evaporator has access to the inside

These walk-ins are often installed inside a building interior, such that heat rejection from the refrigeration system adds to the building heat gain. This practice reduces cost by simplifying the installation. This packaged style of installation is limited to systems with compressor size of at most 3 hp. Estimates by industry representatives of the percentage of walk-ins having heat rejection in the building interior range from less than ten percent to sixty percent.

Most larger walk-ins (requiring 3hp or larger compressors) use split refrigeration systems rather than packaged systems. A split system has the condensing unit (which consists of the compressor and the condenser) separated from the unit cooler, which is an evaporator with fans and an expansion device. The condensing unit is usually located on the building rooftop or at ground-level outdoors. In this configuration, condenser heat and noise can be kept outdoors rather than in the building interior. This arrangement is more efficient because lower condensing temperatures are possible, due to lower average ambient temperatures, and the arrangement reduces heat gain to the building interior itself.

Based on inventory estimates from NAFEM, Food Management and one manufacturer, there is an estimated installed base of about 880,000 walk-in coolers, freezers, and combination cooler-freezers. Annual sales of walk-ins have been estimated to be 30,000 units per year[12]. The market value for walk-in sales in 1994 has been estimated as $440 million[13]. Annual sales of walk-in refrigeration systems are higher than sales of the walk-ins themselves since replacement systems and components are also needed for existing walk-ins (see discussion of equipment life in Section 4.7.4). Figure 4-33 and Figure 4-34 show the walk-in inventory breakdown for the commercial sector by temperature level and by end user category. The “other mercantile” category includes package stores, florists, pharmacies, etc.

---

[12] Source: personal communication with Jim Aemmerling of Bally Enginered Structures, 1993;
[13] Source: The Freedonia Group, “The Market for Commercial Refrigeration Equipment in the U.S., February 1995

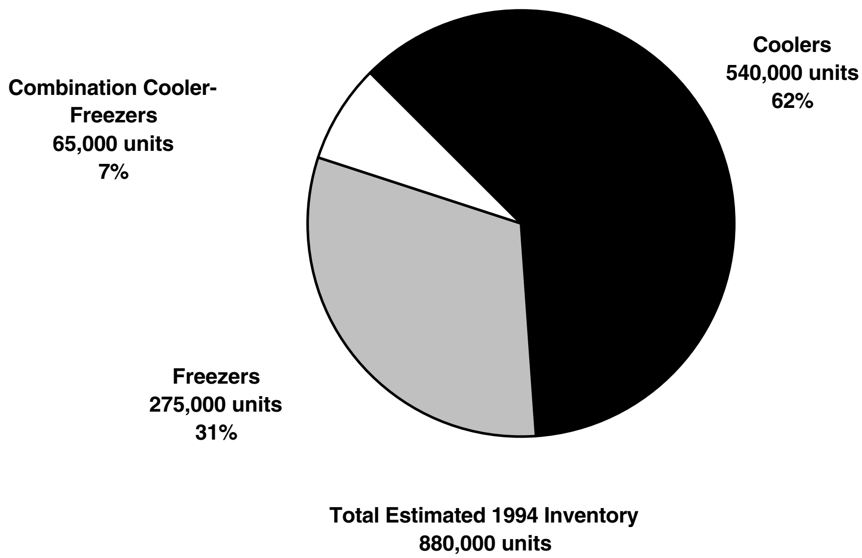


Figure 4-33: Walk-in Coolers and Freezers - 1994 Equipment Inventory

Sources: NAFEM, Food Management, ADL estimates

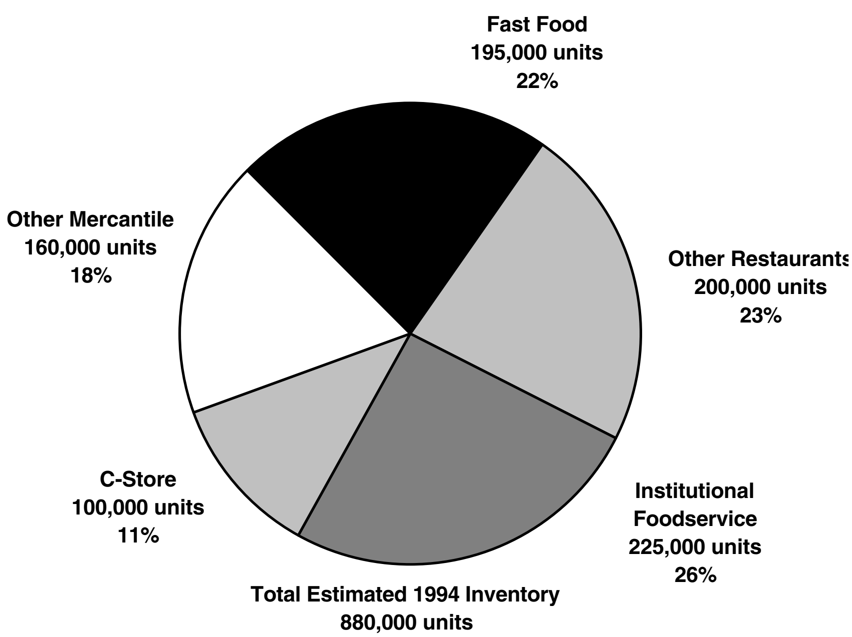


Figure 4-34: Walk-in Coolers and Freezers - End Use Applications

Sources: NAFEM, Food Management, ADL estimates

Based on an estimated inventory of 880,000 units, walk-in coolers and freezers consume approximately 16.5 TWh annually (see Table 4-39), which represents about 180 trillion Btu of primary energy.

Table 4-39: Commercial Sector Overview - Walk-in Coolers and Freezers

Unit Type	Estimated Inventory	Average Unit Energy Consumption [1], kWh/yr	Total Energy Consumption, TWh/yr	Total Energy Consumption, %
Coolers	540,000	16,200	8.8	53
Freezers	275,000	21,400	5.8	35
Cooler-Freezers	65,000	30,200	1.9	12
Total	**880,000**	**-**	**16.5**	**100**

[1] Includes compressor, fans, lighting, defrost, and antisweat

4.7.1 Equipment Description and Illustration

The purpose of walk-ins is for temporary storage of refrigerated or frozen food products for commercial food sales and commercial and institutional foodservice establishments, and storage of other perishable products, such as flowers.

Figure 4-35 shows the physical characteristics of a typical self-contained walk-in used in full-service restaurants. In contrast, a convenience store walk-in with glass doors for merchandising access is shown in Figure 4-36. Walk-ins generally have from 80 to 250 sq ft of floor area and have about 8 feet ceiling height. Most walk-ins are assembled on-site using prefabricated wall panels. Some smaller walk-ins, however, are shipped totally preassembled. Figure 4-35 shows the ceiling placement of the self-contained refrigeration systems used for many smaller walk-ins. A unit cooler and condensing unit which comprise the split system configuration is shown in Figure 4-37.

The typical compressor size for walk-ins ranges from 1 1/2hp to 5 hp. For example, for restaurants, the most common walk-ins are a 2 hp cooler and a 3 hp freezer. Walk-ins which are expected to cool or freeze products which enter at room temperature must have larger capacity. For instance, a 240 sq ft walk-in cooler serving a convenience store is likely to have a 5 hp compressor. The components of the refrigeration system for a walk-in cooler include a semi-hermetic compressor (some smaller systems use hermetic compressors), evaporator fans and condenser fans. Refrigerant flow is governed by a thermostatic expansion valve (some small units use capillary flow restrictors). Most units have fans with shaded-pole motors. The fans of high-efficiency units generally have PSC motors. In the past, CFC-12 and R-502 were the refrigerants used by most manufacturers. Today, many units are manufactured using HCFC-22, but the trend is toward the use of HFC blends such as R404A.

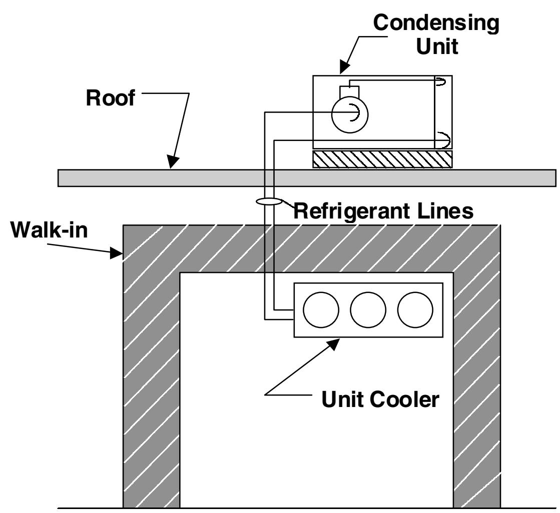
Walk-ins are typically insulated to R-27 with 3 to 4 inches of blown polyurethane foam. Some manufacturers use thicker walls for lower temperatures. Freezer floors generally require insulation in order to prevent freezing of the ground. The walls are generally constructed of galvanized steel. However, stainless steel and aluminum are also used, as is fiberglass-reinforced plastic.

There is at least one door on the walk-in. Walk-in doors are generally well-insulated and have durable gaskets. Antisweat heaters are used for access doors of walk-in freezers, but not coolers. Merchandising doors are frequently used along one wall of a walk-in for convenience store or package store applications. These doors, which face the store interior, (see Figure 4-36), are generally constructed of multilayered insulated glass. The arrangement allows easy restocking of the display shelves from within the walk-in. Merchandising doors used for walk-ins generally have antisweat heaters, even for medium temperature applications.

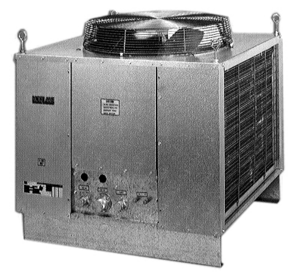
Figure 4-35: Walk-in Cooler/Freezer - Equipment Illustration

Figure 4-36: A Walk-in Cooler with Merchandising Doors

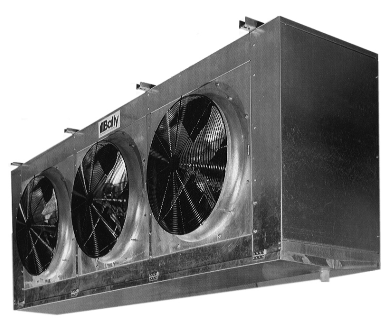
Source: Norlake



Schematic



Condensing Unit



Unit Cooler

Figure 4-37: Components of a Split Refrigeration System Serving a Walk-In

Source: Bally and Norlake

The interiors of most walk-ins are generally lit with one or more incandescent lights. Connected lighting power is typically about one watt per square foot of floorspace. Walk-ins with merchandising doors may also have fluorescent lighting for product illumination. This usually consists of a single row of T12 lamps, but can also involve mullion-mounted lamps.

4.7.2 Energy Consumption

Prototypical descriptions of two walk-in types were developed for presentation of energy usage characteristics and as a baseline for estimates of energy savings potential. The first prototypical walk-in is a storage-only freezer (no merchandising doors and no cool-down requirement) with a self-contained refrigeration system and interior heat rejection. The second is a convenience-store cooler with ten merchandising doors and a split refrigeration system sized for product cool-down capability and rejecting heat externally. These two system types were chosen in order to represent a range of walk-in concepts with two example systems. Data for the first system was provided by Norlake. Data for the second was provided by Bally.

Table 4-40 below presents the pertinent design data for the two prototypical walk-ins. The freezer has an 80 sq ft floor area and is cooled with a 1-1/2 hp semi-hermetic reciprocating compressor. The cooler is 24 feet long and 10 feet wide, and is served by a 5 hp Discus semi-hermetic compressor. Both systems have liquid-suction heat exchangers. These energy saving devices are used to enhance reliability (by preventing flow of liquid to the compressor) rather than efficiency.[14] Note that the refrigerant temperatures approach the internal and external air temperatures more closely for the split system cooler. The condenser temperature is 10ºF above ambient for the cooler and 23ºF above ambient for the freezer. The evaporator temperature is 10ºF lower than the walk-in temperature for the cooler and 16ºF lower for the freezer.

[14] Source: personal communication with Len Fritz of Bally, 8/10/95

Table 4-40: Characteristics of the Prototypical Freezer and Cooler

	Walk-in Freezer	Walk-in Display Cooler
Floor Size (ft^2)	80	240
Width (ft)	8	24
Depth (ft)	10	10
Height (ft)	7'7"	8'6"
Wall Thickness (in.)	4	4
Wall R-value	30	28.6
Merchandising Doors (ft)	-	(10) 2' x 6' 1 5/8"
Number of panes	-	2[1]
Access Doors (ft)	(1) 3' x 6' 6"	(1) 3' x 6' 6"
Refrigerant Type	404A	R-22
Compressor HP	1 1/2	5
Compressor Type	Semi-Hermetic Reciprocating	Semi- Hermetic Reciprocating (Copeland Discus)
Ambient Temperature (°F)[2]	90	95
Walk-in Temperature (°F)	-10	35
Condensing Temperature (°F)	113	105
Evaporating Temperature (°F)	-26	25
Compressor Capacity (kBtuh)	4.929	44.97
Compressor Power (W)	1445	3850
EER (Btuh/W)	3.41	11.7
Liquid Suction Heat Exchanger	Yes	Yes
Antisweat Wattage (W)	230[3]	300[4]
Antisweat 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	-

[1] Double pane insulated (inert gas) door.
[2] Actual Ambient Temperature varies - the reported temperature is the compressor design point
[3] Access Door Antisweat
[4] Merchandising Doors only

The energy consumption for the prototypical freezer and cooler are presented in Table 4-41 and Table 4-42 below.

Table 4-41: Storage-only Walk-in Freezer - Energy Consumption Breakdown

Component	Power Consumption (W)	Duty Cycle (%)	Energy Consumption (kwh/yr)	Energy Consumption (%)
Compressor	1445	70	8,861	57
Evaporator Fans (2)	180	100	1,577	10
Condenser Fan	329	70	2,017	13
Coil Defrost[1]	1500	4.2	552	3
Drip Pan Heater[1]	500	4.2	183	1
Antisweat Heater	230	100	2,015	13
Lighting	80	50	350	2
Total	-	-	15,555	100

[1] Operated for 60 minutes every 24 hours.

Table 4-42: Merchandising Walk-in Cooler - Energy Consumption Breakdown

Component	Power Consumption (W)	Duty Cycle (%)	Energy Consumption (kwh/yr)	Energy Consumption (%)
Compressor	3,850	66[1]	22,259	53
Evaporator Fans (8)	800	100	7,008	17
Condenser Fans (2)	1508	66	8,718	20
Antisweat Heater	300	100	2,628	6
Display Lighting[2]	236	66	1,364	3
Box Lighting	75	50	329	1
Total	-	-	42,306	100

[1] Sized to operate no more than 16 hrs/day, yet duty cycle is typically less.
[2] Display Lighting is provided by 4 60" T12 50W fluorescent lamp (w/ 2 lamps per ballast)

The steady-state refrigeration loads on the walk-ins are presented in Table 4-43 below. Note that the oversizing of the compressor as compared with the steady state load is much more dramatic for the cooler. This is because the cooler's compressor is sized for pulling down product temperature, and because of the need for added capacity during times of frequent merchandising door opening. A 240 sq ft cooler designed for maintaining temperature would have a 2 hp or 3 hp compressor.

Table 4-43: Walk-in Refrigeration Load Breakdown (Btu/Hr)

	Walk-in Freezer (80 ft^2)	Walk-in Display Cooler (240 ft^2)
Evaporator Fans	614	2,730
Coil Defrost	215	-
Pan Heater	71	-
Lighting	137	660
Wall Losses	1,103	walls: 1,270 merchandising doors: 4,146
Infiltration	150	420
Total	2,290	9,226
Compressor Capacity	4,929	44,970

Additional design details about the heat exchangers of the walk-ins is presented in Table 4-44 below. The heat exchangers are all of standard fin-tube construction. The cooler's evaporator is able to provide significantly more cooling per cfm than that of the freezer. This reflects the fin spacing restrictions for low-temperature evaporators, which must have sufficient spacing so that air flow is still acceptable when frost layers are thick. Condenser air flow is roughly 1,100 cfm per hp for the cooler. Both walk-ins have shaded-pole evaporator fan motors. The condenser fan motors are more efficient (efficient motor designs are more common for these larger motor sizes–1/6 and 1/2 hp as opposed to 1/20 and 1/40 hp for the evaporator fan motors). Fan blades are typically pressed aluminum.

Table 4-44: Prototypical Walk-in Heat Exchangers

	Walk-in Freezer	Walk-in Cooler
Evaporator		Note: Data for 1 of 2 evaporators
Face Area (in^2)	288	*
Air Flow (CFM)	1680	3200
Number of Fans	2	4
Fan Type	Propeller: 7", steel hub, pressed aluminum blades	Propeller: 12", steel hub, pressed aluminum blades
Fan Wattage (W)	90[1] each	100 each
Fan Motor Type	Shaded Pole (1/40 hp)	Shaded Pole (1/20 hp)
Condenser		
Face Area (in^2)	270	*
Air Flow (CFM)	1625	*
Number of Fans	1	2
Fan Type	Propeller: 18", steel hub, pressed aluminum blades	Propeller: steel hub, pressed aluminum blades
Fan Wattage (W)	329	530[1] each
Fan Motor Type	Capacitor Start Induction Run (1/6 hp)	PSC (1/2 hp)

[1]Based on the data of Table 5-2
*Data not available

4.7.3 Manufacturing, Purchase, and Installation Costs

The list price for a 10'x24' cooler with a floor and with merchandising doors is about $31,000. About $11,500 of this price represents the doors, lighting, and shelving for display along one of the 24-foot walls. The floor represents $3,000, and the refrigeration system represents about $7,000 of the cost.

A 10'x10' freezer with merchandising doors has a list price of about $22,000. As mentioned in Section 4.7.2, such a freezer would have an insulated floor. The merchandising doors represent about $5,500 of the price, and the refrigeration system represents $9,500 of the price.

The prototypical self-contained 8'x10' freezer discussed in this study has a list price of about $12,000.

End-users will generally pay from 60% to 90% of list price, depending on volume of purchase and relationship with the supplier.

Installation of a small walk-in with a self-contained refrigeration system can be completed by two men in a day. The installation cost would hence be in the range from $500 to $1,000. The installation of a large walk-in using a split system located at a distance from the walk-in would cost about five times as much, $2,500 to $5,000.

4.7.4 Life, Reliability, and Maintenance Characteristics

The expected life of the insulated box comprising the walk-in is from 12 to 25 years. The expected lifetime of a semi-hermetic compressor serving a commercial refrigeration application is 8 to 12 years[15]. Hence, the refrigeration system's compressor will likely be replaced once or twice during the walk-in's expected life. This may involve replacement of the entire condensing unit, but in the past, if the unit was otherwise in good condition, just the compressor would be replaced. Due to the recent phaseout of CFC's and the planned phaseout of HCFC's, there has been accelerated replacement of refrigeration equipment. Instead of replacing just a failed compressor, the entire condensing unit or the entire refrigeration system may be replaced.

Recommended maintenance for walk-ins includes monitoring of the refrigerant pressures to confirm that they are within normal ranges, and cleaning of heat exchanger surfaces of debris as necessary (external ground-mounted condensers are especially susceptible to getting clogged with leaves of other windblown material). In many cases, however, maintenance is done only when the system fails or is cooling inadequately.

---

[15] Source: personal communication with Robert Gray of Jelason Products, 4/5/96

4.7.5 Major Manufacturers

The market for walk-ins is fairly fragmented, with a large number of manufacturers, none having dominant market share.

In early 1994, the largest manufacturer of walk-ins was Bally, whose sales levels of $50 million annually represented about 10% of the walk-in market at the time. Bally filed for bankruptcy protection in 1994 and commenced a search for a buyer. Although the company restarted production in 1995, assets were split amongst two purchasers. Besides Bally, major manufacturers of walk-ins are Master-Bilt, the Shannon Group (which has two walk-in divisions, Kolpak and Tonka), Norlake, and the major supermarket refrigeration suppliers, especially Hussmann and Kysor Industrial. The balance of walk-in sales is made up by many smaller companies and also companies such as Traulsen, whose main source of revenue is in self-contained commercial refrigeration units.

4.7.6 Major End-Users

The major end-uses of walk-ins are identified in Figure 4-34 above. These are fast food restaurants, sit-down restaurants, institutional food service, convenience stores, and other mercantile applications (package stores, florists, pharmacies, etc.). The largest end-users are the major chains of fast food restaurants (i.e., McDonalds, Kentucky Fried Chicken, Burger King, Taco Bell, and Pizza Hut), large sit-down restaurant chains such as Dennys, and large convenience store chains (for instance Southland Corporation and Circle K).

Most large end-users purchase walk-in boxes and their refrigeration systems directly from the manufacturer. Other end-users purchase walk-ins through food sales and food service equipment dealers, and through refrigeration equipment wholesalers.

5. Energy-Saving Technologies

5.1 Supermarket Refrigeration

5.1.1 Energy-Saving Technologies - Supermarket Refrigeration

This section describes the energy saving technologies which are applicable to supermarket refrigeration systems, and their energy savings potential. Detailed calculation of the economics of the options is presented in Section 5.1.2.

Current Technologies

Evaporative Condensers

Heat rejection for most supermarkets is done with remotely located air-cooled condensers. Standard air-cooled condensers consist of fin-and-tube heat exchangers fitted with propeller fans (see Figure 5-1). The heat sink temperature for these condensers is the dry bulb temperature of the ambient air.

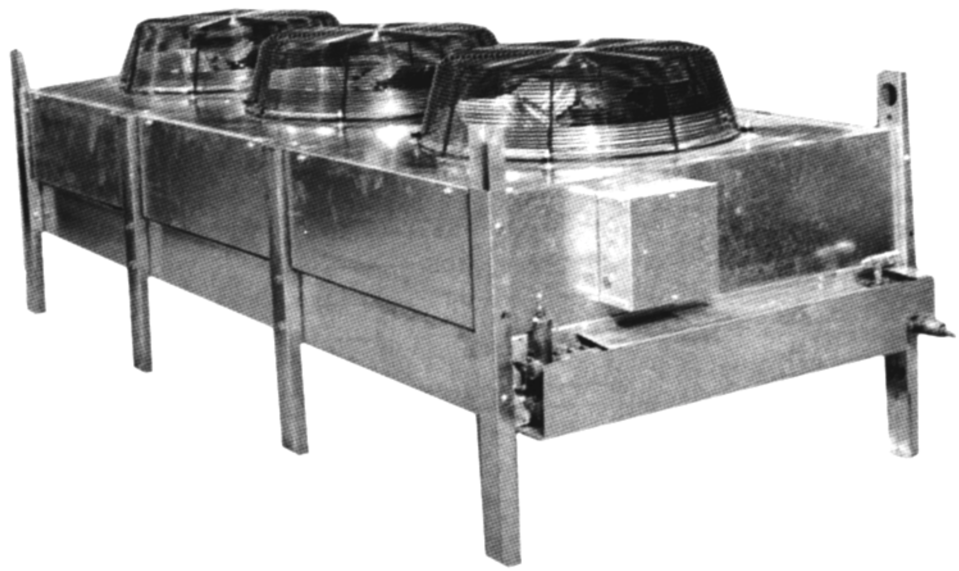


Figure 5-1: Air-Cooled Condenser

Source: Hussmann

Evaporative condensers consist of bare-tube copper-tube heat exchangers over which water is sprayed. Air is blown over the tubes from the bottom. Due to evaporation, the water temperature approaches the air's wet bulb temperature, which can be significantly lower than the dry bulb. As an example, the 1% design dry and wet bulb temperatures (temperatures which will be exceeded or equalled 1% of the time) for Boston are 90°F and 75°F. In drier climates, the difference is greater. For instance, in Phoenix the 1%

design temperatures are 109°F and 76°F. An industrial-grade evaporative condenser is shown in Figure 5-2.

Figure 5-2: Evaporative Condenser

Source: Imeco

The lower heat sink temperature of an evaporative condenser allows either a lower head pressure, resulting in improved efficiency, or a smaller condenser, resulting in lower system costs. Drawbacks to evaporative condensers are (1) the need for supply water to replace the evaporated and drained water flows, and (2) the need for chemical treatment. Water must be drained from the condenser in order to remove the makeup water's mineral content.

In the economic analysis of the next section, it is assumed that energy savings are generated by the use of evaporative condensers due to the possible reduction in head pressure.

Floating Head Pressure and Very Low Head Pressure

Traditionally, refrigeration systems were controlled to maintain constant head pressure. This can be done with a valve responding to pressure level or temperature which floods the condenser during times of low ambient temperature. Because of the reduced heat transfer surface available for condensation, a high pressure is maintained. The reason for such operation was that the system's throttling controls, such as thermostatic expansion valves, could operate properly only with sufficient driving pressure drop. Evaporator capacity would be limited by the valve throughput capability at low head pressure.

Today, expansion valves with balanced-port design allow more flexibility in head pressure control. The valves provide the proper refrigerant flow rate over a much wider range of pressure differential. Hence, operation with lower head pressure is possible.

In a fixed head pressure system, the condensing temperature is maintained above a minimum of 90°F to 95°F degrees. Floating head pressure systems allow condensing temperatures down to about 70°F. The so-called very low head pressure systems allow even further condenser temperature reductions, down to 50°F for medium temperature applications. For low temperature applications, the lower suction pressure allows further reductions, depending on the expansion valves used in the system.

The extreme condenser temperature reduction of the very low head pressure system requires additional complexity. It may be necessary to provide a refrigerant pump to boost refrigerant pressure prior to flow to the expansion valves. Also, the head pressure must be raised during defrost cycles to a pressure corresponding to 70°F condensing temperature if hot-gas defrost is used. Otherwise, the compressor discharge gas is not warm enough to adequately defrost the evaporator coils. Liquid lines must be insulated when pressure is low and the refrigerant liquid is cool to avoid heat gain from warm internal spaces. Otherwise, the refrigerant may flash, thus interfering with smooth operation of the expansion valves. Refrigeration capacity is also lost when the cool liquid is warmed during transfer to the display cases.

The savings possible with floating and very low head pressure are minimized in stores where the hot gas is used in heat reclaim coils for space heating. The low head pressures are possible during times of low ambient temperature when more space heating is required. Nevertheless, floating head pressure is fairly common, although very low head pressure is not. The economic analysis examines the savings potential of floating head pressure.

Ambient Subcooling

Ambient subcooling involves the use of an oversized condenser or an additional subcooling heat exchanger to subcool the condensed high-pressure refrigerant. Subcooling reduces the enthalpy of the liquid refrigerant, which is equal to the enthalpy of the two-phase stream of refrigerant leaving the expansion valve and entering the evaporator. The specific capacity of the refrigerant in Btu/lb is increased, hence reducing the required mass flow rate of refrigerant to be compressed, and the required compressor electric load.

Ambient subcooling is effective only when the head pressure control is preventing further reduction in head pressure. Otherwise, reduction in head pressure is more efficient than simply reducing the liquid temperature. Hence, the savings for ambient subcooling are generated during times of low ambient temperature, when head pressure is being maintained at a high level. This will represent a large percentage of operating hours in cooler climates.

Mechanical Subcooling

As with ambient subcooling, mechanical subcooling involves further reduction in enthalpy of the condensed liquid refrigerant. Mechanical subcooling is provided by expansion of part of the refrigerant liquid in a subcooling heat exchanger, as shown in Figure 5-3. The expanded refrigerant is compressed from an intermediate pressure to the common discharge pressure. The specific refrigerant capacity of the main stream of liquid is increased, thus reducing the compressor electric load for this stream of refrigerant. The additional electric load of the subcooling compressor must be subtracted from the potential savings. However, since the subcooling compressor has a higher suction pressure, its specific work requirement in kWh/lb of refrigerant is less. The result is an overall savings in electricity usage.

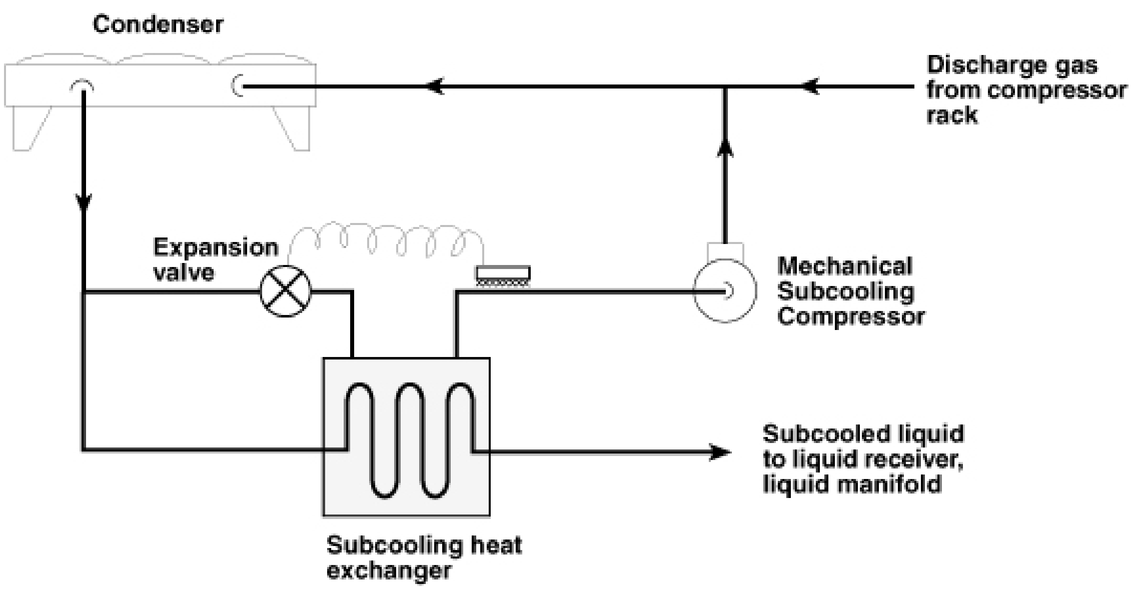


Figure 5-3: Mechanical Subcooling

Typical options for mechanical subcooling include installation of a dedicated subcooling compressor or the use of the medium temperature rack to provide subcooling for the low temperature rack.

Heat Reclaim

A refrigeration system incorporating heat reclaim is shown schematically in Figure 4-7 in Section 4.1 above. The heat reclaim coil mounted in the store's air handling unit is used during times when space heating or reheat is required. Reheat is used during dehumidification, which involves overcooling of the air stream to reduce its moisture content followed by reheating. The use of heat reclaim reduces the usage of fossil fuels for heating. Reclaimed heat can also be used for water heating.

Hot Gas Defrost

The water vapor which is removed from the air in the refrigerated space by an evaporator coil collects on the coil surface. Where evaporator surface temperatures are less than the freezing point, the water will freeze. The growing ice layer reduces cooling performance by increasing the thermal resistance to heat transfer and reducing air flow.

Evaporator coil frost can be removed in the following ways:

- ***Off-cycle*** defrost involves shutting off flow of refrigerant to the coil while leaving the evaporator fan running. This method is used where air temperatures are two or more degrees above the freezing point. The case air warms and melts the frost.
- ***Electric*** defrost is used where the air temperature is not high enough to defrost the coil, and where defrost must occur quickly in order to prevent any significant rise in case temperature.
- ***Hot Gas*** defrost involves the use of the hot compressor discharge gas to warm the evaporator from the refrigerant side. This method can be used for a large range of air temperatures. Electricity usage is reduced in comparison to the electric defrost method because available heat which would otherwise be rejected in the condenser is used. The hot gas defrost system requires more complicated piping and control than electric defrost. An additional drawback is the thermal stress inflicted upon the refrigerant piping by the alternating flow of hot and cold refrigerant. Recent trends in defrost are back towards electric defrost for this reason. Possible leaking caused by repeated thermal stressing of refrigerant piping can be quite costly due to today's high refrigerant prices.

Variable Speed Drives

The uneven parallel configuration of most state-of-the-art supermarket compressor racks is intended to improve part load performance and energy efficiency of the compressor plant. Originally single compressors served each refrigeration circuit, resulting in inefficient short cycling during times of low load. In a parallel arrangement, much greater turndown is possible.

Further improvements are claimed to be possible with the use of variable speed drives. When this technology is applied to a compressor rack, one of the rack's larger compressors is driven at variable speed. This compressor is controlled as the lead compressor, operating variably at all loadings to bridge the gaps between on-off control of the other compressors. It is claimed that significant savings are possible with such operation, but significant improvement over a well-designed and well-tuned uneven parallel system is not likely. Less than five percent of supermarket systems being sold involve variable speed drives.

Economic analysis for this technology is not presented.

Liquid-Suction Heat Exchanger

This measure involves installation of heat exchangers for cooling of the liquid flow to an expansion valve by the suction gas leaving the evaporator (see Figure 5-4). The heat exchanger provides additional subcooling for the entering liquid by further superheating the suction vapor. Heat gains to the suction vapor in the return piping to the compressor rack are also reduced. The compressor work is increased because the suction vapor has greater enthalpy. The potential gains depend on the refrigerant and the system pressure levels. The savings of the device may also be balanced by the additional pressure drop on the suction side of the heat exchanger.

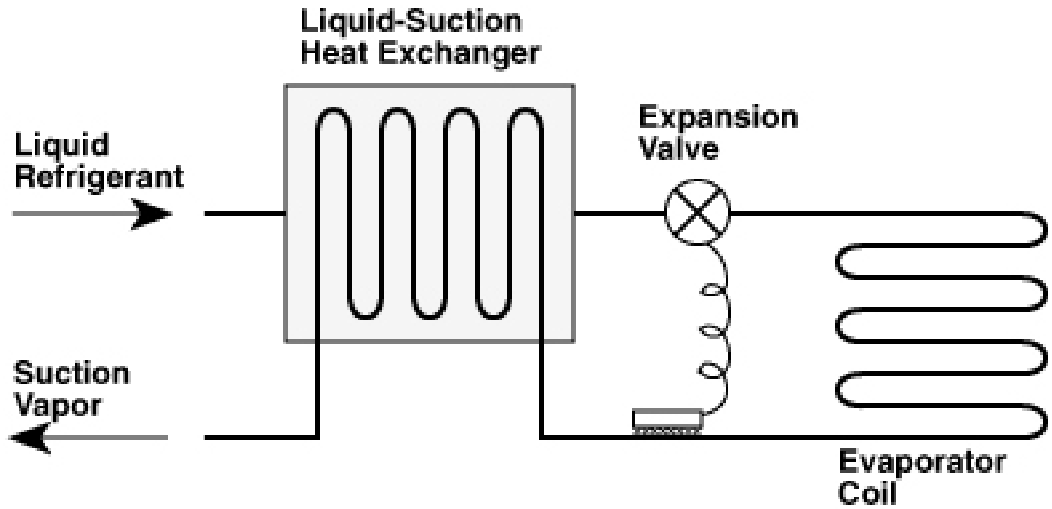


Figure 5-4: Liquid-Suction Heat Exchanger

Another drawback for the device is that the increased suction temperature results in higher discharge temperatures. In some situations, use of these heat exchangers is limited by the possibility of compressor over-heating problems. The possibility for such problems depends on the evaporator temperature level, the refrigerant, and other system-related factors such as forced cooling of the compressor.

Antisweat Heater Controls

Antisweat heaters are required to prevent condensation of moisture on external surfaces of display cases which are below the dew point of the surrounding air. Door gaskets of freezer cases are the most typical example of surfaces needing antisweat heating. The heaters prevent condensation and subsequent freezing of the gasket. In many cases antisweat heaters are simply energized at all times. Control of antisweat heaters requires measurement of the local dewpoint or humidity level. The heaters can be turned on when a given dewpoint temperature is exceeded, or the heaters can be cycled, with on-time increasing with dewpoint. Dewpoint sensors can be factory-installed in individual cases. More economical, however, is field installation of a single sensor and controller for a case lineup (a row of 2 to 4 cases). In the latter situation, care must be taken with sensor location so that cases located in more humid areas have adequate antisweat heating.

New Technologies

Lighting: Electronic Ballasts

For many refrigerated display cases, lighting represents a significant fraction of the total energy consumption. Two types of cases which use a significant amount of lighting are open multi-deck cases and closed reach-in cases.

Since supermarkets have differing views on how lighting enhances food sales, lighting configurations in these display cases can vary significantly depending on a particular supermarket's specifications. Table 5-1 shows various possible lighting configurations and their associated power consumptions for an open multi-deck display case.

Most lighted display cases in supermarkets use fluorescent lighting with magnetic ballasts. T12 fluorescent lamps of various lengths are used depending on the case size. A row of lighting in a 12' display case will typically consist of either three 48" lamps or two 72" lamps. Supermarkets will specify high-output lighting (HO, 800 mA) or very-high-output lighting (VHO, 1500 mA) as part of a lighting system to enhance sales in certain display cases.

Since lighting systems among refrigerated display cases are so diverse, only energy-saving technologies which would have the greatest overall impact in supermarkets in general should be considered. It is recommended that electronic ballasts be considered as a basic energy-saving option over standard magnetic ballasts.

Table 5-1:   Lighting Power Consumption by Lighting Configuration - Multi-Deck Display Case

Canopy First Row			Canopy Second Row			Bottom Ledge (One Row)			Shelves (4) (Four Rows)			Power Consumption, W per linear ft of case
VHO	HO	T12	VHO	HO	T12	VHO	HO	T12	VHO	HO	T12	
		x										12
	x											18
		x						x				24
		x			x			x				34
		x			x			x			x	80
	x			x			x				x	100
x			x			x					x	140

Note: One row of fluorescent lighting corresponds to three 48" lamps with magnetic ballasts.
VHO: Very High Output
HO: High Output
T12: Standard output, 11/2" diameter fluorescent

High Efficiency Fan Motors

Most fan motors used in commercial refrigeration applications are inexpensive and inefficient single-phase shaded pole motors. The efficiency of permanent split capacitor (PSC) or electronically commutated (ECM) motors is significantly better. Estimates of costs and power requirements for these three motor types are compared in Table 5-2 below. Typical wattages and OEM costs are listed. Not all the listed ECM motor sizes are currently being offered by manufacturers. GE currently offers 4W, 9W, 16W, 1/3 hp, and 1/2 hp output fan motors. Nevertheless, the technology can be applied to intermediate sizes, which is in fact currently under consideration.

The table compares a number of motor sizes, including the three most common sizes of evaporator fan motor (6, 9, and 25-Watt output). The table also lists numbers of the fans for a typical supermarket as reported in Reference 4 (Energy International, 1994).

High Efficiency Fan Blades

The evaporator fans typically used in supermarket display cases have sheet metal blades with diameters in the range of 6 to 10 inches. The blades are supplied by a fan blade manufacturer and mounted to the motor by the equipment manufacturer. Economics of fan blade design and manufacture favor large production numbers in order to minimize production costs. For this reason, fan blades are usually used in a range of applications, not all of which are optimum for the blade design. The OEM fan blade cost for a quantity of thousands of blades (6-10 inch size) is in the range $1 to $1.25. The typical fan efficiency for an axial flow sheet metal fan is 40% when mounted in a test rig (the flow path contortions typical for refrigeration equipment will result in reduced efficiencies). Evaporator fans may have lower efficiencies due to the higher required pressure drops, for which sheet metal fans are poorly suited.

Table 5-2: Shaded-Pole and High Efficiency Fan Motors

Motor Output (W)	Shaded Pole		PSC		ECM		Number per Supermarket[1]
	Wattage	OEM Cost	Wattage	OEM Cost	Wattage	OEM Cost	
6	40	$7	15	$25	8.5	$35	67
9	53	$10	21	$28	12.5	$40	86
15	75	$15	33	$33	20.5	$42	
20	90	$20	42	$35	27	$45	
25	110	$25	51	$37	33	$48	31
37 (1/20hp)		$30	70	$40	49	$52	
50 (1/15hp)			90	$43	65	$54	
125 (1/6hp)			202	$51	155	$64	
249 (1/3hp)			370	$57	304	$71	
373 (1/2hp)			530	$60	450	$75	

Source: ADL estimates based on discussions with motor manufacturers (OEM cost at commercial refrigeration industry component purchase volumes)

[1] **Source: "Assessment of Refrigerated Display Cases", EPRI Report, May 1994**

Required fan shaft power could be reduced about 10 to 20 percent if the fan blade were optimized for each given application. This would result in higher fan blade prices. The production numbers per blade shape would be reduced, thus increasing tooling costs. The blade manufacturer and the equipment manufacturer would have to invest more engineering time. Tooling costs for a short run (20,000 to 50,000 units) for the 6 to 10 inch blades would cost between $12,000 and $20,000. The higher cost would apply to plastic blades. The trend in the industry is to share tooling costs with the OEM suppliers. One fan blade manufacturer indicated that retooling costs would be entirely paid for by the OEM. Hence, the OEM cost for a blade design of which 20,000 units are required could double. Engineering and inventory costs for the OEM would also increase. In spite of cost additions, improvement in fan blades does reduce energy usage enough to be worth consideration. Savings and cost estimates are detailed in Section 5.1.2.

Insulation

Typical insulation thickness for supermarket display cases ranges from 1.5 to 2 inches. Blow-in polyurethane foam is used for most cases. The impact of increases in insulation thickness and insulation quality is limited for open cases by the fact that a large portion of the cooling load is due to the opening. Space in all cases is tight, limiting the possible increases in insulation thickness. The costs of increasing insulation thickness include added material costs (polyurethane and blowing agent), product redesign costs, and manufacturing plant retooling costs. Per square foot of wall area, the added OEM material costs for an additional inch of insulation thickness are about $0.33. This cost includes 0.27 oz of blowing agent at 8¢ per ounce and 2 oz of polyurethane at 16 ¢ per ounce. The costs per lineal foot of display case depend on the case type, but the cost averages $6.50/ft for the four case types considered in this study.

A quick analysis shows that increase in display case insulation thickness is not viable from an energy savings view unless reduced storage volume is accepted. Table 5-3 below summarizes case load reduction and internal volume reduction resulting from an increase in insulation thickness from 1½ to 2½ inches. The percentage of the direct case load (not including defrost, fan, or antisweat contributions) represented by wall losses is reported in Reference 5. The insulation thickness increase will reduce the wall load by about 38%. The table shows the load reductions as a percentage of the total case load. The range is from 1 to 3 percent. Percent volume reductions, also tabulated, are roughly 10%. Hence, the number of cases which would need to be added to a supermarket in order to maintain total case volume would greatly outweigh the energy savings potential.

Table 5-3: Insulation Thickness Increase: 1½" to 21/2"

			Load Reduction		
	Direct Case Load (Btu/hr/ft)[1]	Percent Wall Loss[2]	Btu/hr/ft	Percent of Total Case Load	Percent Volume Reduction
Multideck Meat (MT)	1366	3	16	1	8
Multideck Other (MT)	1438	3	16	1	8
Reach-In (LT)	352	13	17	3	9
Single-Level Open (LT)	446	9	15	3	11

[1] **From Table 4.1.5 - "Case Load Breakdown", Last Column plus lighting load**
[2] **Source: "An Electric Utility's Adventures in Commercial Refrigeration", Food Service Refrigeration, Flannick et. a.l, October 1994.**

Current standard-practice insulation has a conductivity of about 0.137 Btu/in /hr/ft^{2}F. Improved-technology polyurethane foam insulation which has a reduced conductivity of 0.120 Btu-in/hr ft^{2}F is also now available. The improvement is due mainly to the formation of smaller cells within the foam insulation structure and better cell-size consistency. Use of the better foam would reduce case wall load by 12%, assuming no wall thickness change. Implementation of the improved foaming technology requires the purchase of new foaming equipment, which can cost several million dollars. The added cost of the display cases would depend on production quantities and corporate policy for amortizing the initial investment cost.

Table 5-4 below presents the display case load reductions possible when improving the foam's thermal resistance. COP's of 2.5 for medium temperature and 1.3 for low temperature are assumed to convert case load to compressor load.

Table 5-4: Insulation Improvement

Case Type	Direct Load (Btu/hr/ft) [1]	Percent Wall Loss [2] (Btu/hr/ft)	Load Reduction [3] (Btu/hr/ft)	Case Inventory (ft)	Electricity Usage Savings (kWh/yr)	Electric Demand Savings (kW)
Multideck Meat (MT)	1366	3	5	120	380	0.07
Multideck Other (MT)	1438	3	5	260	870	0.16
Reach-In (LT)	352	13	5	268	1830	0.33
Single-Level Open (LT)	446	9	5	128	766	0.14

[1] From Table 4.1.5 "Case Load Breakdown" Last Column plus Lighting Load

[2] Source: Flannick et. al., "An Electric Utility's Adventures in Commercial Refrigeration," Heating, Piping, and Air-conditioning, October, 1994

[3] Reduction of Insulation Conductivity from 0.137 to 0.120 Btu-in/hr ft^{2}F.

The importance of case volume suggests that technologies which would allow reduction in insulation thickness while maintaining R-Value would be of interest. Vacuum panels could provide such performance. Vacuum panels are airtight panels sealed with glass or plastic which are evaluated to eliminate a conduction path. They are generally filled with supporting powder which prevents collapse of the external seal. Much work needs to be done to demonstrate the reliability of vacuum panels, to show that they will perform for many years without leaking and losing their insulating value.

Coil Improvements

It is possible that redesign of heat exchanger coil parameters (such as face area, air flow, refrigerant circuiting, and tube sizes) may reduce energy consumption. A simple analysis for these changes is presented below. In the analysis, it is assumed that coil area and air will increase, while face velocities remain the same.

It is assumed that the temperature differences in the evaporator coils between saturated refrigerant and entering air temperature are 20F. The assumed condenser approach temperatures are 15F for the medium temperature system and 10F for the low temperature system. Improved system COP's may be possible if these temperature differences could be reduced. Table 5-5 below shows system COP estimates for different values of approach temperature. The current assumptions are based on evaporator/condenser conditions of -25/110 for low temperature and 15/115 for medium temperature.

Table 5-5: System COP for various Coil Temperature Differences

	Condenser ΔT	Evaporator ΔT	
		20F	10F
Medium Temperature:	15	2.5*	2.9
	10	2.7	3.1
	5	2.9	3.4
Low Temperature	10	1.3*	1.6
	5	1.5	1.8

* current assumptions

A simple analysis of the benefits of heat exchanger improvement assumes that the flow rates of air and the size of the heat exchangers are increased by the same ratio. Temperature differences between air and refrigerant and also between air inlet and outlet will be decreased by the same ratio. These changes will result in zero net change in the heat transferred. The benefits of improved heat exchanger design must then be balanced with increases in fan power.

For the remote condenser, where the only flow resistance is represented by the heat exchanger, the fan power must increase by the same ratio as the air flow rate and area. Hence, reduction of condenser temperature difference from 10F to 5F for the low temperature system will require a doubling of fan power. The COP increases from 1.3 to 1.5, resulting in compressor power savings of 9 kW for the design condition of 300 mBh. In this prototypical system, the condenser fan power will also increase by about 9 kW for this change, negating the savings in compressor power. Similarly, increases in size of the medium temperature condenser are not warranted: a decrease in approach temperature from 15F to 10F would require an additional 6 kW of fan power while saving about 6.45 kW in compressor power. This analysis suggests that the condensers are sized with appropriate temperature differences. The opportunity for savings would be less at off-design conditions.

Improvement to evaporator coils will also likely require additional fan power. An accurate assessment of the improvement potential would require more detail than is available. A rough analysis assumes that fan power will double in order to sustain a 50% reduction in temperature difference. In addition, case volume is assumed to be reduced by 10% to make room for larger coils and larger air ducts. Table 5-6 below shows that the result of these changes is an increase in the total power requirement.

Table 5-6: Evaporator Coil Improvement

	Before				After			
	Case Load Total (mBh)[1]	Compressor Power (kW)[2]	Evap Fan Power (kW)	Sum Power (kW)	Case Load Total[3] (mBh)	Compressor Power [4] (kW)	Evap Fan Power (kW)	Sum Power (kW)
Multideck Meat (MT)	180	21.1	3.2	24.3	210	21.2	6.4	27.6
Multideck Other (MT)	390	45.7	3.3	49.0	441	44.6	6.6	51.2
Reach-In (LT)	150	33.8	5.4	39.2	185	33.9	10.8	44.7
Single-Level Open (LT)	70	I5.8	1.3	17.1	82	15.0	2.6	17.6

[1] From Table 4.1.5 in Section 4.1.1
[2] Assuming COP's of 2.5 for MT and 1.3 for LT
[3] Assuming additional fan power and 10% more cases to maintain constant overall storage volume.
[4] Assuming COP's of 2.9 for MT and 1.6 for LT

These analyses of coil improvements are somewhat simplistic in that increase in fan power was assumed to be required in order to increase heat transfer. A more thorough analysis would require knowledge of dimensional data, air and refrigerant flow rates, coil circuiting, etc. Also, the penalty for increasing air flow would be diminished if more efficient fan motors were installed. This design option may warrant some additional attention. It is not included in the economic analysis, however.

Defrost Control

Control of defrost involves (a) initiation of the defrost cycle and (b) termination of the cycle. In the past both were done with a timer. The cycle was started when it was expected that a large frost layer had developed, and the cycle duration was set long enough to ensure complete defrosting for worst-case situations (i.e., humid summer days with frequent case door openings).

Currently temperature-based termination of the cycle is accepted practice. This control simply shuts off the heating cycle when the coil temperature reaches a value indicating complete defrost. However, initiation of defrost still occurs based on a preset time schedule.

Demand defrost (controlled initiation) is not yet accepted practice. Two forms of this control involve (1) measurement of the air temperature drop across the coil and (2) detection of frost buildup with photocells. The first system works as follows. As the coil is collecting frost, the air flow drops more rapidly than the delivered cooling. Hence, there is an increase in the difference in temperature between the inlet and outlet air. Problems with the system are associated with possible reduction of airflow for reasons other than coil frosting: dust collection, evaporator fan problems, varying product fill levels, and external air flow disturbances. A rough estimate of the possible savings are that half of the defrosts during the six cooler months of the year (when store humidity is lower) could be eliminated. Only electricity usage savings are assumed.

The cost of demand defrost to the end user should be about $50 per case. This includes the cost of two temperature sensors and the necessary controller modifications. The control could also be approximated by adjusting timeclock settings in less humid months.

Advanced Technologies

Various advanced technologies for supermarket refrigeration are discussed below. Economic analysis is presented in Section 5.1.2 for one of these: Demand Defrost Control.

Alternative Refrigerants

Before the CFC phaseout, the most common refrigerants used for supermarket refrigeration were R-12 for medium temperature and R-502 for low temperature. In addition, a significant portion of systems were set up with R-22. Recently, the standard refrigerant for both temperature levels has been R-22, whose phaseout for use in new equipment is scheduled for the year 2010. The problem of high compressor discharge temperature with this refrigerant has been mitigated with proper system and component design. R-134A (an HFC) and blends of HFC's are now gradually being accepted as the appropriate long-term solution. The blends are mixtures of the refrigerants R-134a, R-125, R-143a, and/or R32. The compositions and applicable temperature levels are shown in Table 5-7 below.

Table 5-7: New Supermarket System Refrigerants

Designation	Supplier	Composition (wt. %)	Applicable Temperature Level(s)	Temperature Glide
R-507	Allied	50% R-125-50% R1432	Low, Med	0°F
R-407A	ICI	20% R-32-40% R-125 40% R-134a	Low, Med	9°F
R-407B	ICI	10% R-32-70% R-125 20% R-134a	Low, Med	6°F
R-404A	Dupont	44% R-125-52% R-143a-4% R-134a	Low, Med	1.5°F @ -20°F
R-134a	Several	100% R-134A	Med	0

An alternative non-chlorinated refrigerant is ammonia. This refrigerant is used for most industrial applications, and it is occasionally used in other situations. Ammonia has a thermodynamic efficiency roughly equal to that of the halocarbon refrigerants. It has better transport properties and is significantly less expensive than the alternative refrigerants, but it is toxic, flammable (in certain concentration ranges), and is corrosive. Because of its toxicity and its potential to spoil food, ammonia's use in supermarkets would require the use of a secondary refrigerant to transfer the cooling load heat to a closed and isolated ammonia system (see Figure 5-5). The figure shows a secondary circuit using a halocarbon refrigerant. This would require less display case modification than use of glycol or brine.

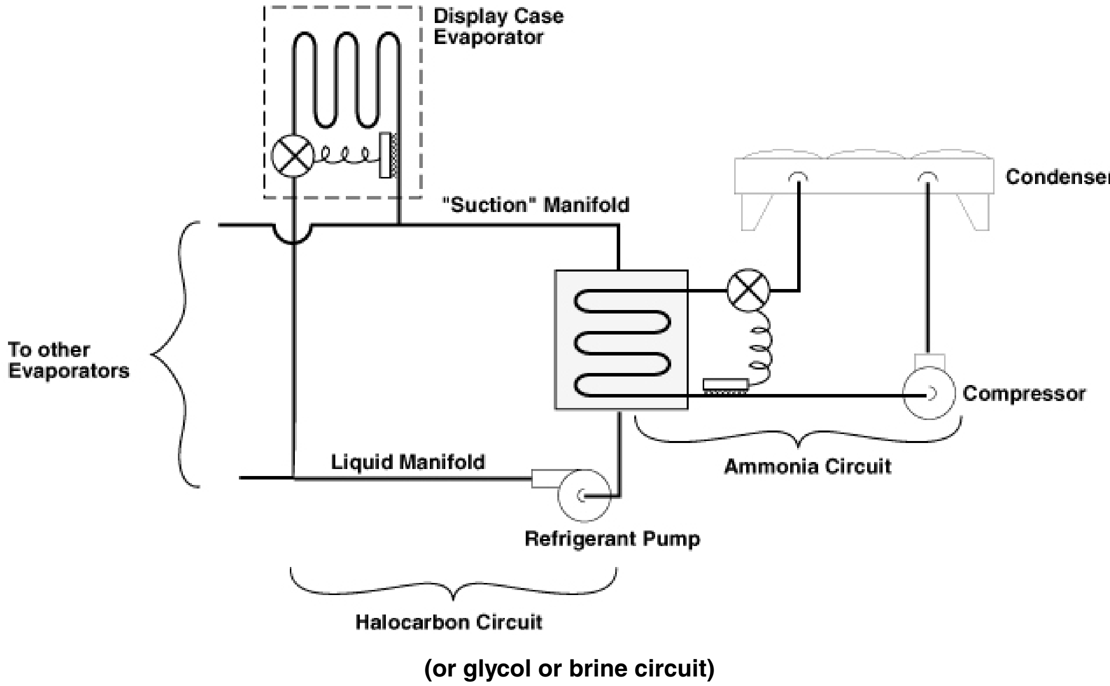


Figure 5-5: Refrigeration System for Ammonia with Secondary Refrigeration Circuit

Propane and iso-butane are additional non-CFC refrigerants which are fairly efficient and inexpensive. Their major drawback is high flammability. Propane is used in residential refrigerators in Europe but has not found acceptance in the U.S. except in limited industrial applications.

Figure 5-6 and Figure 5-7 show the thermodynamic cycle efficiencies of a number of alternative refrigerants for two temperature conditions: -25/110 (saturated suction/saturated discharge temperatures) for low temperature and 15/115 for medium temperature. For ammonia, propane, and iso-butane, evaporator temperatures ten degrees lower are assumed in order to account for the additional temperature difference in the primary/secondary refrigerant heat exchanger. In all calculations, the evaporator exit superheat is assumed to be 10F and the condenser exit subcooling is zero. The compressor efficiency is assumed to be 75%. The analysis does not take the effects of transport property differences into consideration. Pumping power for the secondary refrigerants is also not considered.

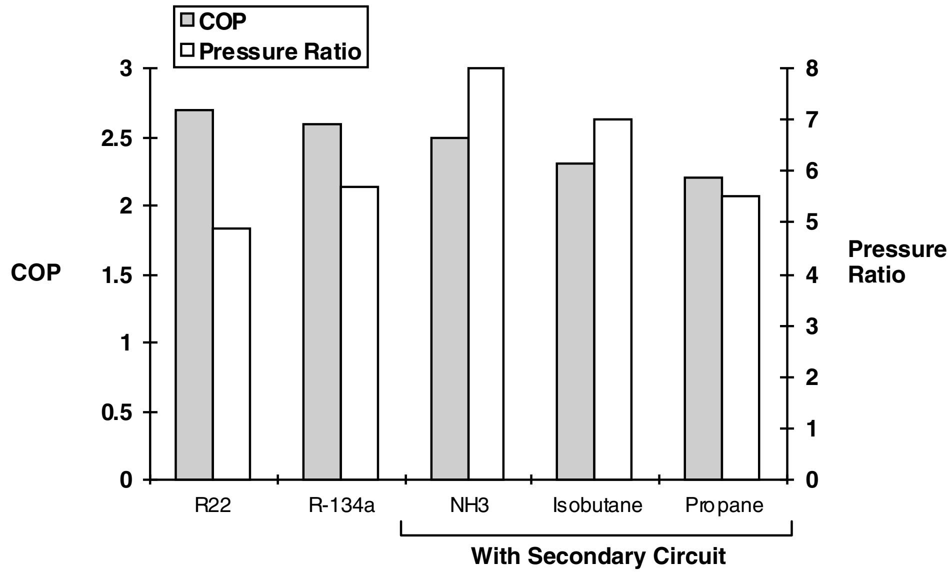


Figure 5-6: Refrigerant COP's -- M edium Temperature

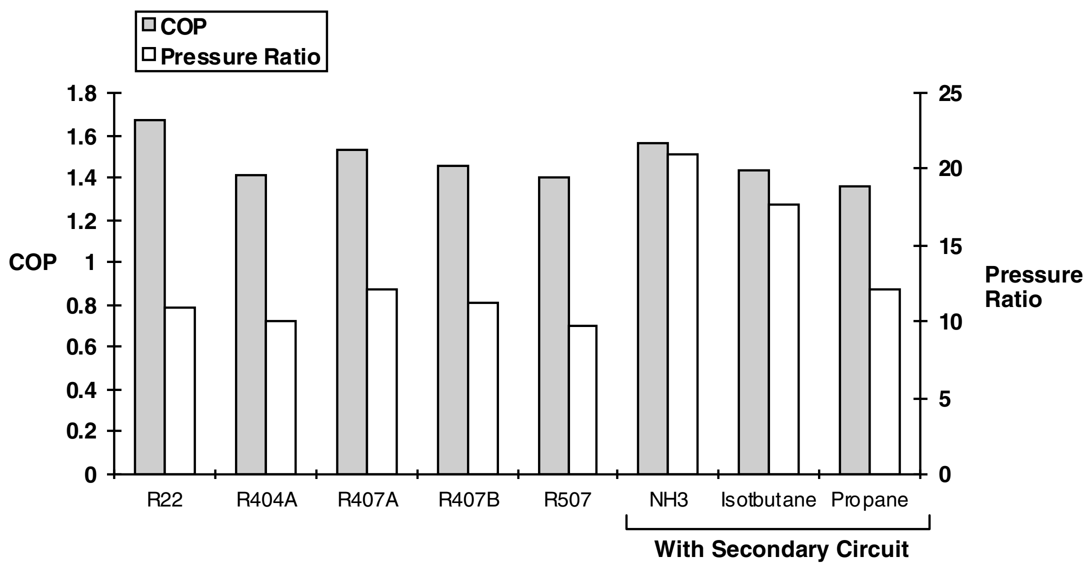


Figure 5-7: Refrigerant COP's -- Low Temperature

The best COP for both temperature levels is calculated for HCFC-22. Ammonia has the best COP of the refrigerants requiring a secondary circuit. However, because of the additional temperature difference on the evaporator side, little or no energy savings can be expected with the use of ammonia in comparison with the HFC-based refrigerants. In addition, the high pressure ratio and extremely high discharge temperatures (392°F for medium temperature and 528°F for low temperature) for this simplistic cycle for ammonia indicate that alternative equipment or a somewhat more complex cycle is required: multiple compression with intercooling, a reciprocating compressor with water-cooled cylinder heads, or a screw compressor with oil cooling of the refrigerant. Multistage compression or liquid injection is also required for HCFC-22 for low temperature duty.

The possibility of better performance with an alternative refrigerant is by no means ruled out. A thorough assessment would require design of cycles appropriate for each refrigerant individually. The analysis presented does, however, indicate that there is no replacement refrigerant which is an obvious choice for improvement.

Refrigeration systems using ammonia may potentially be the most attractive for reducing refrigeration energy use. As mentioned, a realistic analysis of this refrigerant's potential would require a somewhat more detailed analysis. First, an adequate secondary refrigerant must be identified. The limited number of field tests using ammonia have used liquid secondary refrigerants (glycol or brine) and have focused only on medium temperature applications. The use of halocarbons or carbon dioxide as a secondary refrigerant, using phase change to transfer the cooling effect, deserves some consideration. Also, the improvements represented by ammonia's exceptional transport properties and the various energy-saving technologies developed in the industrial refrigeration sector (where ammonia is the dominant refrigerant) should be examined.

Engine-Driven Compressors

One option for reducing primary energy usage is the use of engine-driven compressors rather than the conventional electric-motor-driven compressors. This step would require reconfiguration of the entire system, for the following reasons.

1) Engine-driven equipment requires the use of open-drive compressors rather than the standard semi-hermetic construction used for most supermarket equipment.

2) The current trend is for systems with a fairly large number of small compressors so that part loads can be handled efficiently. Such design is not practical for engine-driven systems because installation and maintenance costs will strongly favor larger engines. The least expensive engines for industrial applications are automotive-derivative; these engines have power output in the 50 to 150 hp range. However, the variable-speed capability of engines reduces the need for many small compressors.

3) Reliability of engines has improved over the years as a result of intensive research and development by the Gas Research Institute and other organizations. However, engine reliability is not 100%, and any engine installation in supermarkets would require enough redundancy to assure continued operation in the event of engine failure. Hybrid systems combining electric motor and engine drives is the logical choice. System complexity and control complexity would increase, especially since engine operation is economical in many cases only when the engine is operated in a peak shaving mode.

Hussmann has developed a prototype engine-driven refrigeration system in conjunction with Rotary Power International (RPI). The system uses an RPI 75-hp rotary engine, the major parts for which are obtained from Mazda. The compressor is an open-drive screw compressor manufactured by Bitzer. One such system is currently being operated in a test facility in a Big Y supermarket in Springfield, MA. Other supermarket chains have also expressed interest in testing the system. The technology has received marketing support from the gas industry, and is supported in some regions by incentive programs by gas and/or combination gas/electric utilities.

Energy savings with the use of engine drives depend on the efficiency of conversion of primary energy to shaft power of the engine and of the electric power system. The prototype Hussmann compressor unit has a primary energy efficiency of 23%. This compares with an efficiency range of 25-28% (includes production and distribution losses) for electric power production and the use of an electric motor. This represents a heat rate of 10,867 Btu/kWh and a motor efficiency from 80 to 90 percent. Additional investigation would be required to determine if an alternative engine which is suitable for operation of supermarket compressors has better efficiency. A maximum efficiency for such an engine would be in the 27-30% range. Hence, gas engine-driven refrigeration is not likely to reduce primary energy significantly, unless heat recovery is used to increase utilization of the input.

Additional savings can be generated if waste heat from the engine exhaust and engine cooling system can be used for space or water heating. This heat could also be used to operate an absorption cycle which can provide either space cooling or liquid refrigerant subcooling, in the latter case improving the refrigeration system performance. Detailed analysis of an engine system with heat recovery is beyond the scope of this study.

Absorption

Absorption refrigeration is another option for shifting refrigeration load to direct on-site use of fossil fuels from electricity. Current single-stage absorption technology would not save primary energy in comparison with electric-drive refrigeration. Development of advanced ammonia-water cycles with COP's in the range of 0.5 to 0.6 (based on fuel high heating value) for low temperature refrigeration is underway. The vapor-exchange generator-absorber heat exchange (VX-GAX) system is being developed by Energy Concepts with funding from the DOE/Oakridge Laboratory high temperature lift absorption heat pump program. This cycle is based on the ammonia/water pair and is a variant of the GAX cycle being considered for residential heat pumps. Funding for the Acurex system has been secured for conducting field tests of the concept. This system uses ammonia as refrigerant with an ammonia-based solution of sodium thiocyanate (NaSCN) and sodium iodide (NaI) as the sorbent. The program is being supported the Natural Gas Consortium, which includes a group of gas utilities and the Gas Research Institute (GRI), and also by Oak Ridge laboratory. Initial work was supported by GRI.

COP's in the range 0.5 to 0.6 for an absorption machine would make direct-fired absorption competitive with electricity-driven systems on a primary energy usage basis.

It is not likely, however, that these absorption systems will use significantly less primary energy than electric systems. The COP projections are based on laboratory tests which must be repeated in the field to increase confidence in the viability of the systems. The absorption cycles use ammonia as the refrigerant and would for this reason require a secondary refrigerant loop. This will reduce achievable COP levels and increase first costs. Further development of the technology is required in order to accurately assess its potential in increasing refrigeration efficiency.

Chemisorption

Chemisorption is a noncontinuous sorption process involving a solid sorbent to which the refrigerant (ammonia) can be alternately adsorbed and desorbed, depending on the sorbent temperature. The sorbent alternates between operation as an "absorber" at ambient temperature and a "desorber" at high temperature, thus emulating the absorption refrigeration process.

Because the refrigerant for the chemisorption process is ammonia, a secondary circuit would be required for its use in supermarkets. The cycle COP (based on delivered heat) for a single-stage low temperature system using a secondary refrigerant loop is in the range from 0.3 to 0.35. This would be a COP based on fuel high heating value of up to 0.29 if a burner with 82% efficiency is used. Although less efficient than the advanced absorption cycle, system costs are likely to be less. Two-stage chemisorption systems could also be developed. These might have cycle COP's in the range from 0.5 to 0.6.

Comparison of the gas-fired systems and a conventional electric system is presented in Table 5-8.

Table 5-8: Comparison of Gas-Fired Refrigeraiton Options

	Electric Use[1] (kW/ton)	Equipment Gas Use (mBtu per ton-hr)	Primary Energy Use[2] (mBtu/ton-hr)	Primary Energy COP
Electric	2.3	0	25	0.48
Gas Engine	0.25	22	25	0.48
Chemisorption[4]	0.56	42[3]	48[3]	0.25[3]
Advanced Absorption	0.46	21[3]	26	0.46

Supermarket Low Temperature: -20°F Evaporator; 110°F Condenser
[1]Includes parasitics for heat rejection, burner fan, solution pumps
[2]Calculated based on 10,867 Btu/kWh and zero distribution losses for gas
[3]Assuming 82% burner efficiency.
[4]Assuming use of a single-stage cycle. Two-stage cycles may have primary energy COP's ranging from 0.35 to 0.4.

The analysis assumptions are:

- Evaporator temperature -20°F for the vapor compression cycle, -30°F for the sorption cycles
- Condenser (and absorber) temperature 110°F
- Gas engine system:
  - COP based on high heating value of the gas: 0.55[1]

[1] Performance of FES industrial system. The Hussmann engine-driven system does not achieve this performance level.

  - Shaft power (25% of gas input) and all delivered cooling rejected at 0.14 kW per ton of rejected heat
  - Engine jacket heat (30% of gas input) rejected at 0.1 kw per ton of rejected heat
- Advanced Absorption[2]:
  - COP: 0.57 (based on fuel high heating value)
  - 80% reduction in electricity usage
- Chemisorption:
  - Cycle COP: 0.35[3]; 82% burner efficiency
  - 0.15 kW/ton burner fan[3]
  - Two thirds of the gas input and all of the delivered cooling rejected at 0.14 kW per ton of rejected heat

The above comparison shows that electric-driven conventional systems are more efficient than current gas-fired systems. However, the comparison does not consider the possibility of using the gas-fired systems for heating needs, which would improve their outlook. Furthermore, all the possibilities for gas-fired systems have not yet been fully examined. Systems in which engine waste heat is used to drive absorption cycles may have the greatest efficiency potential.

5.1.2 Economic Analysis - Supermarket Refrigeration

The economic analysis for energy efficiency improvements to supermarket refrigeration systems is presented in this section. The analysis includes (1) calculation of electricity usage and demand savings and fossil fuel savings associated with the measure, (2) estimation of non-energy operation and maintenance costs, (3) determination of the cost premium for installation of the measure, (4) calculation of savings in annual operating costs, (5) calculation of the simple payback period, and (6) estimation of the impact on U.S. energy usage. Specific discussion of the calculation of items (1) through (3) are discussed below for each measure. Dollar savings are calculated based on utility rate structures for three cities which are representative of high, medium, and low electricity costs: New York, Raleigh, NC, and Olympia, WA. The average cost per kWh of electricity, per kW of electric demand, and per MMBtu of natural gas were determined for these cities for general commercial rates applied to an energy profile typical for supermarkets. The utility costs are listed in Table 5-9 below.

Table 5-9: Utility Rates (Supermarkets)

City	New York	Raleigh, NC	Olympia, WA
Cost Level	High	Medium	Low
Electricity Usage $/kWh	$0.0661	$0.0530	$0.0451
Electric Demand $/kW	$23.75	$5.04	$5.68
Gas $/MMBtu	$6.38	$5.60	$9.68

2 Source: Energy Concepts
3 Source: Rocky Research

The economic analysis is presented for each measure individually in order to indicate clearly the impact of the changes. Costs and savings are calculated based on a comparison with the baseline system presented in Section 4.1.

Table 5-10 gives an overall summary of the economics of the examined energy efficiency measures. The measures are grouped by focus on (a) central refrigeration equipment (numbers 1 to 8) and on (b) display cases and walk-ins. Table 5-11 gives an overview of the measures' energy savings potential.

Table 5-10: Economic Analysis - Supermarkets

Baseline: Usage 1,600,000 kWh; Demand 242 kW

						Simple Payback Period		
		Reduction kWh/yr	Load Reduction kW	Cost Premium	Savings - Med Cost (Raleigh, NC)	High Cost New York	Med Cost Raleigh, NC	Low Cost Olympia, WA
1.	Evaporative Condenser[1]	49,000	7.3	($7,100)	($560)	NA	NA	NA
2.	Floating Head Pressure	49,000	9.6	$8,000	$3,200	1.3	2.5	2.8
3.	Ambient Subcooling	8,000	2.2	$6,100	$560	5.3	11	12
4.	Mechanical Subcooling	23,000	6.6	$8,000	$1,600	2.4	4.9	5.4
5.	Heat Reclaim (970 MMBtu gas savings)[2]			$13,700	$5,400	2.2	2.5	1.5
6.	Hot Gas Defrost	49,500		$3,800	$2,600	1.2	1.4	1.7
7.	Liquid Suction Heat Exchanger Low Temperature	37,900	6.9	$10,000	$2,400	2.2	4.1	4.6
8.	Liquid Suction Heat Exchanger Med. Temperature	28,400	5.1	$25,000	$1,800	7.5	14	15
9.	High-Efficiency Lighting	31,880	3.6	$3,850	$1,900	1.2	2.0	2.3
10.	PSC Evap Fan Motors	101,700	11.6	$7,600	$6,100	0.8	1.2	1.4
11.	ECM Evap Fan Motors	128,800	14.7	$12,600	$7,700	1.0	1.6	1.9
12.	Antisweat Htr Controls	90,400		$7,500	$4,800	1.3	1.6	1.8
13.	Improved Insulation	5,255	0.6	$11,000	$315	21	35	40
14.	Defrost Control (electric defrost)	20,900		$3,300	$1,100	2.4	3.0	3.5
15.	Defrost Control (hot gas defrost)	8,522		$3,300	$450	5.9	7.3	8.6
16.	High-Efficiency Fan Blades	50,389	5.8	$144	$3,000	0.03	0.05	0.05

[1] Additional Non-Energy Costs of $3,600 associated with evaporative condensers.
[2] Savings for heat reclaim are in HVAC and water-heating energy use. Projected heat reclaim savings represent 5.6% of refrigeration system primary energy use.

Note: Current Technologies are 1, 2, 3, 4, 5, 6, 7, 8, 12; New technologies are 9, 10, 11, 13, 16; Advanced Technologies are 14 and 15

Some of the energy saving measures discussed herein are currently used in existing supermarket refrigeration systems. The impact of the measures on overall U.S. primary usage is calculated based on an assumption of the number of supermarkets which do not currently have them in place. The impact estimate is based on an assumed inventory of 30,000 supermarkets. The calculation is adjusted by the ratio of average supermarket size (27,000 sq. ft.) to the size of our prototypical store (45,000 sq. ft.). A heat rate of 10,867 Btu/kWh is assumed in conversion of electricity into primary energy.

Table 5-11: Energy Savings Potential -- Supermarkets

		Reduction kWh/yr	kW	Refrigeration System Energy Reduction per Supermarket (%)	Percent of Supermarkets	Reduction Potential Primary Energy (triills)
1.	Evaporative Condenser	49,000	7.3	3.1	96	10
2.	Floating Head Pressure	49,000	9.6	3.1	38	4
3.	Ambient Subcooling	8,000	2.2	0.5	63	1
4.	Mechanical Subcooling	23,000	6.6	1.4	35	2
5.	Heat Reclaim (970 MMBtu gas savings)			*	15	3
6.	Hot Gas Defrost	49,500		3.1	31	3
7.	Liquid Suction Heat Exchanger Low Temperature	37,900	6.9	2.4	50	4
8.	Liquid Suction Heat Exchanger Med. Temperature	28,400	5.1	1.8	75	4
9.	High-Efficiency Lighting	31,880	3.6	2.0	100	6
10.	PSC Evap Fan Motors	101,700	11.6	6.4	100	21
11.	ECM Evap Fan Motors	128,800	14.7	8.1	100	26
12.	Antisweat Htr Controls	90,400		5.7	25	5
13.	Improved Insulation	5,255	0.6	0.3	100	1
14.	Defrost Control (electric defrost)	20,900		1.3	31	1
15.	Defrost Control (hot gas defrost)	8,522		0.5	69	1
16.	High-Efficiency Fan Blades	50,389	5.8	3.2	100	10

Central System Technologies

The measures focused on the central refrigeration system (compressors and condensers) and the liquid-suction heat exchanger measure have been analyzed based on Reference No. 6 . These are measures number 1 through 5. The percent electricity usage and demand reductions reported in the reference were applied to the prototypical supermarket description of Section 4.1. Note that usage and demand as reported in the reference does not include evaporator fan, lighting, and antisweat loads. Hence, the reduction percentages are applied to usage of 1,000,000 kWh and demand of 174 kW, representing the compressor, condenser, and defrost contributions for the prototypical supermarket of this study.

Savings possible with heat reclaim are calculated assuming that one quarter of the heat rejection load can be used for space or water heating during 4 winter months. This conservative estimate would result in annual savings of 970 MMBtu of gas, assuming that gas combustion efficiency is 75%.

Non-energy operating costs estimated in the reference were scaled up by the assumed sales area (25,600 sq. ft. for the reference vs. 45,000 sq ft in our case) and also by a factor to account for inflation. This latter factor, assuming 3% inflation for 2 1/2 years, is 1.08.

Installed cost premiums were calculated based on the above reference and Reference No. 7. Costs are again scaled by a factor of 1.08 to account for inflation.

Display Case Technologies

The display case technologies are numbers 7 through 16 in the summary tables above. For Hot Gas Defrost and the Liquid-Suction Heat Exchangers (numbers 6, 7, and 8), demand and usage reductions as reported in Reference 6 are applied to the prototypical supermarket of this study. Costs for the options as reported in the reference are increased by 8 percent to account for inflation. The economics for the other options for supermarket display cases are discussed individually below.

Lighting

An estimated potential energy savings of 15% could be achieved through the use of electronic ballasts. In addition, it is estimated that all of the heat generated by the lighting contributes to the caseload. By reducing lighting consumption, the compressor load is also reduced.

Table 5-12 shows the estimated lighting energy reductions and associated compressor usage reductions for display cases with lighting. Conversion to compressor usage is done assuming COP's of 2.5 for medium temperature and 1.3 for low temperature.

Table 5-12: Display Case Lighting Energy Savings (for prototypical 45,000 sq ft supermarket)

Case Type	Lighting Electricity Usage Reduction (kWh/yr)	Compressor Electricity Usage Reduction (kWh/yr)
Multideck Meat (MT)	1,845	740
Multideck Other (MT)	6,300	2,520
Reach-In (LT)	11,565	8,900
Totals	19,710	12,160

Based on manufacturer cost data, the cost premium for electronic ballast lighting is about \$4.20 per linear foot of multideck cases and about \$8.40 per linear foot of reach-in cases. This is equivalent to a cost premium of about \$3,850 for a 45,000 ft^2 supermarket. The payback period ranges about one to two years for new equipment.

High Efficiency Evaporator Fan Motors

Table 5-2 presents estimates of display case motor counts for a typical supermarket. These estimated counts have been adjusted so that total evaporator fan power input agrees with the evaporator fan load of our prototype supermarket system. It is assumed that all existing motors are of the shaded pole type. The following Table 5-13 shows the assumed motor breakdown count.

Table 5-13: Evaporator Fan Motor Power Requirements (for prototypical 45,000 sq ft supermarket)

		Shaded Pole		PSC		ECM	
Motor Output (W)	Total Number	Total Cost(OEM)	Total Power (W)	Total Cost (OEM)	Total Power (W)	Total Cost (OEM)	Total Power (W)
6	85	$595	3,400	$2,125	1,275	$2,975	720
9	100	$1,000	5,300	$2,800	2,100	$4,000	1,250
25	40	$1,000	4,400	$1,480	2,040	$1,920	1,320
Totals		$2,600	13,100	$6,400	5,400	$8,900	3,290

Note: Per Motor OEM costs and wattages are listed in Table 5-2

The table also lists the power requirements for all the evaporator fan motors of the store, assuming that the motors are of the shaded pole, PSC, or ECM type. Based on total power as listed in the table for the three motor types, the savings in evaporator fan power are 7.7 kW when using PSC motors and 9.8 kW using ECM motors. Assuming 24 hour per day operation, electricity usage savings are 67,450 for the PSC motors and 85,900 kWh for the ECM's.

Additional savings due to the reduction in case load are calculated as follows (See Table 5-14). The electricity usage and demand savings are apportioned to the medium and low temperature systems based on the assumed 750 kBtu/hr and 300 kBtu/hr case load breakdown for these systems. System COP's of 2.5 for medium temperature and 1.3 for low temperature are used to convert case load to compressor power. The additional savings are about 50% of the fan power savings: 34,200 kWh and 3.9 kW for the PSC motors; 42,900 kWh and 4.9 kW for the ECM's.

Table 5-14: Compressor Load Reductions due to Use of High-Efficiency Evaporator Fan Motors

		Shaded Pole	PSC	EMC
	Total Motor Load (kW)	13.1	5.4	3.3
A	Motor Load in Low Temperature System (kW)	3.74	1.54	0.94
B	Motor Load in Medium Temperature System (kW)	9.36	3.86	2.36
C	Associated Low Temp Compressor Load (kW) A÷1.3	2.88	1.18	0.80
D	Associated Medium Temp. Compressor Load (kW) B÷2.5	3.74	1.54	0.94
E	Total Associated Compressor Load (kW) C + D	6.6	2.7	1.7

The OEM costs for installation of PSC instead of shaded pole motors is $3,800. The incremental cost premium for the ECM motors is $6,300 (additional mark-ups of 100% are assumed for the end-user). Table 5-10 shows the measured payback periods for the cities examined. The payback period for PSC motors is slightly shorter. However, payback in all examined locations is three years or less.

High Efficiency Fan Blades

The economics for high efficiency fan blades depend on the size of production runs possible for each specific fan blade design. This will depend on the number of each type of refrigerated display case sold by each of the major manufacturers. Assuming four fans per case, 50 cases per store, and 2,500 stores newly built or remodeled per year, the annual supermarket fan requirement is 500,000 units. If these fans are divided among four manufacturers and ten pressure/flow requirements, the number of fans per application per manufacturer is 12,500. A $16,000 tooling cost is assumed to be distributed over this rate of fan blades for a four year period. The engineering costs are accounted for by assuming that the end-user markup is 100% (this markup also includes profit and distribution costs). The OEM cost premium per blade is $0.32, and the end-user cost premium per blade is $0.64. This represents a fairly optimistic view of the cost premium, because (1) the required payback to the OEM is likely to be not more than 2 years, (2) the number of pressure/flow requirements is likely to be more than 10 per store, and (3) the market is not evenly split amongst four manufacturers. The cost per blade could be four times higher. Nevertheless, the quick payback for the measure for the end-user suggests that it is a promising option which should be pursued.

Antisweat Heater Controls

It is assumed that one third of antisweat heating electricity usage can be eliminated with the use of controls (the calculations for the prototypical store assume that the heaters are always operating). In addition, one-half of the heater load is assumed to contribute to the caseload--elimination of part of the heater load in this way also reduces the compressor load.

Table 5-15 below shows assumed antisweat heater electricity reductions and associated compressor usage reductions for the display cases with antisweat heaters. Conversion to compressor usage is done assuming COP's of 2.5 for medium temperature and 1.3 for low temperature.

Table 5-15: Antisweat Heater Control Energy Savings (prototypical 45,000 sq ft supermarket)

Case Type	Temperature Level	Baseline Annual Usage (kWh)	Annual Usage Reduction (kWh)	Annual Compressor Electricity Usage Reduction (kWh)
Multideck Meat (120ft)	Med	10,500	3,500	700
Multideck Other (260 ft)	Med	0	0	0
Reach-In (268 ft)	Low	159,800	53,300	20,500
Single-Level Open (128 ft)	Low	26,900	9,000	3,400
Totals		197,200	65,800	24,600

There are no assumed demand savings for the measure because the antisweat heaters are likely to be operating at nearly 100% level at times of high demand. Although this is not the case during winter months in temperate climates, the assumption is made to simplify the analysis.

The cost for installation of antisweat heater controls is determined assuming that one sensor/controller is installed for each display case line-up. This is somewhat less expensive than ordering of individual controllers with the cases. The cost for this installation is about $500 per sensor. It is assumed that on average one sensor will be required for every three 12-foot cases having antisweat heaters[4]. The total cost for the prototype store, representing about 15 sensors, is $7,500. The payback period is less than two years for all three considered cities (See Table 5-10).

Insulation Improvement

As discussed in Section 5.1.1, the savings possible for reducing case insulation from a thermal conductivity of 0.137 Btu-in/hr ft^2F to 0.120 Btu-in/hr ft^2 F include about 5,000 kWh/yr electricity usage and 0.6 kW demand for the 45,000 sq. ft. prototypical supermarket. If the $10,000,000 typical cost for improved foam blowing equipment is spread over the cost of 30,000 cases for four years, and the end-user markup is 100%, the per-case cost premium is very roughly $170 per case. The annual case sales estimate assumes that a given manufacturer serves one quarter of the market for 50 cases per store and 2,500 stores per year (1,500 remodels and 1,000 new stores).

The case count for the prototypical supermarket is 65 (784 ft of 12-foot cases). The end-user cost of improved insulation would be $11,000. The simple payback period is quite long--at least 20 years. The economics are summarized in Table 5-10.

Defrost Control

It is assumed that one half of the defrost load can be eliminated during the six cooler months. Compressor savings are also achieved due to reduced case loads. The savings in kWh are summarized in Table 5-16 below. Only the compressor electricity usage savings would be realized in stores with hot gas defrost. The payback period is about three years in stores with electric defrost and about 7 years in stores with hot gas defrost.

Table 5-16: Defrost Control (Prototypical 45,000 sq. ft. Supermarket)

Case Type	Baseline Defrost Electricity Usage (kWh)	Defrost Electricity Usage Reduction kWh/yr	Compressor Electricity Usage Reduction kWh/yr
Multideck Meat (MT)	7,900	1,975	790
Reach-In (LT)	18,700	4,675	3,596
Single-Level Open (LT)	9,400	2,350	1,808
Meat Walk-In (MT)	2,900	725	290
Low Temp. Walk-Ins (LT)	10,600	2,650	2,038
Totals	**49,500**	**12,375**	**8,522**

5.1.3 Barriers to Implementation

1) Selection of display cases is dominated by supermarket chain merchandising people, for whom increased product sales is much more important than energy efficiency. First, there is a lack of awareness of the value of energy-savings potential among people selecting the cases. Second, the incentives for these people are for purchase of less expensive rather than more efficient equipment. Return on investment is considered better for design features which enhance sales than for energy-saving features.

[4] Source: Discussion with a supermarket engineer

2) First cost is a barrier. Paybacks of 2 to 3 years are required by most supermarkets. Some even require paybacks of less than one year.

3) Reliability and product track record are extremely important. Untested technology is not readily accepted. New technologies will have to be field tested and proven before they are generally accepted. Large supermarket chains are fairly active in testing new technologies in demonstration stores. The limited initial use of a new concept in this fashion reduces the risk to the supermarket chain.

4) There is too much variability in supermarket refrigeration systems for efficiency standards to be practical. Supermarket energy usage is dependent on a large number of factors, including HVAC and building shell. There are a large number of system configurations adapted to suit individual stores. Hence it is difficult apply to specific supermarkets the generalizations regarding expected energy usage.

5) Evaporative Condensers have added 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^{o}F typical design condenser temperature (compared with 110^{o}F to 115^{o}F for supermarkets). The desire to keep maintenance costs to a minimum is more of an issue with supermarkets.

6) Refrigerant leakage is a problem which has been claimed to be exacerbated by hot gas defrost. The costs of replacing leaking refrigerant can approach $5000 per year. The thermal cycling caused by hot gas defrost causes stresses which can weaken pipe connections, increasing the leakage. The magnitude of this problem is not clear, but there is a perception that hot gas defrost may increase the problem, which reduces its acceptance.

5.2 Beverage Merchandiser

5.2.1 Energy-Saving Technologies

This section describes the energy saving technologies which are applicable to beverage merchandisers, and their energy savings potential. Detailed calculation of savings and economics and a summary table are presented in Section 5.2.2

New Technologies

Lighting

Reduction in lighting energy consumption in beverage merchandisers also reduces compressor power by decreasing the internal load. Since the lighting level is strongly related to sales levels, reductions in electricity usage must be done through the installation of high-efficiency lamps and ballasts, rather than reduction in light output.

Most beverage merchandisers use T12 fluorescent lighting with magnetic ballasts. The lighting configuration for a one door merchandiser will usually consist of one horizontal 20-watt lamp to light the logo and one vertical 40-watt lamp along the door hinges to light the product. Bottling companies will sometimes specify high-output lighting as part of a lighting system to enhance sales.

It is recommended that electronic ballasts be considered as a basic energy-saving option over standard magnetic ballasts.

An estimated potential energy savings of 25% could be achieved through the use electronic ballasts. In addition, it is estimated that two thirds of the heat generated by the lighting contributes to the caseload. By reducing lighting consumption, the compressor load is also reduced.

It is estimated that about 270 kWh of lighting energy can be saved annually through the use of electronic ballasts. The associated compressor energy savings is 105 kWh based on a 1.72 COP at operating conditions (see Tables 4-14 - 4-16).

High-Efficiency Evaporator and Condenser Fan Motors

Most evaporator and condenser fan motors are inexpensive and inefficient single-phase shaded pole motors. The efficiency of permanent split capacitor (PSC) or ECM motors is significantly better. These three motor types are compared in Table 5.2 in Section 5.1.1.

Additional savings are achievable when using higher efficiency motors for evaporator fans due to the reduced refrigeration load.

The prototype beverage merchandiser uses two 9-Watt output shaded pole evaporator fan, motors, which use 53 Watts of power each. Replacement with PSC motors would save 64 W in fan power, while ECM's would save 81 W. Additional compressor savings of about 60% of these values would be achieved.

The condenser fan motor is assumed to be a shaded pole motor with 57-Watt input and 9-Watt output. Savings of 36 W with a PSC replacement and 44.5 W with an ECM are possible.

Insulation

The insulation thickness for the prototype beverage merchandiser is 1.5 inches. The lack of space in these units limits the possible insulation thickness increases. A one-inch increase in insulation thickness would reduce wall losses by 38%. The resulting reduction in compressor energy usage, about 115 kWh per year, represents about 25% of the unit's annual usage. Assuming constant exterior dimensions, the reduction is internal volume would be about 4.7 cu ft., a 17% reduction. The insulation thickness increase is reasonable only if the internal volume reduction is acceptable, or if exterior demensions can be increased. The costs of increasing insulation thickness include added material costs (polyurethane and blowing agent), product redesign costs, and manufacturing plant retooling costs. The OEM material cost for an additional inch of insulation is about \$0.33 per square foot (see Section 5.1.1).

Wall losses could also be reduced by the use of improved insulation, which has a conductivity of 0.120 Btu-in/hr-ft^2 F, about 12% lower than standard practice insulation. The conductivity reduction is due to the foam's cell size. Implementation of such a measure would required installation at the factory of improved foaming equipment, an investment which would cost several million dollars. The resulting additional equipment cost would depend on production numbers and corporate policy for amortizing the initial investment cost.

High Efficiency Compressor

The prototypical beverage merchandiser has a standard efficiency hermetic reciprocating compressor with a resistor start, induction run (RSIR) motor. The compressor's rated COP is 1.72 at the typical rating conditions of 20^{o}F evaporator and 120^{o}F condenser. The efficiency of the motor is about 70% and the overall compressor efficiency is 48%.

Most commercial refrigerators, beverage merchandisers and vending machines use Tecumseh's AE-Line compressors up to 1/3 hp. Medium temperature units requiring greater capacity use AK-Line compressors. Low temperature units requiring greater capacity use AJ-Line compressors.

Currently, there is little demand by OEM's for high-efficiency compressors for commercial refrigeration equipment. Americold supplies small high-efficiency compressors to manufacturers of residential refrigerator-freezers (i.e. Sub-Zero, GE, Frigidaire, Maytag, Amana, etc.). These compressors are low-suction pressures units ranging from 600 Btuh to 1,200 Btuh in capacity. HFC-134a is the refrigerant used. Reported compressor efficiencies for the Americold RH series are up to 5.6 EER at -10^{o}F evaporator temperature and 130^{o}F condensing temperature. These units 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 5.6 EER Americold compressor is therefore 55%.

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.

The OEM costs for 1/3 hp compressors are about $40. Currently available high-efficiency compressors are reported to have a 10% cost premium (Tecumseh). An $8 OEM cost premium is used in the economic analysis.

Improvements in compressor efficiency could also be achieved with the use of ECM compressor motors. Data for these motors in 1/3 and 1/2 hp sizes is presented in Table 5.1.2 above. Currently such compressors are not available, except in limited numbers for special order.

Advanced Technologies

Variable-Speed Compressors

The use of ECM motors would allow variable speed operation of the compressors with appropriate controls. Variable speed operation would allow further reductions in energy usage for the following reason:

(1) When refrigerant flow is reduced during part-load operation, the condenser and evaporator(designed for full flow conditions) are more effective. Temperature drops decrease, resulting in reduced pressure rise across the compressor.

(2) Close matching of load eliminates the cycling which occurs with single-stage compressors. Maintaining a constant pressure is more efficient because losses at higher pressure rise are greater than gains at lesser pressure rise.

(3) During the off-cycle, the pressure in the system equilibrates. At the intermediate pressure, refrigerant vapor will condense in the cold evaporator rather than the condenser. Essentially, some of the heat rejection load is rejected to the evaporator during this time, reducing overall system performance. Variable speed operation would eliminate compressor off-time and the related inefficiencies.

For the economic analyses for the self-contained equipment, it is assumed that reductions in compressor power of 15% - 20% are possible with variable speed operation. This reduction range has been achieved in tests at ADL with two-speed compressor operation with a residential refrigerator/freezer. A 20% savings potential is assumed for the beverage merchandiser and refrigerated vending machine, which have low compressor duty cycle in current configurations.

5.2.2 Economic Analysis

Table 5-17 summarizes the economic analysis for energy-saving technologies for beverage merchandisers. The analysis for beverage merchandisers and the other self-contained refrigeration units uses an average per kWh charge rather than examination separately of demand and usage charges. The average charges for the three cities considered (New York, Raleigh, NC, and Olympia, WA) are listed at the top of the table.

The analysis for the combination of features includes the ECM evaporator fan motor and the high-efficiency compressor.

Table 5-17: Economic Analysis Beverage Merchandiser

Baseline Energy Usage 3923 kWh/yr

						Simple payback Period (yrs)		
	Technology Option	**End-User Cost Premium**	**Load Reduction (W)**	**Energy Reduction (kWh/yr)**	**Energy Reduction (%)**	**High Rate ($0.1834/ kWh)**	**Medium Rate ($0.0782/ kWh)**	**Medium-Low Rate ($0.0743/ kWh)**
1	Thicker Insulation	$56	13	116	3.0	2.6	6.2	6.5
2	Improved Insulation	$208	4	37	0.9	31	72	75
3	PSC Evap Fan Motor	$72	101	887	23	0.4	1.0	1.1
4	ECM Evap. Fan Motor	$120	127	1118	29	0.6	1.4	1.4
5	PSC Cond. Fan Motor	$36	30	118	3.0	1.7	3.9	4.1
6	ECM Cond. Fan Motor	$60	45	175	4.5	1.9	4.4	4.6
7	High-Efficiency Compressor	$16	--	335	9	0.3	0.6	0.6
8	ECM Compressor Motor	$100	64	251	6	2.1	5.0	5.4
9	Variable Speed Compressor	$150	64	536	14	1.5	3.7	3.8
10	Lighting Improvement	$30	43	380	9.7	0.4	1.0	1.1
11	High Efficiency Fan Blades	$3	34	254	3	0.1	0.2	0.2
12	Combination	$136	127	1371	35	0.5	1.3	1.3

12 Includes ECM Evap fan motor and high-efficiency compressor (items 4 and 7).
Note: New Technologies are 1, 2, 3, 4, 5, 6, 7, 9, 10, 11, - Advanced Technology is 9.

he calculations made for the economic analysis are briefly outlined below.

1) Thicker Insulation: Reduction of the 204 Btu/hr wall load (Table 4.2.4) by 38%(increase in total wall resistance, including inside and outside air layers, from 13.5 to 21.2 R-value, due to insulation thickness increase from 1.5 to 2.5 inches). COP of 1.72. Insulation material costs $0.33/sqft for 69 sqft of wall area. Additional $500,000 retooling costs distributed among 96,000 units: 40 percent of annual sales of 60,000 units for four years; markup to the end user of 100%.

2) Improved Insulation: Reduction of the wall load by 12%. Distribution of $10,000,000 foam blowing equipment capital costs among 96,000 units. End-user markup of 100%.

3&4) High-Efficiency Evaporator Fan Motors: Motor costs and power requirements as shown in Table 5-2. Replacement of two 9-Watt output shaded pole motors with two 9-Watt output PSC or ECM motors. Additional compressor load savings based on the 1.72 COP.

5&6) High-Efficiency Condenser Fan Motors: Motor costs and power requirements as shown in Table 5-2. Replacement of one 9-Watt output shaded pole motor with a 9-Watt output PSC or ECM motor.

7) High-Efficiency Compressor: Increase in COP from 1.72 to 2.15, resulting in a 20% reduction in compressor power input. OEM cost of $8, end user markup of 100%.

8&9) ECM Compressor Motor/Variable Speed Compressor: Motor costs and power requirements as shown in Table 5-2. Replacement of the existing 1/3 hp motor with an ECM motor (efficiency increase from 70% to 82%). Cost premium of $100 for the ECM motor ($100 ECM motor cost, $50 standard motor cost, 100% end-user markup). Additional 20% reduction in compressor energy usage for variable speed operation. End-user cost for variable speed controls of $50.

10) High Efficiency Lighting: Based on discussions with lighting suppliers, the cost premium for a 48", two-lamp electronic ballast is about $15. The same cost is assumed for the beverage merchandiser, for which a 48" and a 24" lamp are used. Assume a 100% markup in the merchandiser. Approximately 2/3 of the heat from lighting enters the refrigerated space. Efficacy improves from 65 LPW to 87 LPW (25% savings).

11) High-Efficiency Fan Blades: Reduction of 15% in evaporator and condenser fan load. Additional savings for evaporator fan due to reduced compressor load calculated based on a COP of 1.72. Cost premium of $1.00 per fan blade. (tooling costs of $16,000 distributed over a short production run of 32,000; end-user markup of 100%).

12) Combination: High-Efficiency Compressor and ECM Evaporator Fan Motor: Savings for the improved compressor as described for #7. The ECM evaporator fan motor savings are reduced because conversion of the reduction in case load to compressor power savings involves the improved COP.

5.2.3 Barriers to Implementation

1) Most beverage merchandisers are owned by bottling companies, who do not pay utility bills for the buildings where the units are located. This effectively eliminates any incentive of the machine owners to select machines with higher priced, more efficient features.

2) Energy costs are small compared with typical beverage sales revenues. Energy costs for a single-door beverage merchandiser (for Raleigh, NC, representing average electricity costs) are about $300 annually. The total revenues for beverage sales average about $8000 for such a machine. Although the energy costs are not completely insignificant, they do not represent a large part of the sales revenues. This reduces awareness of energy as a major concern. It also increases the tendency to disregard energy issues in evaluating sales-boosting design changes, such as an increase in lighting intensity.

3) Space in beverage merchandisers is tight. Therefore increases in insulation thickness are undesirable. A decrease in storage volume is likely to reduce the sales capacity of a given machine, which is unacceptable for the reasons mentioned above.

4) The production numbers for commercial refrigeration equipment are not high enough. This applies not only to beverage merchandisers. It is in contrast with residential refrigerator/ freezers, which are produced in the millions, at least an order of magnitude greater than production rates for commercial equipment. The engineering and tooling costs associated with commercial equipment cannot be absorbed as easily by increases in product costs, reducing the attractiveness for manufacturers of developing more efficient equipment.

5.3 Reach-In Freezer (Single-door)

5.3.1 Energy Saving Technologies

The following energy saving technologies which are applicable to reach-in freezers are described briefly in Section 5.2.1: High-Efficiency Fan Motors, Insulation, High-Efficiency Compressors, and Coil Improvements. Application of the technologies to the reach-in freezer will result in different energy usage reductions, but concepts are identical. The fan motors used in reach-in freezers are permanent split capacitor (PSC) - type. Energy savings in fan motor power are possible by replacement with ECM motors.

Additional energy saving technologies for the reach-in freezer are discussed in this section: hot gas defrost, hot gas antisweat heating, liquid-suction heat exchange, and control of defrost.

New Technologies

Hot Gas Defrost

The need for defrost and the three main defrost options are discussed in Section 5.1.1. Prototypical Reach-in freezers use electric defrost. Savings could be achieved by the implementation of hot gas defrost. Costs would increase due to additional controls and refrigerant piping.

Liquid-Suction Heat Exchangers
Liquid-suction heat exchangers are discussed in Section 5.1.1 in the context of supermarket refrigeration (see Figure 5-4). Compressor power could be reduced in the range of 2 to 5 percent with such a device for the operating conditions typical for the reach-in freezer.

Advanced Technologies

Hot Gas Antisweat Heating
Antisweat heating is discussed in Section 5.1.1. The gaskets of reach-in freezers also require antisweat heating. This function, normally provided by electric heaters, could also be provided by a hot gas line running in the door frame. Although manufacturers have claimed that this is a difficult technology to implement, it has been used successfully in residential freezers.

Defrost Control
Controlled initiation defrost may be applicable to reach-in freezers (see discussion of this technology in Section 5.1.1). The most promising technique involves monitoring the temperature drop across cooling coils to determine whether air flow rates have dropped. The technology is still under development for application to supermarket display cases. It may be effective in reducing defrost energy usage in reach-in freezers but must be adequately tested.

5.3.2 Economic Analysis
Table 5-18 summarizes the economic analysis for energy-saving technologies for reach-in freezers.

The analysis for the combination of features includes: ECM motors for the evaporator and condenser fans, hot gas antisweat, and high-efficiency compressors.

Table 5-18: Economic Analysis Reach-In Freezers

Baseline Energy Usage 5,198 kWh/yr

	Technology Option	End-User Cost Premium	Load Reduction (W)	Energy Reduction (kWh/yr)	Energy Reduction (%)	Simple payback Period (yrs) High Rate ($0.1834/kWh)	Medium Rate ($0.0782/kWh)	Medium-Low Rate ($0.0743/kWh)
1	Thicker Insulation	$84	22	197	3.8	2.3	5.5	5.7
2	Improved Insulation	$625	9	81	1.6	42	100	104
3	ECM Evap Fan Motor	$24	13.5	118	2.3	1.1	2.6	2.7
4	ECM Cond Fan Motor	$24	21	138	2.7	0.9	2.2	2.3
5	High-Efficiency Compressor	$24	127	831	16	0.1	0.4	0.4
6	ECM Compressor Motor	$110	130	814	16	0.7	1.8	1.9
7	Variable Speed Compressor	$160	130	986	19	0.9	2.1	2.2
8	Hot Gas Defrost	$83	600	329	6.3	1.4	3.2	3.4
9	Hot Gas Antisweat	$67	85	745	14	0.5	1.2	1.2
10	Liquid-Suction Heat Exchanger	$75	--	174	3.4	2.3	5.5	5.8
11	Defrost Control	$50	--	148	4.4	1.8	4.3	4.5
12	High-Efficiency Fan Blades	$2	16	116	2.2	0.1	0.2	0.2
13	Combination	$139	294	1819	35	0.4	1.0	1.0

Note: New Technologies are 1, 2, 3, 4, 5, 6, 8, 10, 12; Advanced Technologies are 7, 9, 11

The calculations made for the economic analysis are briefly outlined below.

1) Thicker Insulation: Reduction of the 329 Btu/hr wall load (Table 4-21) by 29%(increase in total wall resistance, including inside and outside air layers, from 17 to 24 R-value, due to insulation thickness increase from 2¼ to 3¼ inches). COP of 1.25. Insulation material costs $0.33/sqft for 80 sqft of wall and door area. Additional $500,000 retooling costs distributed among 32,000 units: 10 percent of annual sales of 80,000 units for four years; markup to the end user of 100%.

2) Improved Insulation: Reduction of the wall load by 12%. Distribution of $10,000,000 foam blowing equipment capital costs among 32,000 units. End-user markup of 100%.

3) ECM Evaporator Fan Motor: Motor costs and power requirements for the ECM motor as shown in Table 5-2. Replacement of one 9-Watt output PSC motor with

one 9-Watt output ECM motor; $28 OEM cost for the PSC motor. Additional compressor load savings based on the 1.25 COP.

4) <u>ECM Condenser Fan Motor:</u> Motor costs and power requirements as shown in Table 5-2. Replacement of one 1/20 hp PSC motor with a 1/20 hp ECM motor.

5) <u>High-Efficiency Compressor:</u> Increase in COP from 1.25 to 1.67, a 25% reduction in compressor power input. OEM cost of $12, end user markup of 100%.

6&7) <u>ECM Compressor Motor/Variable Speed Compressor:</u> Motor costs and power requirements as shown in Table 5-2. Replacement of the existing 1/2 hp motor with an ECM motor (efficiency increase from 70% to 83%). Cost premium of $110 for the ECM motor ($110 ECM motor cost, $55 standard motor cost, 100% end-user mark-up). Additional 15% reduction in compressor energy usage for variable speed operation. End-user cost for variable speed controls of $50.

8) <u>Hot Gas Defrost:</u> Elimination of electric defrost energy of 600W, 329 kWh per year (Table 4-22). Cost of $83 based on supermarket hot gas defrost cost of $3800 for 46 display cases.

9) <u>Hot Gas or Liquid Antisweat:</u> Elimination of electric antisweat energy of 85 W, 745 kWh per year (Table 4-22). Engineering and retooling cost of $500,000 distributed among 32,000 units. Additional 18 feet copper refrigerant pipe @ $1. Markup to end-user of 100%.

10) <u>Liquid-Suction Heat Exchanger:</u> Installed cost of $372/ton (based on estimates for supermarket cases in Reference 6) plus eight percent for inflation. Five percent improvement in cycle COP assumed.

11) <u>Defrost Control:</u> Elimination of half the required defrost energy for six cooler months, additional savings due to reduced internal load based on a 1.25 COP: 0.25*(329)*(1+1/1.25)=148. Defrost energy reported in Table 4-22. Cost of $50 for two sensors and controls.

12) <u>High-Efficiency Fan Blades:</u> Reduction of 15% in evaporator and condenser fan load. Additional savings for evaporator fan due to case load reduction based on 1.25 COP. Cost premium $1 per fan blade (tooling costs of $16,000 distributed over 32,000 fan blades; 100% end-user markup).

13) <u>Combination: ECM Evaporator Fan Motor, ECM Condenser Fan Motor, Hot Gas Antisweat, and High Efficiency Compressor:</u> Savings as calculated for #4 (138 kWh), #5 (831 kWh), and #9 (745 kWh); ECM evaporator fan motor savings reduced due to the improved compressor COP (105 kWh rather than 118 kWh).

5.3.3 Barriers to Implementation

1) The long term success of start-up restaurants is never certain. Investment capital is limited in such situations, making the purchase of high-efficiency equipment with a cost premium undesirable. If the payback is not extremely quick, such equipment is not purchased.

2) The reach-in market is split amongst a relatively large number of manufacturers. Sales are to a fragmented group of large and small end-users. This situation makes implementation of efficiency improvements more difficult.

3) The thermal cycling caused by hot gas defrost has been claimed to be responsible for increased refrigerant leakage. This is limiting and in some cases reducing its acceptance.

4) Manufacturers have claimed to have encountered difficulty in implementing non-electric defrost using hot gas or high pressure liquid. This technology can involve significant modification to the refrigerant loop and requires careful development.

5.4 Reach-In Refrigerator (two-door)

5.4.1 Energy Saving Technologies

The energy saving technologies applicable to reach-in refrigerators are insulation improvement, ECM fan motors, high-efficiency compressors, lighting improvement, coil improvement, and hot gas antisweat heating. These technologies are discussed above in Sections 5.2.1 and 5.3.1.

5.4.2 Economic Analysis

Table 5-19 summarizes the economic analysis for energy-saving technologies for reach-in refrigerators.

The analysis for the combination of features includes ECM motors for evaporator and condenser fans, hot gas antisweat heating, and high-efficiency compressors.

Table 5-19: Economic Analysis Reach-In Refrigerators

Baseline Energy Usage 4,321 kWh

						Simple payback Period (yrs)		
	Technology Option	End-User Cost Premium	Load Reduction (W)	Energy Reduction (kWh/yr)	Energy Reduction (%)	High Rate ($0.1834/ kWh)	Medium Rate ($0.0782/ kWh)	Medium-Low Rate ($0.0743/ kWh)
1	Thicker Insulation	$100	11	97	2.2	5.6	13	14
2	Improved Insulation	$416	5	40	0.9	57	133	140
3	ECM Evap Fan Motor	$48	34	300	5.1	0.9	2.0	2.2
4	ECM Cond Fan Motor	$22	25	142	3.3	0.8	2.0	2.1
5	High-Efficiency Compressor	$16	88	501	12	0.2	0.4	0.4
6	ECM Compressor Motor	$100	64	367	8	1.5	3.5	3.7
7	Variable Speed Compressor	$150	64	688	16	1.2	2.8	2.9
8	Hot Gas Antisweat	$93	99	869	20	.6	1.4	1.4
9	High-Efficiency Fan Blades	$3	24	171	4.0	0.1	0.2	0.2
10	Combination	$179	268	1792	44	0.5	1.3	1.3

Note: New Technologies are 1, 2, 3, 4, 5, 6, 9; Advanced Technologies are 7 and 8.

The calculations made for the economic analysis are briefly outlined below.

1) <u>Thicker Insulation</u>: Reduction of the 265 Btu/hr wall load (Table 4.4.4) by 29%(increase in total wall resistance, including inside and outside air layers, from 17 to 24 R-value, due to insulation thickness increase from 2¼ to 3¼ inches). COP of 2.04. Insulation material costs $0.33/sqft for 120 sqft of wall and door area. Additional $500,000 retooling costs distributed among 48,000 units: 10 percent of annual sales of 120,000 units for four years; markup to the end user of 100%.

2) <u>Improved Insulation:</u> Reduction of the wall load by 12%. Distribution of $10,000,000 foam blowing equipment capital costs among 48,000 units. End user markup of 100%.

3) <u>ECM Evaporator Fan Motor</u>: Motor costs and power requirements for ECM motors as shown in Table 5-2. Replacement of two 9-Watt output PSC motors with two 9-Watt output ECM motors. OEM cost for PSC motors of $28. Additional compressor load savings based on the 2.04 COP. End user markup of 100%.

4) ECM Condenser Fan Motor: Motor costs and power requirements as shown in Table 5-2. Replacement of one 1/15 hp PSC motors with one 1/15 hp ECM motors. OEM cost for PSC motor of $43. End user markup of 100%.

5) High-Efficiency Compressor: Increase in COP from 2.04 to 2.55, a 20% reduction in compressor power input. OEM cost of $8, end user markup of 100%.

6&7) ECM Compressor Motor/Variable Speed Compressor: Motor costs and power requirements as shown in Table 5.1.1. Replacement of the existing 1/3 hp motor with an ECM motor (efficiency increase from 70% to 82%). Cost premium of $100 for the ECM motor ($100 ECM motor cost, $50 standard motor cost, 100% end-user markup). Additional 15% reduction in compressor energy usage for variable speed operation. End user cost for variable speed controls of $50.

8) Hot Gas or Liquid Antisweat: Elimination of electric antisweat energy of 99 W, 869 kWh per year (Table 4.4.5). Engineering and retooling cost of $500,000 distributed among 48,000 units. Additional 36 foot of copper refrigerant pipe @$1. Markup to end-user of 100%.

9) High-Efficiency Fan Blades: Reduction of 15% in evaporator and condenser fan load. Additional savings for evaporator fan due to case load reduction based on 2.04 COP. Cost premium $1 per fan blade (tooling costs of $16,000 distributed over 32,000 fan blades; 100% end-user markup)

10) Combination: ECM Evaporator Fan Motor, ECM Condenser Fan Motor, Hot Gas Antisweat, and High Efficiency Compressor: Savings as calculated for #4 (142 kWh), #5 (501 kWh), and #8 (869 kWh); ECM evaporator fan motor savings reduced due to the improved compressor COP (280 kWh rather than 300 kWh).

5.4.3 Barriers to Implementation

Barriers associated with implementation of energy saving technologies are similar for Reach-in Freezers and Reach-in Refrigerators. See Section 5.3.3

5.5 Ice Machines

5.5.1 Energy-Saving Technologies

This section describes the energy saving technologies which are applicable to ice machines, in particular to the 500 lb/day machine for the baseline energy consumption calculations in section 4.5.

New Technologies

High-Efficiency Compressor
The typical compressor used in ice machines in the 500 lb/day size range is a 3/4 hp capacitor start-induction run reciprocating compressor with efficiencies in the 42 - 45 percent range. Capacitor start-capacitor run compressors are available that have efficiencies 5 - 10 percent higher than the capacitor start induction run compressors. The higher efficiency compressors are $20-30 greater in price than the standard efficiency designs. With markups, the end user cost premium would be about $40.

Energy savings would be realized during the freeze cycle only, since a fixed amount of energy is required for ice harvest. The estimated energy reduction associated with the higher efficiency compressor is about 200 - 400 kWh/yr.

High-Efficiency Condenser Fan Motor
The prototypical condenser fan motor is a 25W-output shaded pole motor which consumes 100W. Comparable PSC motors are available that consume about 51 W, resulting in a saving of 49 W. The estimated reduction of energy consumption is about 200 kWh/yr. The additional end-user cost associated with the PSC motor option is about $24. An ECM motor with comparable output would require 33W, resulting in annual savings of 271 kWh. The cost premium to the end-user for this motor would be about $46.

Improved Evaporator Cold Compartment Insulation
The prototype ice machine evaporator cold compartment was insulated with 1/2" insulation. Doubling the insulation thickness would reduce the heat leak into the cold compartment, resulting in a reduction of energy consumption of about 3 percent. Assuming that $500,000 in engineering and retooling costs would be distributed among four-years sales of 10,000 units per year, that material cost additions would be $0.17 per square foot for 45 square foot, and that the markup to the end-user is 100%, incremental cost premium would be $40 per machine.

Interchanger to Reduced Purge Water Losses
The loss associated with purging cool (~32°F) water that is purged from the ice machine is about 10 percent. An interchanger to exchange heat from the incoming fresh water stream with the cold purge water would recover some of the cooling capacity from the previous freeze cycle. The maximum possible energy saving was estimated assuming an interchanger effectiveness of 100 percent, in which the outlet temperature for each stream are equal for equal mass flow rates of water for each stream. For this case, the estimated energy saving is about 5 percent. Realistic interchanger levels of effectiveness are in the range of 70 - 80 percent, resulting in a savings on the order of only 3 - 4 percent.

Implementation of this measure would be difficult due to the space constraints in ice machines. Significant pressure drop in the purge water flow path to the drain will not be acceptable. In addition, the purge water will have to be drained from the purge water basin prior to entry of fresh water in order to maximize effectiveness. This would involve a relatively large basin or container for the purge water in which an exchanger coil carrying the fresh water is placed. The size requirements would make the device impractical.

Reduced Meltage During Harvest

Ice meltage during harvest for the prototype was taken to be 15 percent, based on performance measurements of a machine similar to the prototypical machine described in Section 4. Reducing meltage can be accomplished by reducing the time the ice remains on the plate exposed to the warm evaporator. The prototype design uses gravity to pull the ice off the plate. This can be accomplished by reducing the time the ice is in contact with the warm evaporator plate by assisting gravity in pulling the ice off the plate. One manufacturer with an evaporator similar to the prototype uses a mechanical assist to push the ice off the plate faster than gravity alone. Another manufacturer's design (which is patented) has a series of plastic baffles which separate the ice cubes. The baffles ensure that the cubes are attached only on the side facing the evaporator. Removal of ice occurs after an 8-second application of hot gas to the evaporator. The claim is that there is negligible ice meltage.

Assuming the meltage rate can be reduced by 50 percent, the energy consumption required for the freeze cycle can be reduced by about 5 percent or about 230 kWh/yr. Implementation of such a device results in about $100 in additional end-user cost.

Other means, such as vibration, to free the ice from the plate more quickly can be conceived, but implementation would require a significant amount of development.

Reduced Thermal Cycling of the Evaporator

In the prototype ice machine, thermal cycling of the evaporator accounts for about 9 percent of the compressor input energy during the freeze cycle. The prototype evaporator design is a copper serpentine attached to the rear of plated copper waffle ice making surfaces. The high heat transfer coefficient of the copper is accompanied by relatively 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 percent, or about 180 - 240 kWh/yr could be realized. The end-user cost premium of such a design would be about $20. Realistically, a reduction of the thermal mass for this evaporator design would probably result in lower thermal conductance which would offset some of the projected savings for the ideal case.

High Efficiency Condenser Fan Blade

Savings of about 15% of fan power are possible with fan blades which are optimized for the ice machine application. This represents about 61 kWh for the prototypical 500-lb machine. A $1 cost premium per fan blade is assumed for the economic analysis ($16,000 tooling costs distributed over 32,000 fan blades; 100% end-user markup).

5.5.2 Economic Analysis

Table 5-20 summarizes the economic analysis for energy-savings technologies for ice machines.

Table 5-20: Economic Analysis for Ice Machines

Baseline Energy Ussage 5000 kWh

					Simple Payback Period (yrs)		
Technology Option	**End-User Cost Premium**	**Load Reduction (W)**	**Energy Reduction (kWh/yr)**	**Energy Reduction (%)**	**High Rate ($0.1834/kWh)**	**Medium Rate ($0.0782/kWh)**	**MediumLow Rate ($0.0743/kWh)**
High-Efficiency Compressor	$40	70	280	5.6	0.8	1.8	1.9
PSC Condenser Fan Motor	$24	49	200	4.0	0.7	1.5	1.6
ECM Condenser Fan Motor	$46	67	271	5.4	0.9	2.2	2.3
Thicker Insulation	$40	-	150	3.0	1.4	3.4	3.6
Reduced Meltage During Harvest	$100	-	230	4.6	2.4	5.6	5.9
Reduced Evaporator Thermal Cycling	$20	-	210	4.2	0.5	1.2	1.3
High-Efficiency Fan Blades	$1	15	61	1.2	0.1	0.2	0.2

Note: All options represent new technologies

5.5.3 Barriers to Implementation

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 undesirable.

2) As discussed, the design of ice machine evaporator coils is an involved and complicated process which does not lend itself readily to analysis. Evaporator coils have not been optimized for each model size, which reduces overall efficiency. Also, manufacturers are reluctant to modify evaporator coil designs, due to the high development costs.

3) Reductions in purge water amounts have the potential to decrease energy use. Such reductions are, however, associated with the risk of increased scale buildup, which can in itself reduce efficiency. Frequent cleaning of the machines is required to eliminate scale and reduce the risk of waterborne diseases.

4) Manufacturers require paybacks of at most 12 to 24 months for retooling costs associated with design modifications (the engineering costs are usually not taken into consideration in evaluating changes, because they do not represent capital costs). Publicly owned companies usually require less than 12 months for such paybacks. This makes manufacturers reluctant to implement product changes whose costs cannot be recovered quickly 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.

5.6 Refrigerated Vending Machine

5.6.1 Energy Saving Technologies

The energy saving technologies applicable to refrigerated vending machines are insulation improvement, high-efficiency fan motors, high-efficiency compressors, lighting improvement, and coil improvement. These technologies are discussed above in Sections 5.2.1 and 5.3.1.

5.6.2 Economic Analysis

Table 5-21 summarizes the economic analysis for energy-saving technologies for beverage merchandisers.

Table 5-21: Refrigerated Vending Machines

Baseline Energy Usage 2763 kWh[1]

						Simple Payback Period (yrs)		
	Technology Option	End-User Cost Premium	Load Reduction (W)	Energy Reduction (kWh/yr)	Energy Reduction (%)	High Rate ($0.1834 kWh)	Medium Rate ($0.0782 kWh)	Medium-Low Rate ($0.0743 kWh)
1	Thicker Insulation	$54	17	150	5.4	2.0	4.6	4.8
2	PSC Evap Fan Motor	$36	35	305	11	0.6	1.5	1.6
3	ECM Evap. Fan Motor	$56	45	395	14	0.8	1.8	1.9
4	PSC Cond Fan Motor	$36	22	67	2.4	2.9	6.9	7.2
5	ECM Cond. Fan Motor	$56	29	87	3.1	3.5	8.2	8.7
6	High-Efficiency Compressor	$16	85	260	9	0.3	0.8	0.8
7	ECM Compressor Motor	$100	62	191	7	2.9	6.7	7.0
8	Variable Speed Compressor	$150	62	413	15	2.0	4.6	4.9
9	Lighting Improvement	$30	29	255	9.2	0.6	1.5	1.6
10	High-Efficiency Fan Blades	$2	14	92	3.3	0.1	0.3	0.3
11	Combination	$72	104	778	28	0.5	1.2	1.2

[1]This baseline is for a machine with standard output lighting. For a machine with high-output lighting, the baseline is 3165 kWh.

Note: New technologies are 1, 2, 3, 4, 5, 6, 7, 9, 10; Advanced Technology is 8.

The calculations made for the economic analysis are briefly outlined below.

1) Thicker Insulation: Reduction of the 253 Btu/hr wall load (Table 4-37) by 40%(increase in total wall resistance, including inside and outside air layers, from 12 to 20 R-value, due to insulation thickness increase from 1.25 to 2.25 inches). COP of 1.72. Insulation material costs $0.33/sqft for 76 sqft of wall area. Additional $500,000 retooling costs distributed among 216,000 units: 20 percent of annual sales of 270,000 units for four years; markup to the end user of 100%.

2&3) High-Efficiency Evaporator Fan Motors: Motor costs and power requirements for the PSC and ECM motor as shown in Table 5-2. Shaded pole motor OEM cost of $7. Replacement of one 6-Watt output shaded pole motor with one 6-Watt output PSC or ECM motor. Additional compressor load savings based on the 1.72 COP. End user markup of 100%.

4&5) High-Efficiency Condenser Fan Motors: Motor costs and power requirements for the PSC and ECM motors as shown in Table 5-2. Shaded pole motor OEM cost of $7. Replacement of one 6-Watt output shaded pole motor with a 6-Watt output PSC or ECM motor. End user markup of 100%.

6) High-Efficiency Compressor: Increase in COP from 1.72 to 2.15, resulting in a 20% reduction in compressor power input. OEM cost of $8, end user markup of 100%.

7&8) ECM Compressor Motor/Variable Speed Compressor: Replacement of the existing 1/3 hp motor with an ECM motor (efficiency increase from 70% to 82%). Cost premium of $100 for the ECM motor ($100 ECM motor cost, $50 standard motor cost, 100% end-user markup). Additional 20% reduction in compressor energy usage for variable speed operation. End-user cost for variable speed controls of $50.

9) High-Efficiency Lighting: Based on discussions with lighting suppliers, the cost premium for a 48”, two-lamp electronic ballast is about $15. Assume a 100% markup for the vending machine. None of the heat from lighting enters the refrigerated space. Efficacy improves from 65 LPW to 87 LPW (25% savings). Savings calculated for a machine with standard (not high-output) lighting.

10) High-Efficiency Fan Blades: Reduction of 15% in evaporator and condenser fan load. Additional savings for evaporator fan due to case load reduction based on 1.72 COP. Cost premium $1 per fan blade (tooling costs of $16,000 distributed over 32,000 fan blades; 100% end-user markup).

11) Combination: High-Efficiency Compressor and ECM Evaporator Fan Motor: Savings for the improved compressor as described for #6. The ECM evaporator

fan motor savings are reduced because conversion of the reduction in case load to compressor power savings involves the improved COP.

5.6.3 Barriers to Implementation

Issues affecting vending machines are similar to those affecting beverage merchandisers.

1) Most vending machines are owned by bottling companies, who do not pay utility bills for the buildings where the units are located. This effectively eliminates any incentive of the machine owners to select machines with higher priced, more efficient features.

2) Energy costs are small compared with typical beverage sales revenues. Energy costs for a canned beverage vending machine (for Raleigh, NC, representing average electricity costs) are about $240 annually. The total revenues for beverage sales average about $8000 for such a machine. Although the energy costs are not completely insignificant, they do not represent a large part of the sales revenues. This reduces awareness of energy as a major concern. It also increases the tendency to disregard energy issues in evaluating sales-boosting design changes, such as an increase in lighting intensity.

3) Space in vending machines is tight. Therefore increases in insulation thickness are undesirable. A decrease in storage volume is likely to reduce the sales capacity of a given machine, which is unacceptable for the reasons mentioned above. Vending machines must fit through doorways, so there is little room for external dimension increase.

5.7 Walk-In Coolers and Freezers

5.7.1 Energy-Saving Technologies

This section describes the energy saving technologies which are applicable to walk-in coolers and freezers, and their energy savings potential. Detailed calculation of savings and economics and a summary table are presented in Section 5.7.2.

The energy saving technologies examined are high-efficiency fan motors, hot gas defrost, economizer cooling, floating head pressure, ambient subcooling, electronic ballasts for lighting, non-electric antisweat, advanced control of antisweat and defrost, thicker insulation, and external heat rejection.

Current Technologies

Hot Gas Defrost

Defrost is required for all walk-in freezers, and is used for some walk-in coolers. Defrost heating for a freezer is applied not just to the evaporator coil, but also to the coil's drip pan, in order to assure that condensate flows freely from the pan into the drain. All three defrost modes discussed in Section 4.1.1, (off-cycle, electric, and hot gas) are commercially available for walk-ins. Electric defrost is more popular than hot gas defrost for low temperature applications. Hot gas defrost eliminates the coil and pan heating electricity. Its economics for the prototypical walk-in freezer are presented in Section 5.7.2.

Thicker Insulation
The insulation thickness for the prototypical walk-ins is 4". Increase of the insulation thickness to 5" is evaluated in the economic analysis of Section 5.7.2. This benefits the freezer significantly more than the cooler.

New Technologies

Floating Head Pressure
Standard operation of a condensing unit serving a walk-in involves control of head pressure such that the condensing temperature does not fall below a predetermined level. This is done either by cycling the condenser fan or with a control valve which partially floods the condenser, thus reducing its effective heat transfer area. The control assures sufficient pressure for flow of liquid refrigerant through the thermostatic expansion valves, and assures that the refrigerant remains in liquid form prior to expansion.

As in supermarkets, the refrigeration systems serving walk-ins can be controlled for lower head pressure and condenser temperature if expansion valves with better control over a wide pressure range are used. Today's balanced-port expansion valves have this capability. This technology is applicable to the prototypical walk-in cooler discussed in Section 4.7, which has external heat rejection. Savings are possible because the refrigerant pressure ratio is significantly less for most of the year.

The savings for this measure are calculated for Washington D.C., a city with climate assumed to represent an average for the U.S. Implementation of this technology results in a reduction of duty cycle for the compressor and condenser fans from 66% to 50%. The average annual duty cycle was calculated using a bin analysis, with the following assumptions.

- The compressor power is equal to the cooling load divided by the COP.
- The COP is determined for an R22 cycle with varying condensing temperature assuming that condenser temperature is ten degrees above ambient temperature. For standard head pressure control, 90ºF is the lower limit on condenser temperature. Hence, below 80ºF ambient, the COP no longer improves. For the floating head pressure case, the condenser temperature is allowed to float down to 70ºF. Condenser exit subcooling is assumed to be zero, and the temperature of the vapor exiting the liquid-suction heat exchanger is assumed to be halfway between the evaporator and condenser temperatures. The compressor's isentropic efficiency is set in order to match the design condition at 95ºF ambient temperature.

- The cooling load is assumed to vary with outdoor temperature as follows. The wall and infiltration loads are assumed to be proportional to exterior/interior temperature difference and to be as listed in Table 4-43 at the design condition (95ºF ambient). The average total refrigeration load at design conditions (including cooldown of warm product and frequent door openings) is assumed to be equal to 90% of the compressor capacity. All loads other than the walls and infiltration are assumed to be constant.

Ambient Subcooling

Ambient subcooling can be used for a walk-in with external heat rejection in situations where the liquid refrigerant can be further cooled by the ambient air. This will be effective when the head pressure level is not allowed to float down. For instance, if on a 40ºF day the head pressure control is maintaining a 90ºF condenser temperature, the condensed liquid can be cooled further, thus increasing evaporator capacity. Implementation of ambient subcooling requires the installation of an additional heat exchanger for subcooling downstream of the liquid receiver. One concept for such a system (the Sierra system manufactured by Russell) involves combining the condenser, liquid receiver, and subcooler into a single unit. The system is claimed to have a cost which is comparable to that of a conventional system.

The savings potential for ambient subcooling has been calculated for Washington D.C. Implementation of this technology results in a reduction of duty cycle for the compressor and condenser fans from 66% to 58%. The calculation assumptions are as outlined for the floating head pressure calculation above, with the following differences.

- The COP calculation assumes that head pressure control limits the condenser temperature to be no less than 90ºF. For the baseline, condensed liquid with zero subcooling is assumed to enter the liquid-suction heat exchanger. For ambient subcooling, the liquid is first subcooled with ambient air. The subcooling heat exchanger is assumed to be 75% effective: the liquid temperature reduction is assumed be 75% of the difference between the condenser temperature and the ambient temperature. For both the baseline case and the ambient subcooling case, the temperature of the gas leaving the liquid-suction heat exchanger is assumed to be halfway between the evaporator temperature and the temperature of liquid entering the heat exchanger.

External Heat Rejection

As dicussed in Section 4.7.2, a percentage of smaller walk-in coolers and freezers with packaged refrigeration systems are installed in interior spaces and reject heat to the interior space. Efficiency of the these systems could be improved by external rejection of heat. This would require external placement of the condenser or of the entire condensing unit.

As discussed, the reason for the use of packaged systems which reject heat internally is the convenience of installation. There is no need to install or to provide power for an external condensing unit or condenser, and no need to lay the interconnecting refrigerant lines. Furthermore, even though precharged units are available, many installations involve more complicated routing of refrigerant lines, which makes the use of precharged units more difficult. This is not required when installing a walk-in with an entirely packaged refrigeration system.

External heat rejection will not only reduce the electricity usage of the walk-in refrigeration system, but will also reduce the heat load in the internal space. This heat can in some cases be removed by an exhaust fan, but the use of a dedicated exhaust fan defeats the goal of convenient installation. In any case, use of internal heat rejection will 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.

The compressor electricity savings for external heat rejection have been estimated for the prototypical storage-only walk-in freezer. For the representative city, Washington D.C., the compressor and condenser duty cycle is reduced from 70% to 61%. Pertinent assumptions are as follows.

- Compressor power requirement is equal to refrigeration load divided by COP
- Refrigeration load for the design condition (90ºF building temperature) is assumed to be 90%. The wall loss and infiltration loads (see Table 4-43) are proportional to temperature difference between the building interior and the walk-in interior. The average annual building temperature is 75ºF.
- COP is calculated for an R-404A cycle with evaporator and condenser temperature differences as indicated for the prototypical walk-in freezer in Table 4-40. For the baseline calculation, the condenser temperature is fixed at 113ºF. For the external heat rejection calculation, head pressure control that limits condenser temperature to 70ºF and above is used. Liquid subcooling is zero. The suction gas temperature is halfway between condenser temperature and evaporator temperature.

Economizer Cooling

Walk-in coolers typically have interior temperatures between 35ºF and 40ºF. In Northern areas of the U.S., the external temperatures are lower than this for many hours of the year. In order to maintain design simplicity, walk-in refrigeration systems generally are not designed to take advantage of this opportunity for free cooling. A economizer cooling system consists of access holes to the outside, a supply fan to move cold outdoor air into the walk-in and the required controls. Controls are required in order to properly sequence the compressor and the economizing cooling fan. Ductwork would be required if the walk-in is located in the building interior, and dampers are needed to seal the walk-in from external air during warmer months. Such a system is marketed as part of a package of walk-in energy-saving controls and hardware for retrofit and new applications by Air Enterprises, Inc. of Essex, Vt.

The savings potential is a strong function of climate, specifically, the number of hours during the year when outdoor temperature is 30ºF or less. The potential savings for the option have been calculated for the prototypical walk-in cooler with merchandising doors using a bin analysis with weather data for cities which represent five climate zones. The resulting reduction in compressor power requirement for the cooler for the five cities are presented in Table 5-22 below.

Table 5-22: Compressor Electricity Reductions With Economizer Cooling

Climate Zone	Heating Degree Day Range	Cooling Degree Day Range	Representative City	Compressor Electricity Reduction (%)
1	>7000	<2000	Minneapolis	26%
2	5500 to 7000	<2000	Boston	16%
3	4000 to 5500	<2000	Washington	8%
4	<4000	<2000	Atlanta	2%
5	<4000	>2000	Dallas	2%

Calculation assumptions are as follows.

- The compressor power requirement at outdoor temperatures greater than 30ºF is equal to the cooling load divided by the COP. Up to 30ºF the compressor power is equal to zero.
- The COP is determined for an R22 cycle as described above in the discussion regarding floating head pressure control. For the economizer cooling calculation, standard head pressure control is assumed, which imposes a 90ºF lower limit on condenser temperature.
- The cooling load is calculated as described above for the floating head pressure control caculation.

High Efficiency Fan Blades

High-efficiency fan blades are discussed for supermarket applications in Section 5.1.1. Improved-design fan blades could result in a 10 to 20 percent reduction in fan power requirements in walk-in applications. The economics of improved condenser and evaporator fans are presented for both the prototypical walk-in cooler and the freezer in Section 5.7.2.

High Efficiency Fan Motors

The evaporator fans of both prototypical walk-ins described in Section 4.7.2 use shaded-pole motors. The condenser fans use capacitor-start induction-run (CSIR) and PSC motors. The power requirements could be reduced with the use of PSC or ECM motors for the evaporator fans and with ECM motors for the condenser fans. Improvement in evaporator fan motor efficiency will also reduce the compressor power requirement by reducing the refrigeration load. The economics of these options are presented in Section 5.7.2.

Electronic Ballasts

The fluorescent display lighting of the prototypical walk-in cooler uses standard magnetic ballasts. The lighting power requirement could be reduced by 25% with the use of electronic ballasts. This will also reduce the refrigeration load and the compressor power because the lighting equipment is in the cooled space.

Antisweat Heating Control

Antisweat heating around the access door perimeter of a walk-in freezer and around the perimeter of merchandising doors in walk-ins used for merchandising is required to eliminate condensation and frosting of these locations. Antisweat heating is generally done with electric resistance heaters. For the prototypical walk-ins of this study, the heaters are constantly on.

Control of antisweat with a dew point sensor is discussed in Section 5.1.1 for supermarket reach-in display cases. Such a control scheme is also appropriate for walk-in antisweat heating.

Evaporator Fan Shutdown

Standard control of walk-in evaporator fans is to keep them running at all times, even though compressor duty cycle is in the 60 to 70 percent range. In some cases this is required to maintain even temperatures by keeping the air moving. However, this need would depend on the application, and could be better served by a smaller fan which does not draw air through the evaporator, thus increasing the fan power requirement. The economic analysis for this technology assumes that fan operation time can be reduced 20% for both the cooler and freezer.

Advanced Technologies

Non-Electric Antisweat

The use of hot gas or liquid for antisweat heating is discussed in Section 5.3.1 for Reach-in Freezers. The concept should also be applicable to walk-ins. It would require piping of a separate hot gas loop to an antisweat circuit rather than directly to the condenser, or the diversion of the liquid flow through such an antisweat circuit prior to piping to the expansion valve. The economics of such a system for the prototypical walk-ins are presented in Section 5.7.2.

Demand Defrost Control

The use of demand defrost control is discussed in Section 5.1.1 for supermarkets. The concept is also applicable to low temperature walk-ins. Economics are presented in Section 5.7.2.

5.7.2 Economic Analysis

Table 5-23 below summarizes the economic analyses for energy-saving technologies for the prototypical walk-in cooler.

Table 5-23: Economic Analysis- Walk-in Coolers

	Technology Option	Reduction (kWh)	Reduction (W)	Energy Reduction (%)	Cost Premiums	Simple Payback Periods (yrs) High Rate ($0.1834 kWh)	Medium Rate ($0.0782 kWh)	Medium-Low Rate ($0.0743 kWh)
1	Floating Head Pressure	7,744	-	18%	$174	0.1	0.3	0.3
2	Ambient Subcooling	3,872	-	9%	$525	0.7	1.7	1.8
3	Economizer Cooling	2,393	-	6%	$3,750	8.5	20	21
4	Antisweat Heat Controls	1,004	-	2%	$500	2.7	6.4	6.7
5	Thicker Insulation	190	22	0.4%	$428	12.3	28.8	30.3
6	Evaporator Fan Shutdown	1,811	-	4%	$100	0.3	0.7	0.7
7	PSC Evaporator Fan Motors	2,102	240	5%	$160	0.4	1.0	1.0
8	ECM Evaporator Fan Motors	3,574	408	8%	$352	0.5	1.3	1.3
9	ECM Condenser Fan Motors	925	160	2%	$60	0.4	0.8	0.9
10	Electronic Ballasts	440	76	1%	$80	1.0	2.3	2.4
11	High Efficiency Fan Blades	2,666	381	6%	$120	0.3	0.6	0.6
12	Non-electric antisweat	2,628	300	6%	$750	1.6	3.6	3.9
13	Combination*	13,377	568	32%	$1,111	0.5	1.1	1.1

*Includes technologies #1, #2, #6, #8, and #9

The calculations made for the economic analysis on the walk-in cooler are briefly outlined below.

1) <u>Floating Head Pressure</u>: Reduction of the duty cycle of the compressor and condenser fans from 66% to 50%, as discussed in Section 5.7.1 above. Cost premium for the measure, which covers the modified expansion valves and modified head pressure control, is assumed equal to the per-circuit cost for floating head pressure in supermarkets ($8000 for 46 circuits).

2) <u>Ambient Subcooling</u>: Reduction of the duty cycle of the compressor and condenser fans from 66% to 58%, as discussed in Section 5.7.1 above. As mentioned, one commercially available system providing ambient subcooling is claimed to have no associated cost premium. A 10% premium of 75% of the $7,000 refrigeration system list price is used as a conservative estimate of the cost premium.

3) <u>Economizer Cooling</u>: Energy savings based on the use of ambient air for cooling, during the hours when the ambient temperature is below 30° F. This represents 740

hours for the assumed typical city Washington D.C. The percent reduction in compressor power is 8% (see Table 5-23 above). This is a conservative estimate of a national average reduction: the range is from 2% to 26%, depending on the climate zone. End-user cost of components for the installation: dampers and actuators $1,200, fan $250, controller $300, thermostats $100, miscellaneous (wiring, etc.) $300. Additional installation cost based on two days work for two men at $50/hr each.

4) Antisweat Control: Energy Savings are based on a reduction of the antisweat heating load by 1/3. Additional savings in compressor power based on the 3.42 COP are calculated assuming that 1/2 of the antisweat heating load contributes to the internal load. The installed cost for the required sensor and controller is about $500.

5) Thicker Insulation: Reduction of the 1,270 Btu/hr wall load (see Table 4-43) by 20% through increasing the total wall resistance from R-28.6 to R-35.75, due to the insulation thickness increase from 4 inches to 5 inches. Compressor load savings are based on the 3.42 COP. Insulation material OEM costs of $0.33/sqft for 648 sqft of wall area, with no additional tooling or engineering costs for the thicker panels. Markup to end-user of 100%.

6) Evaporator Fan Shutdown: Reduction of the evaporator fan duty cycle from 100% to 80%. Fan power savings and additional compressor power savings based on the 3.42 COP. Cost premium based on $50 OEM cost for a contactor to control the evaporator fans, 100% markup to the end-user.

7&8) High Efficiency Evaporator Fan Motors: Replacement of two 1/20 hp (37W output) shaded pole motors with two 1/20 hp PSC or ECM motors. OEM motor costs and power requirements for the PSC and ECM motors as shown in Table 5-2. Shaded pole motor power requirement 100W and OEM cost $30. Additional compressor load savings based on a 3.42 COP.

9) High Efficiency Condenser Fan Motors: Replacement of a 1/2 hp PSC motor with a 1/2 hp ECM motor. OEM motor costs and power requirements as shown in Table 5-2. Markup to the end-user of 100%. Energy savings are based on a 66% duty cycle.

10) Electronic Ballasts for Display Lighting: OEM cost premium for electronic ballast serving two 60" fluorescent lamps of $20. Energy usage reduction of 25% of the 236W display lighting load. Added compressor power savings based on the 3.42 COP.

11) High Efficiency Fan Blades: Fan power savings of 15%. Additional compressor power reduction due to evaporator fan load reduction based on the 3.42 COP. OEM cost of $4 for evaporator fan blades and $14 for condenser fan blades. High efficiency blades will double cost, with 100% markup to end-user.

12) Non-electric Antisweat: Elimination of 2,628 kwh of electric antisweat. $1,000,000 of engineering and retooling cost distribuyed over 8,000 units (assuming 2,000 annual sales for 4 years). End-user markup of 100%. Additional 250 ft. copper piping @ $1/ft OEM cost.

13) Combination: Floating Head Pressure, Ambient Subcooling, Evaporator Fan Shutdown, ECM Evaporator and Condenser Fans: Reduction in evaporator fan electricity usage of 4,257 kWh. Reduction of the duty cycle of the compressor and condenser fans from 66% to 48%, based on the reduced load of the evaporator fans, the reduced head pressure for ambient temperatures less than 80ºF, and further reduction in liquid temperature for ambient temperatures less than 60ºF. Additional reduction of the condenser fan electricity usage due to use of the ECM motor.

Table 5-24 below summarizes the economic analyses for energy-saving technologies for the prototypical walk-in freezer.

Table 5-24 : Economic Analysis - Walk-in Freezers

						Simple Payback Periods (yrs)		
	Technology Option	Reduction (kWh)	Reduction (W)	Energy Reduction (%)	Cost Premiums	High Rate ($0.1834 kWh)	Medium Rate ($0.0782 kWh)	Medium-Low Rate ($0.0743 kWh)
1	External Heat Rejection	1,446	-	9%	$800	3.0	7.1	7.5
2	Hot Gas Defrost	589	1600	4%	$83	0.8	1.8	1.9
3	Defrost Controls	368	-	2%	$100	1.5	3.5	3.7
4	Antisweat Heat Controls	1,008	-	6%	$500	2.7	6.3	6.7
5	Thicker Insulation	566	65	4%	$116	1.1	2.6	2.8
6	Evaporator Fan Shutdown	631	-	4%	$100	0.9	2.0	2.1
7	PSC Evaporator Fan Motors	1,682	192	11%	$60	0.2	0.5	0.5
8	ECM Evaporator Fan Motors	2,208	252	14%	$100	0.2	0.6	0.6
9	PSC Condenser Fan Motors	779	127	5%	$22	0.2	0.4	0.4
10	ECM Condenser Fan Motors	1,067	174	7%	$48	0.2	0.6	0.6
11	High Efficiency Fan Blades	776	103	5%	$33	0.2	0.5	0.6
12	Non-electric Antisweat	2,015	230	13%	$225	0.6	1.4	1.5
13	Combination*	5,097	2026	33%	$1,131	1.2	2.8	3.0

*Includes technologies #1, #2, #6, #8, and #10

1) External Heat Rejection: Reduction of the duty cycle of the compressor and condenser fans from 70% to 61%, as discussed in Section 5.7.1 above (based on weather data for Washington D.C.). Head pressure control to maintain a minimum condenser temperature of 70ºF. Cost premium for the measure will be associated primarily with additional installation labor. Assume one additional day required, resulting in an $800 premium (2 men @ $50/hr).

2) Hot Gas Defrost: Eliminates the coil and drip pan electric defrost energy of 2000W and 735 kWh/yr (see Table 4-41). Defrost heat delivered instead with the compressor, effectively operating with a COP of 5 (heat pumped from the 75ºF average building interior temperature to an assumed 50ºF evaporator and pan temperature). Cost premium for the measure is assumed equal to the per-circuit cost for hot gas defrost in supermarkets ($3800 for 46 circuits).

3) Defrost Control: Eliminates half of the 286 Btu/hr average defrost load (see Table 4-43) during 6 cooler months per year. Additional compressor load savings based on the 1.00 COP. OEM cost of $50 based on the use of two sensors and controls; end user markup of 100%.

4) Antisweat Control: Energy Savings are based on a reduction of the antisweat heating load by 1/3. Additional compressor load savings based on the 1.00 COP, assuming that half of the antisweat heating load contributes to the internal box load. The installed cost for the required sensor and controller is about $500.

5) Thicker Insulation: Reduction of the 1,103 Btu/hr wall load (Table 4-43) by 20% through increasing the total wall resistance from 30 to 37.5, due to the insulation thickness increase from 4 inches to 5 inches. Compressor load savings are based on the 1.00 COP. Insulation material OEM costs of $0.33/sqft for 353 sqft of wall area, with no additional tooling or engineering costs for the thicker panels. Markup to end-user of 100%.

6) Evaporator Fan Shutdown: Reduction of the evaporator fan duty cycle from 100% to 80%. Fan power savings and additional compressor power savings based on the 1.00 COP. Cost premium based on $50 OEM cost for a contactor to control the evaporator fans, 100% markup to the end-user.

7&8) High Efficiency Evaporator Fan Motors: Replacement of two 1/40 hp (20W output) shaded pole motors with two 1/40 hp PSC or ECM motors. OEM motor costs and power requirements for the motors as shown in Table 5-2. Additional compressor load savings based on a 1.00 COP.

9&10) High Efficiency Condenser Fan Motors: Replacement of the 1/6 hp CSIR motor with a 1/6 hp PSC or ECM motor. OEM motor costs and power requirements for the PSC and ECM motors as shown in Table 5-2. CSIR motor power requirement 329W and OEM cost $40. Savings based on a 70% fan duty cycle.

11) High Efficiency Fan Blades: Fan power savings of 15%. Additional compressor power reduction due to evaporator fan load reduction based on the 1.00 COP. OEM cost of $1.25 for the evaporator fan blades and $14 for the condenser fan blade. High-efficiency blades will double cost, with 100% markup to the end-user.

12) Non-electric Antisweat: Elimination of 2,015 kWh of electric antisweat. $500,000 of engineering and retooling cost distributed over 8,000 units (assuming 2,000 annual sales for 4 years). Additional 50 ft copper piping @ $1/ft OEM cost. End-user markup of 100%.

13) Combination: External Heat Rejection, Hot Gas Defrost, Evaporator Fan Shutdown, and ECM motors for the evaporator and condenser fans: Reduction of the evaporator fan electricity usage from 1577 kWh to 378 kWh. Reduction of the duty cycle of the compressor and condenser fans from 70% to 54% (due to external heat rejection and the reduced evaporator fan load), resulting in 2,486 kWh savings. Further reduction of 823 kWh in condenser fan electricity due to the use of an ECM motor. Delivery of defrost load utilizing the compressor with a COP of 5 rather than the COP of 1, representing 589 kWh savings.

Table 5-25 below shows development of estimates of the potential annual primary energy savings for the energy saving technologies discussed in this section. For the calculation, it is assumed that half of the combination unit energy usage is associated with each temperature level. Sixty-five percent of walk-ins are assumed to have external heat rejection. Furthermore, 30% of coolers, 20% of freezers, and 10% of combination units are assumed to have merchandising doors. The primary energy usages for the walk-in types are 96 trillion Btu for coolers, 63 trillion Btu for freezers, and 21 trillion Btu for combination units.

Table 5-25: Walk-In Energy Savings Potential

	Technology Option	Energy Savings (%)	Cooler Applica-bility (%)	Freezer Applica-bility (%)	Combo Applicability (%)	Potential Primary Energy Savings (Trilliion Btu)
C1	Floating Head Pressure	18%	65	65	65	21
F11/C1	High Efficiency Fan Blades	5%[f] 6%[c]	0 100	100 0	50 50	10
F8/C8	ECM Evaporator Fan Motors	14%[f] 8%c	0 100	100 0	50 50	19
F10/C9	ECM Condenser Fan Motors	7%[f] 2%c	0 100	100 0	50 50	7
F6/C6	Evaporator Fan Shutdown	4%[f] 4%[c]	0 100	100 0	50 50	7
C2	Ambient Subcooling	9%	65	65	65	11
F2	Hot Gas Defrost	4%	0	100	50	3

	Technology Option	Energy Savings (%)	Cooler Applica-bility (%)	Freezer Applica-bility (%)	Combo Applicability (%)	Potential Primary Energy Savings (Trilliion Btu)
C10	Electronic Ballasts	1%	30	20	10	0.4
F5/C5	Thicker Insulation	4%[f] 0.4%[c]	0 100	100 0	50 50	3
F3	Defrost Controls	2%	0	100	50	1
F4/C4	Antisweat Heat Controls	6%[f] 2%[c]	0 39	100 0	50 5	5
F1	External Heat Rejection	9%	35	35	35	6
C3	Economizer Cooling	6%	100	0	50	6
F12/C12	Non-electric Antisweat	13%[f] 6%[c]	0 100	100 0	50 50	16
F13/C13	Combinations* of Technologies	33%[f] 32%[c]	0 100	100 0	50 50	58

*See Tables 4-23 and 4-24 for a listing of included technologies for each temperature level

[f] for freezers [c] for coolers

5.7.3 Barriers to Implementation

The major hurdle to implementation of efficient equipment is the first cost required for most energy-saving technologies. This problem applies also to walk-ins. Even if life-cycle costs for the energy-efficient equipment is lower, the following barriers still create market acceptance difficulties:

1. Purchase decisions for walk-ins are generally not made based on life-cycle cost or even payback considerations. A general contractor involved in a project where a walk-in is being installed has incentive to select the lowest cost equipment which meets specifications. People in charge of selecting the equipment are generally not in charge of operating it, and there is little communication between these groups.
2. Frequently, there is insufficient cash flow at the time of equipment purchase for consideration of future benefits to sway the decision.
3. There may be questions regarding the viability of the business for the long term, which further reduces the importance of payback after a period of years. As discussed in previous sections, the prospects for new start-up restaurants are not solid, and for these establishments the lowest cost equipment is generally selected. In many cases, used refrigeration equipment is purchased if available.
4. End-users are not convinced or cannot properly assess, whether the added cost of energy-saving technologies will be paid back through savings. This can be caused by a few factors as described below.
   - The complexity of the technology issues relating to refrigeration energy usage and life cycle cost.
   - The complexity of energy cost structures. Commercial electric rates can be complex, having both demand and usage components which can vary with time

of electricity usage. Furthermore, some utilities have a variety of available electric rates, which add further complexity to the question of what energy actually costs. A further complication arises for chains of convenience stores or restaurants which are trying to apply standard design specifications to a group of establishments having different electric utilities.

- The lack of resources among end-users for confident and accurate assessment of either the available technology options or the energy costs. In many cases, new equipment is purchased when the old equipment fails, and there is no time to analyze in detail the economics of the purchase decision. For a new venture, getting the business started quickly is more important than energy cost savings over the long term.

5. The walk-in market is very competitive, with many suppliers, none of whom have a dominant market position. The primary basis of differentiation amongst different competitors is generally first cost. Cost differential as little as $5 to $10 can make a difference in the purchase decision.[5] 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. This variety of options increases the competitiveness of the market and also makes assessment of the best options difficult.

There are a few technical barriers to implementation of the energy-saving options discussed in Section 5.7.1.

- The advanced technologies such as non-electric antisweat and demand defrost control need to be investigated more thoroughly to determine whether they can be adapted to walk-ins.
- Walk-ins generally consist of an insulated box section which has unit 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 separated design and manufacture of the refrigeration systems and the walk-in boxes would make non-electric antisweat difficult, because it would require the walk-in box manufacturer to install refrigerant tubing and provide connections for the refrigeration system. The added installation complexity would provide a further barrier to this technology option.

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

- A number 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. Most refrigeration service technicians providing service for walk-in systems would have

---

[5] Source: personal communication with Paul Wilson of Masterbilt, 4/96

difficulty in properly maintaining systems with such controls.   A training effort would be required in order to enhance understanding of these technologies and to convince technicians that systems with such controls will work reliably.

- As mentioned in Section 5.7.1, the use of floating head pressure control would require the use of balanced-port expansion valves which allow satisfactory refrigerant flow over a range of head pressures. Such valves are generally not used for walk-ins because fixed head pressure control is used. Implementing floating head pressure would involve a coordination of a refrigeration controls manufacturer, the refrigeration system manufacturer, and the walk-in manufacturer in an effort which diverges from current practice. Such cooperation is possible, but takes initiative and represents a barrier to implementation.
- ECM motors of appropriate sizes for walk-in fans are not yet generally available. Even if a unit cooler with ECM motors was installed, finding a replacement in case the motor failed 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.
- As mentioned, the market for walk-ins is fairly fragmented. There are many manufacturers to convince in order to successfully introduce an energy-saving technology into a significant portion of the market. Furthermore, there is no trade association which represents the walk-in manufacturers and 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.

6. References

1 Progressive Grocer, 62nd Annual Report of the Grocery Industry, April 1995 supplement

2 "Evaluation of Alternative Designs for Engine-Driven Supermarket Refrigeration", Energy International, Inc. for GRI, January 1991

3 "Guide for the Selection of Supermarket Refrigeration Systems", Foster-Miller, Inc. for EPRI, March 1990 by D. H. Walker, T. L. Tsaros, and G. I. Deming

4 "Assessment of Refrigerated Display Cases", Energy International, Inc. for EPRI, May 1994

5 "An Electric Utility's Adventures in Commercial Refrigeration", Heating, Piping, and Air-Conditioning, October 1994, pp. 43-48

6 "Field Testing of High-Efficiency Supermarket Refrigeration", Foster-Miller, Inc. for EPRI, December 1992 by D. Walker

7 "TAG Technical Assessment Guide, Volume 2: Electricity End Use, Part 2: Commercial Electricity Use, 1992 edition, by Joseph A. Pietsch for EPRI, December 1992

8 Progressive Grocer's 1995 Marketing Guidebook, published by Trade Dimensions, Division of Interactive Market Systems, Inc., 1994

9 Vending Times' Census of the Industry 1994, by Victor Lavay, et. al., August 1994

10 Dixie-Narco cold beverage venders list price sheet, effective March 1, 1994.

11 Personal communication between Doug Drenten (Dixie-Narco) and Matthew Foran (ADL) on May 31, 1994

12 Personal communication between Bob McGarrah (PepsiCo, Inc.) and Matthew Foran (ADL) on May 31, 1994

13 Personal communication between Doug Drenten (Dixie-Narco) and Robert Zogg (ADL)

14 Personal communication between Vic Lavay (Vending Times) and Matthew Foran (ADL) on June 2, 1994

15 “Single Compressors with Attached Air Cooled Condensers”, Paul Adams, ASHRAE Journal, November 1990, p.22

16 “Energy Efficient Alternatives to Chlorofluorocarbons”, DOE Report DOE/ER/30115-112, June 1993

17 “Commercial Refrigeration Baseline Study”, Easton Consultants, Inc., September 30, 1993

18 Personal communication between John O’Hare (Dixie-Narco) and Matthew Foran (ADL) on May 20, 1994

19 Personal communication between Eddie King (Coca-Cola Company) and Matthew Foran (ADL) on June 6, 1994

20 “Characterization of Commercial Building Appliances”, ADL for DOE, August 1993 by Rusi Patel, et. al.

21 Personal communication between Matthew Foran (ADL) and representatives of Beverage-Air: Mike Branz (5/31/94), Joe Cook (5/31/94), and Jim Rhodenbaugh (7/8/94)