Energy Savings Potential for Commercial Refrigeration Equipment

Energy-Saving Technologies - Supermarket Refrigeration

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.

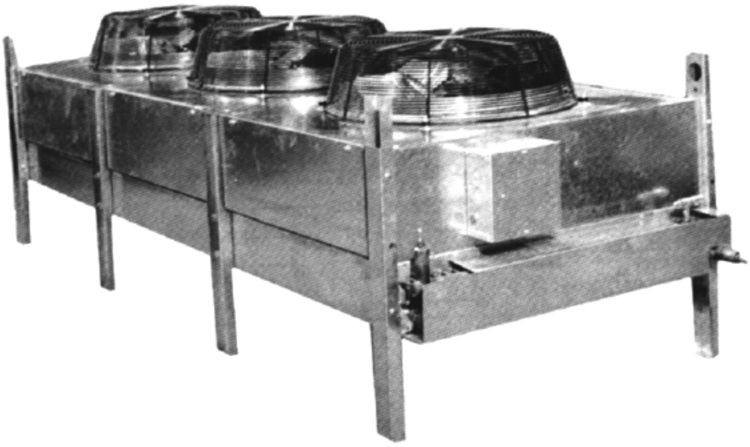


Figure: Air-Cooled Condenser

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. An industrial-grade evaporative condenser is shown below:

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 90^0F to 95^0F degrees. Floating head pressure systems allow condensing temperatures down to about 70^OF. The so-called very low head pressure systems allow even further condenser temperature reductions, down to 50^OF 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^OF 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 nthalpy 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 below. 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.

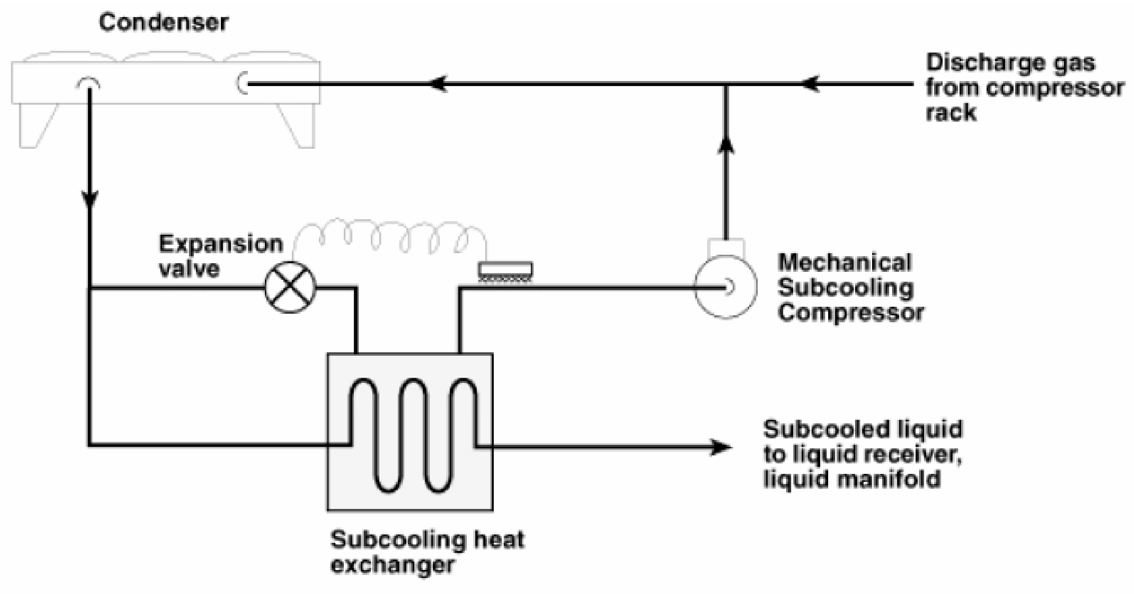


Figure: Mechanical Subcooling

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

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 below). 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.

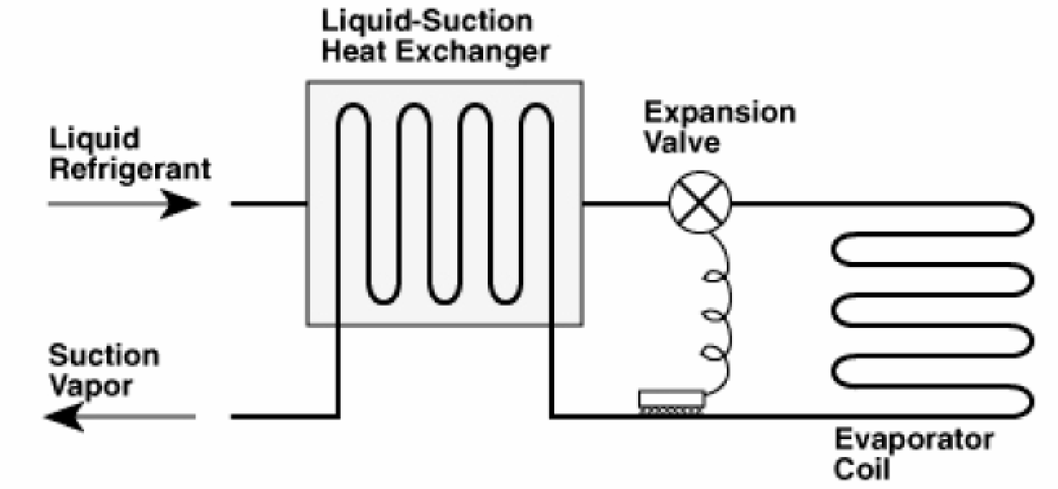


Figure: Liquid-Suction Heat Exchanger

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 ontime 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. 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.

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.

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 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.

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. The higher cost would apply to plastic blades.

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. 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. The insulation thickness increase will reduce the wall load by about 38%. 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.

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.

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. 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.

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.

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.

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.

Defrost Control

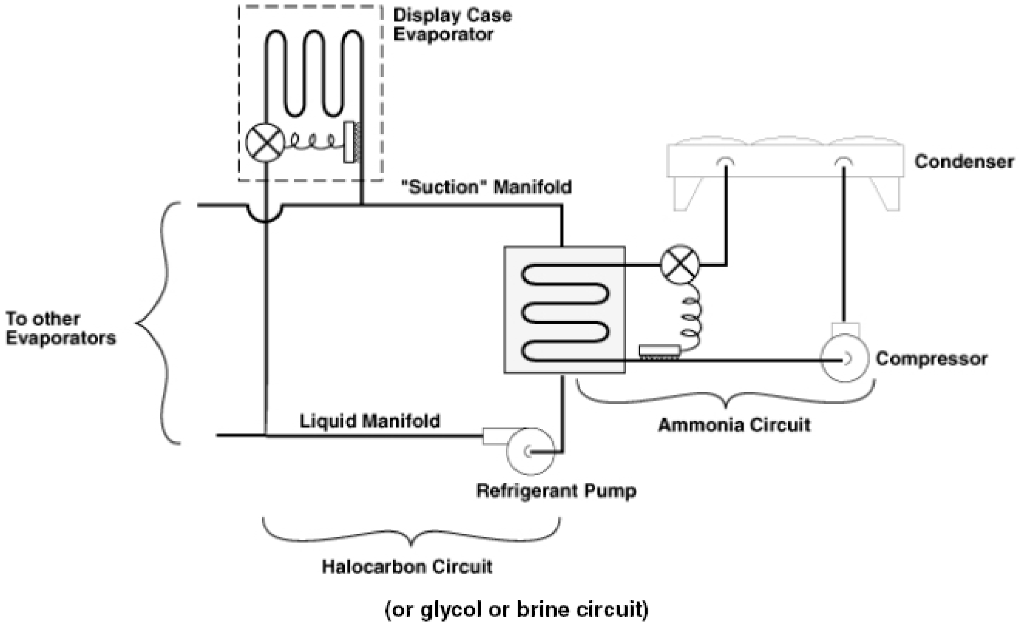
Control of defrost involves (a) initiation of the defrost cycle and (b) termination of the cycle. In the past both were done with a timer. The cycle was started when it was expected that a large frost layer had developed, and the cycle duration was set long enough to ensure complete defrosting for worst-case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

Alternative Refrigerants

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 below). The figure shows a secondary circuit using a halocarbon refrigerant. This would require less display case modification than use of glycol or brine.



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 enginedriven systems because installation and maintenance costs will strongly favor larger engines. The least expensive engines for industrial applications are automotivederivative; 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.

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. 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.

1. Supermarket Refrigeration

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. The limited savings opportunities reflect the sensitivity to energy efficiency in machine room equipment selection decisions.

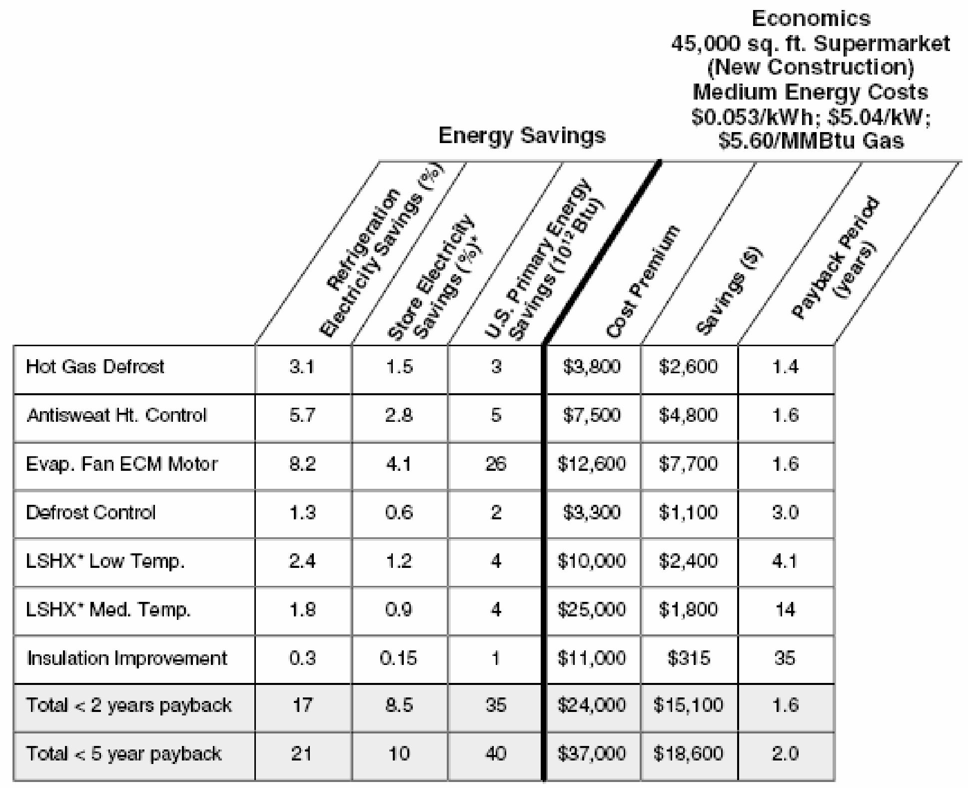
Reduction of 1% of overall usage with a two year payback2 is possible with increased use of evaporative condensers, a technology which currently has little market penetration. This technology should not have a cost premium with respect to air-cooled condensers, but the savings will be overshadowed by 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.

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 below. Savings of 14% (45 trillion Btu3) 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 liquidsuction 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.

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

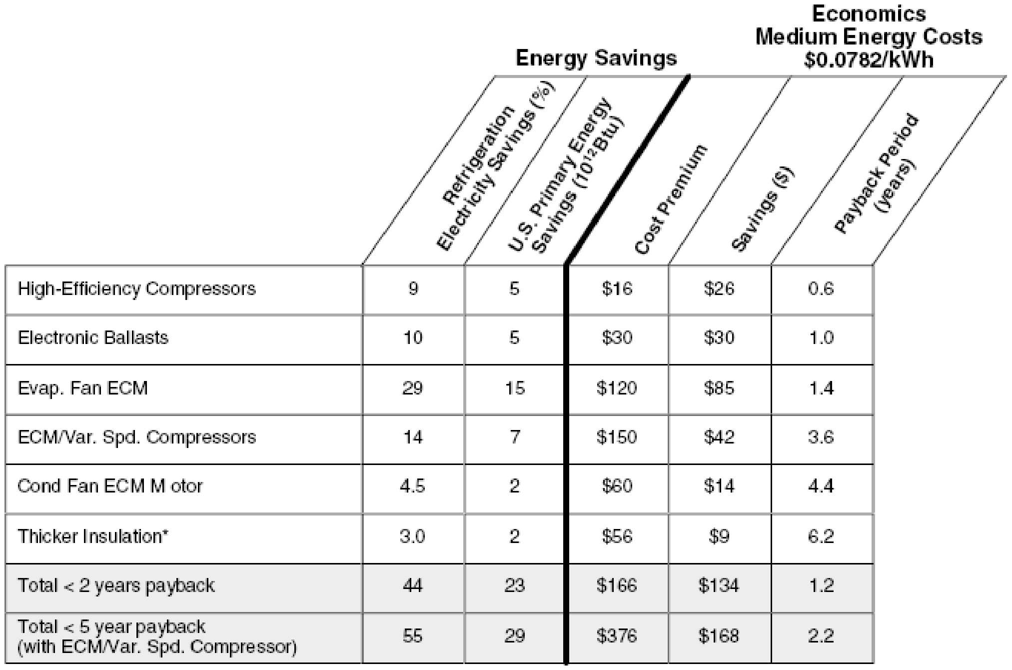
*LSHX = Liquid Suction Heat Exchangers

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

Beverage Merchandiser Energy Savings



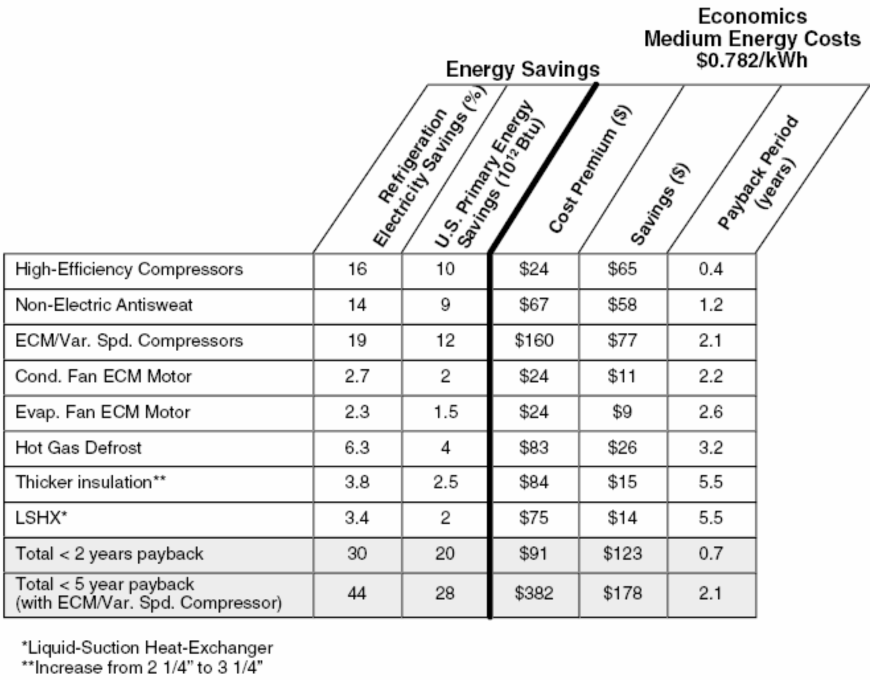
	Energy Savings		Economics Medium Energy Costs $0.0782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium	Savings ($)	Payback Period (years)
High-Efficiency Compressors	9	5	$16	$26	0.6
Electronic Ballasts	10	5	$30	$30	1.0
Evap. Fan ECM	29	15	$120	$85	1.4
ECM/Var. Spd. Compressors	14	7	$150	$42	3.6
Cond Fan ECM Motor	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"

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: Reach-In Freezer Energy Savings



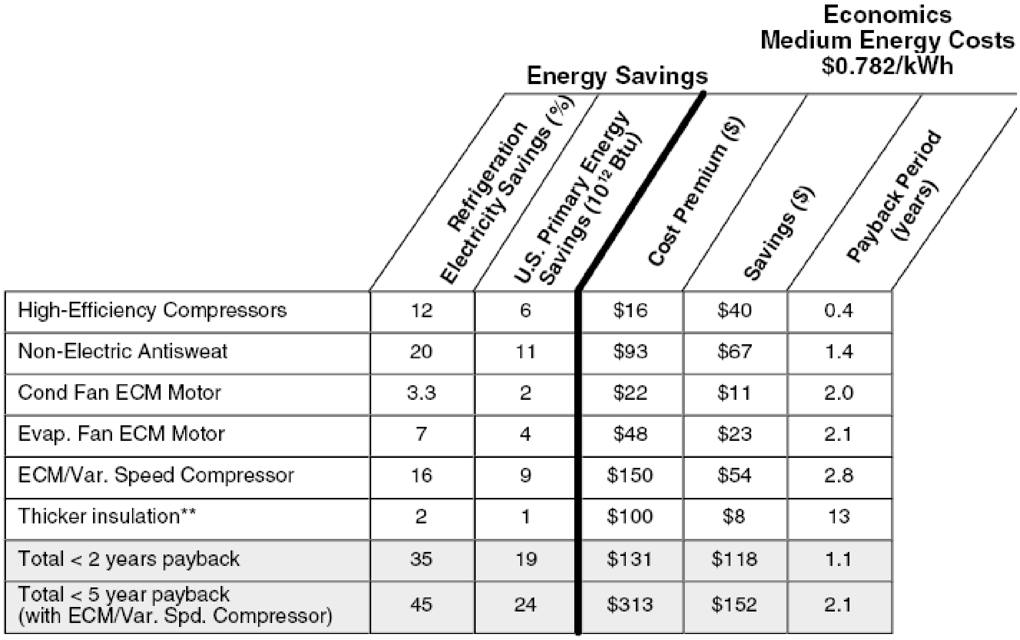
	Energy Savings		Economics Medium Energy Costs $0.782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium ($)	Savings ($)	Payback Period (years)
High-Efficiency Compressors	16	10	$24	$65	0.4
Non-Electric Antisweat	14	9	$67	$58	1.2
ECM/Var. Spd. Compressors	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"

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: Reach-In Refrigerator Energy Savings



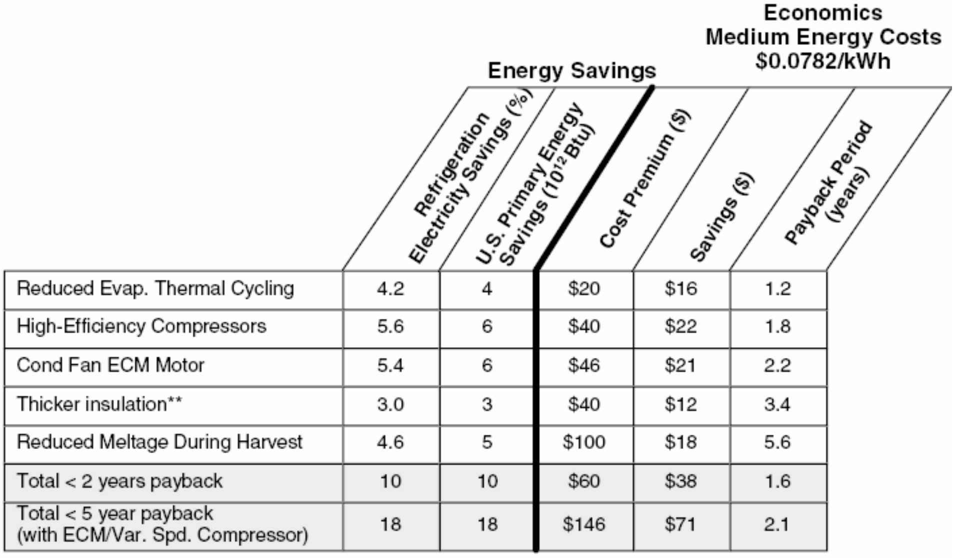
	Energy Savings		Economics Medium Energy Costs $0.782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium ($)	Savings ($)	Payback Period (years)
High-Efficiency Compressors	12	6	$16	$40	0.4
Non-Electric Antisweat	20	11	$93	$67	1.4
Cond Fan ECM Motor	3.3	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”

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 : Ice Machine Energy Savings



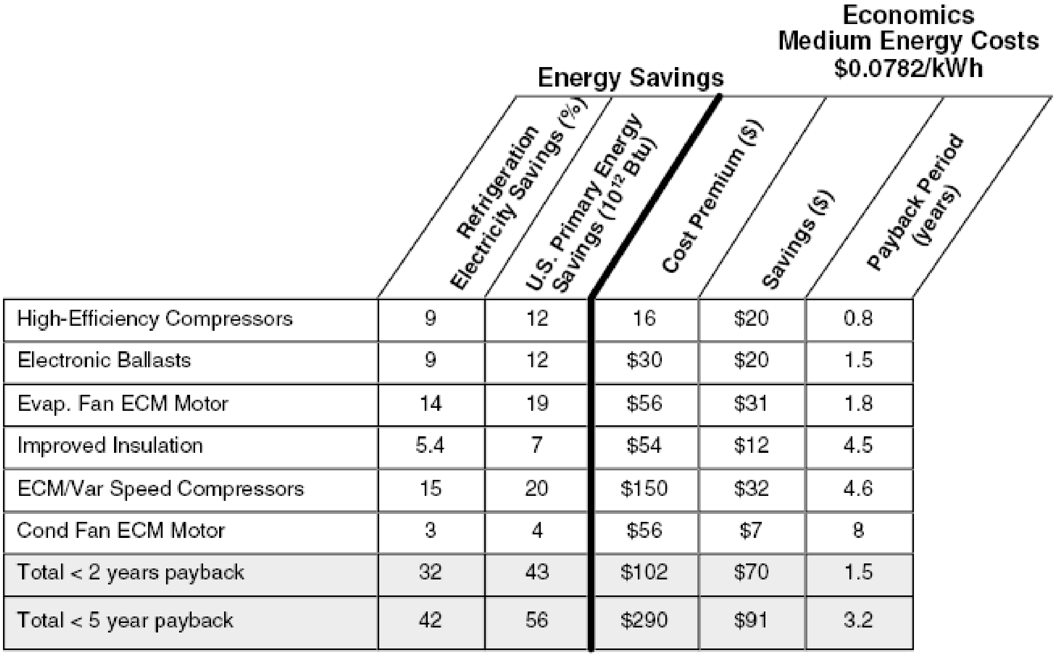
	Energy Savings		Economics Medium Energy Costs $0.0782/kWh		
	Refrigeration Electricity Savings (%)	U.S. Primary Energy Savings (10^{12} Btu)	Cost Premium ($)	Savings ($)	Payback Period (years)
Reduced Evap. Thermal Cycling	4.2	4	$20	$16	1.2
High-Efficiency Compressors	5.6	6	$40	$22	1.8
Cond Fan ECM Motor	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”

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.

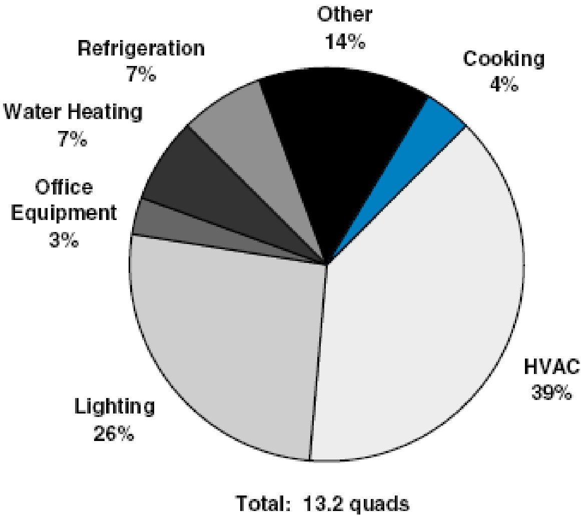
Table : Refrigerated Vending Machine Energy Savings



	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

Introduction

In the commercial sector, energy conservation programs currently put an emphasis on lighting and HVAC equipment, since this equipment accounts 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.



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.

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 few technologies listed below.

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 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

The 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.

Reference:

http://www.eere.energy.gov/buildings/info/documents/pdfs/comm_refridg_equip.pdf