Analysis of Advanced, Low-Charge Refrigeration for Supermarkets

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Present supermarket refrigeration systems require very large refrigerant charges for their operation and can consume as much 1-1.5 million kWh annually. Several new approaches, such as distributed, secondary loop, and advanced self-contained refrigeration systems, are available that utilize significantly less refrigerant and with correspondingly lower refrigerant losses through leakage. New condenser controls have also been developed for multiplex refrigeration systems that allow operation with a refrigerant charge close to the critical level and also allow operation at very low head pressures. Through proper design and implementation, these advanced systems can reduce annual energy consumption by as much as 11.9%. Integration of refrigeration and store HVAC operation is also possible through water-source heat pumps. By incorporating the heat pumps in the heat rejection loop for the refrigeration, the reject heat can be utilized for store space heating without increasing the condensing temperature of the refrigeration. This integrated method was shown to reduce combined operating costs for refrigeration and HVAC by 12.6%.

Background

Supermarkets are the largest users of energy in the commercial sector. A typical supermarket with approximately 40,000 ft^2 of sales area consumes on the order of 2 million kWh annually for total store energy use. Many larger superstores and supercenters also exist that can consume as much as 3-5 million kWh/yr.

One of the largest uses of energy in supermarkets is for refrigeration. Most of the product sold is perishable and must be kept refrigerated during display and for storage. Typical energy consumption for supermarket refrigeration is on the order of half of the store's total. Compressors and condensers account for 30-35% of total store energy consumption. The remainder is consumed by the display and storage cooler fans, display case lighting, and for anti-sweat heaters used to prevent condensate from forming on doors and outside surfaces of display cases.

Figure 1 shows the typical layout of the refrigerated display cases in a supermarket. All refrigerated fixtures in a supermarket employ direct expansion air-refrigerant coils. To reduce noise and control heat rejection, compressors and condensers are kept in a remote machine room located in the back or on the roof of the store. Piping is provided to supply and return refrigerant to the case fixtures.

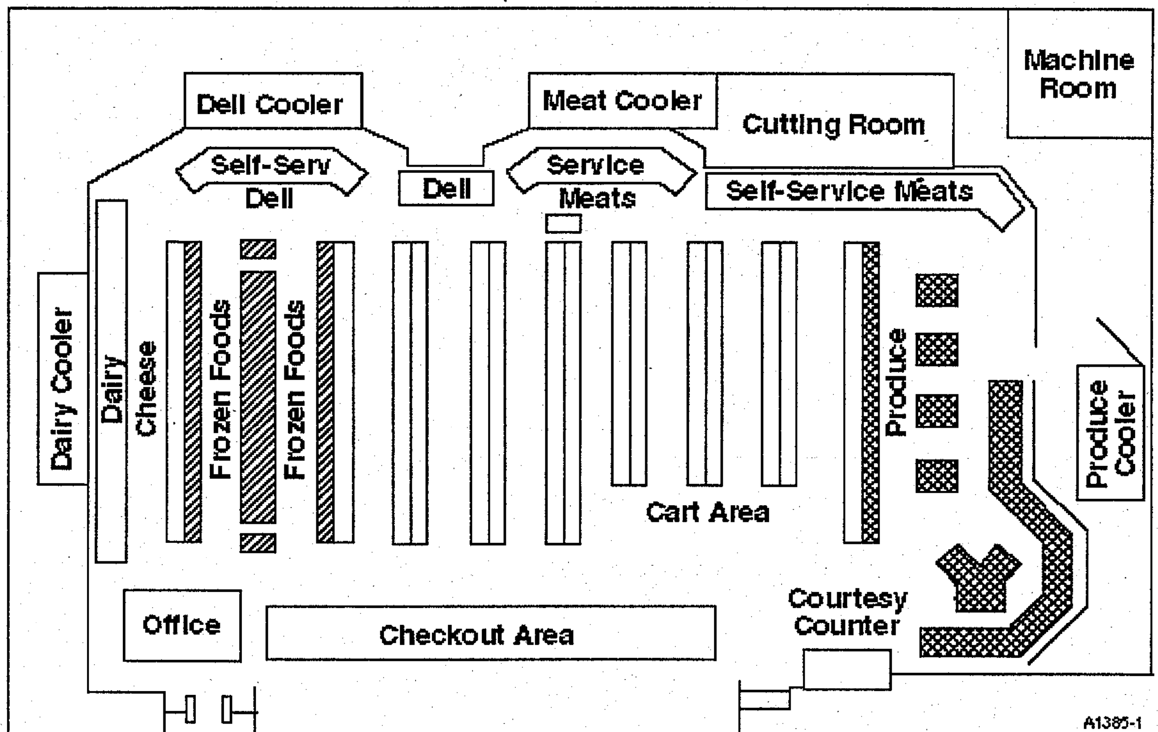

Figure 1 – Layout of a typical supermarket

Figure 2 shows the major elements of a multiplex refrigeration system, which is the most commonly used configuration in supermarkets. Multiple compressors operating at the same saturated suction temperature are mounted on a skid, or rack, and are piped with common suction and discharge refrigeration lines. The use multiple compressors in parallel provides a means of capacity control, since the compressors can be selected and cycled as needed to meet the refrigeration load. An air-cooled condenser is most often employed for heat rejection from the refrigeration system.

As a result of using this layout, the amount of refrigerant needed to charge a supermarket refrigeration system is very large. A typical store will require 3,000- 5,000 lb. of refrigerant. The large amount of piping and pipe joints used in supermarket refrigeration also causes increased leakage, which can amount to a loss of 30-50% of the total charge annually {5}.

With increased concern about the impact of refrigerant leakage on global warming, new supermarket refrigeration system configurations requiring significantly less refrigerant charge are being considered. Examples of low charge refrigeration systems include distributed, secondary loop, and advanced self-contained configurations. Modifications have also been made to multiplex refrigeration systems to reduce the amount of charge needed for their operation. Little is known about the operating or energy consumption characteristics of these low charge systems. Without proper design and operation it is likely that global warming reduction achieved by lowering refrigerant charge and leakage could be negated by secondary global warming caused by increased electrical energy consumption (as measured by the concept of TEWI {1}).

For these reasons, the U.S. Department of Energy initiated an engineering investigation of low charge supermarket refrigeration. The initial work on this investigation involved analysis of distributed and secondary loop refrigeration systems and gave an energy and TEWI comparison with multiplex. The results obtained for this analysis were presented in {2}. Work has continued on this investigation including a field test involving two supermarkets with one equipped with distributed and the other with multiplex refrigeration. Analysis work was expanded to include low-charge multiplex and advanced self-contained systems.

This paper presents all analytical results obtained to this point. Previous analysis results are included for completeness.

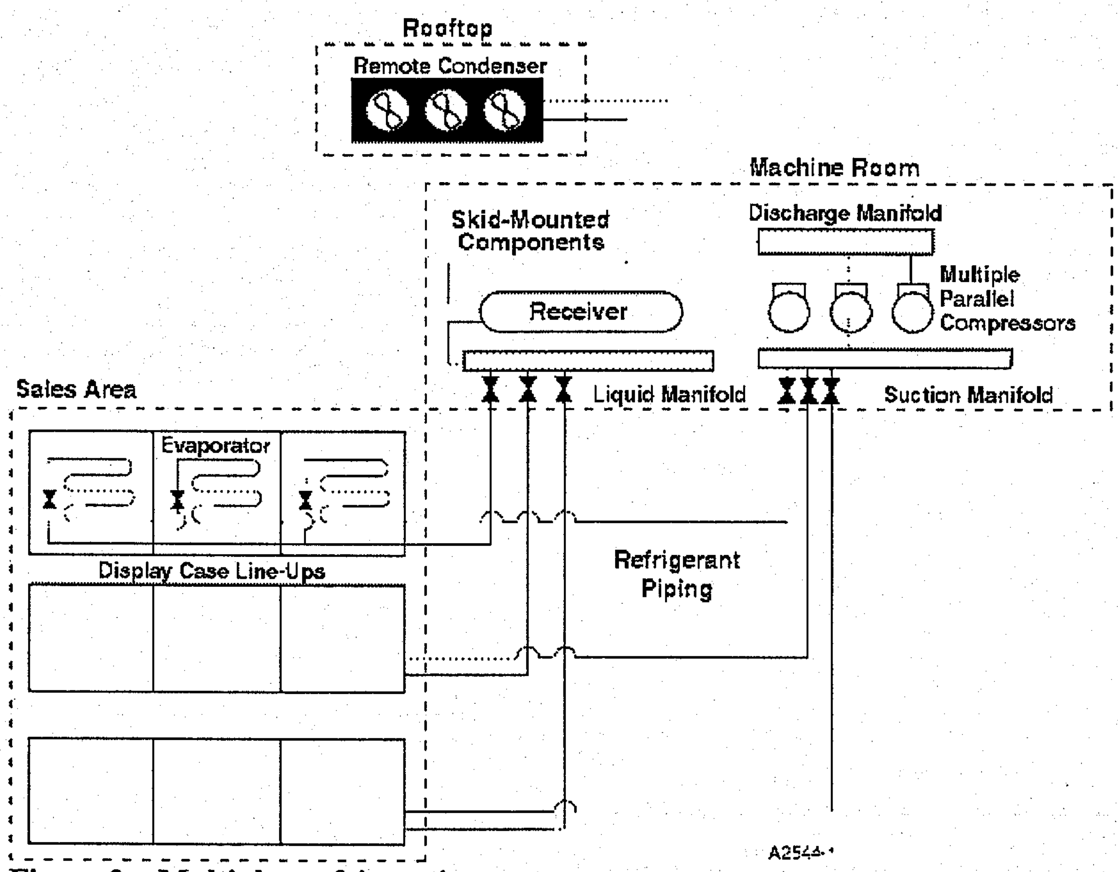

Figure 2 – Multiplex refrigeration system

Distributed Refrigeration

Figure 3 is a diagram showing the major components of a distributed refrigeration system

Multiple compressors are located in cabinets placed on or near the sales floor. The cabinets are close-coupled to the display cases and heat rejection from the cabinets is accomplished through the use of either air-cooled condensers located on the roof above the cabinets or by a glycol loop that connects the cabinets to a fluid cooler.

The distributed refrigeration system employs scroll compressors, because of the very low noise and vibration levels encountered with this type of compressor. These characteristics are necessary if the compressor cabinets are located in or near the sales area. The scroll compressors have no valves, and, in general do not have as high an efficiency as reciprocating units. The no-valve feature of the scroll compressors allows them to operate at a significantly lower condensing temperature. The lowest condensing

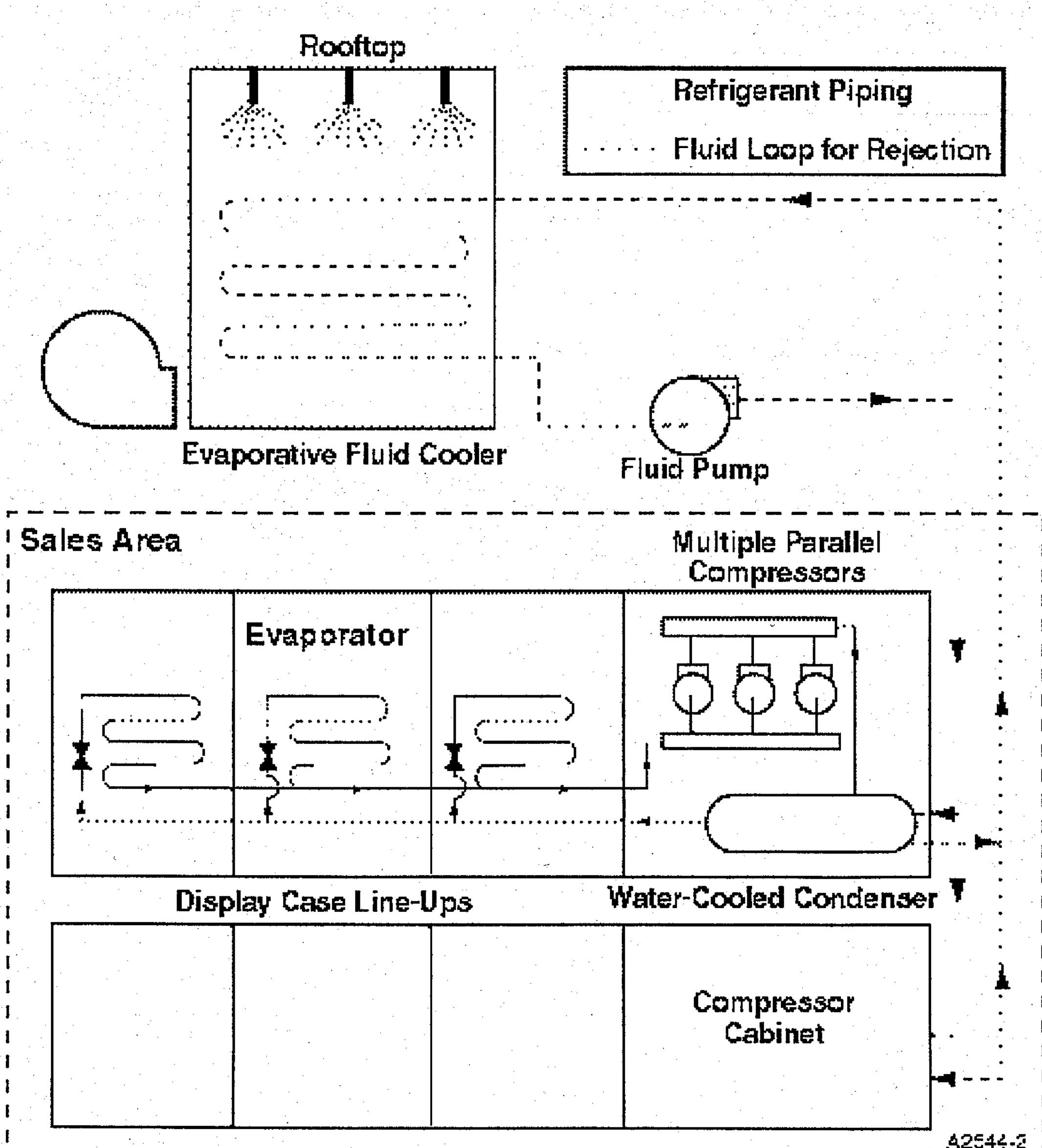

Figure 3 – Distributed refrigeration system

temperature possible occurs at a suction-to-discharge pressure ratio of 2, which, for supermarket systems, means that the lowest condensing temperature possible is on the order of 55 - 60°F for medium temperature refrigeration and 40°F for low temperature refrigeration. The use of the 40°F minimum condensing temperature was not considered here because of the necessity to have 2 glycol loops, which may or may not be practical for actual installations. Minimum condensing temperature was, therefore, limited to 60°F for this assessment.

Scroll compressors also have the potential of providing subcooling through mid-scroll injection of refrigerant vapor. This particular method has not yet been optimized by the compressor manufacturers and was not included as part of this analysis.

The close-coupling of the display cases to the distributed refrigeration cabinets has other ramifications to energy consumption. The shorter suction lines mean that the pressure drop between the case evaporator and the compressor suction manifold is less than that seen with multiplex systems, which means that the saturated suction temperature (SST) of the cabinet will be close to the display case evaporator temperature. The shorter suction lines also mean that less heat gain to the return gas is experienced. The cooler return gas has a higher density and results in higher compressor mass flow rates, which means that less compressor on-time is needed to satisfy the refrigeration load.

The refrigerant charge required for a distributed refrigeration system will be on the order of 900 or 1500 lb. when either water- or air-cooled condensing is employed, respectively. When water-cooled condensers are employed, heat rejection from the water-cooled condensers is done by a glycol loop and a fluid cooler, usually located on the roof of the supermarket. The use of the glycol loop increases the energy consumption of the refrigeration process due to the pump energy needed and higher condensing temperature due to the added temperature rise of the fluid loop. Much of this energy penalty can be negated if an evaporative fluid cooler is employed where heat rejection can take place at close to the ambient wet-bulb temperature.

Secondary Loop Refrigeration

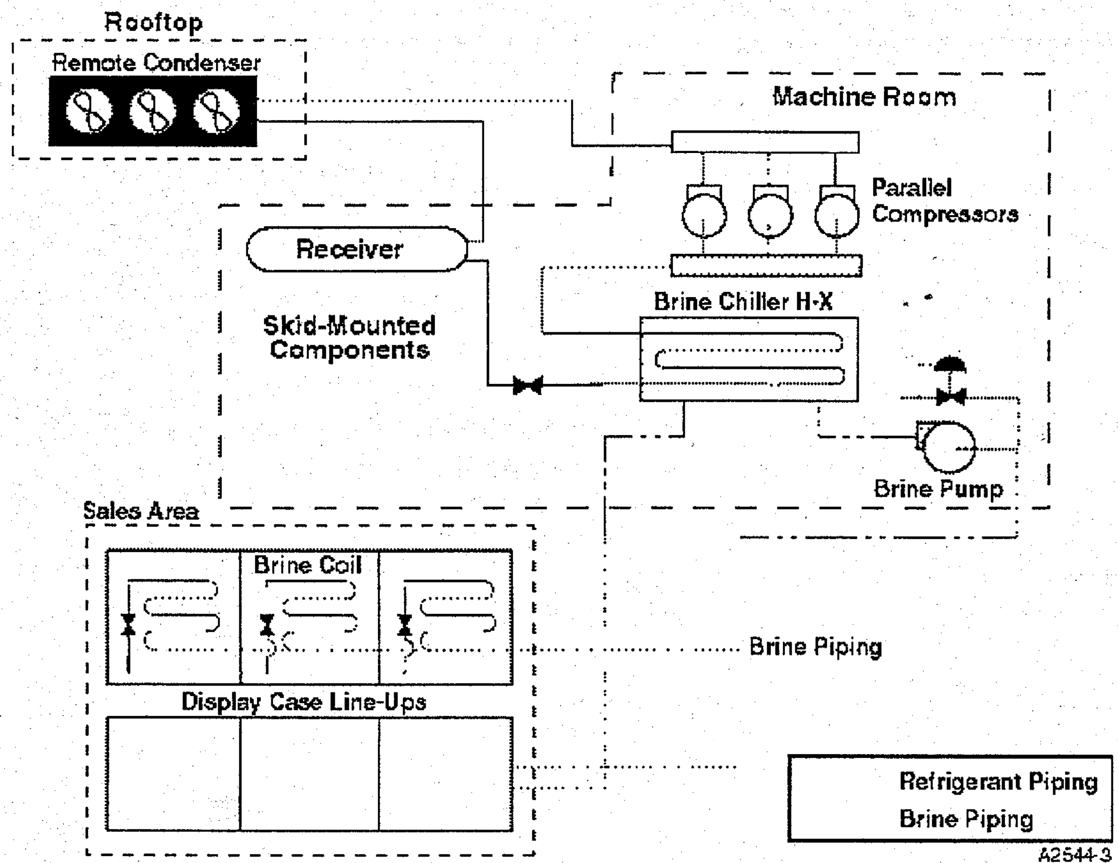

Figure 4 – Secondary loop refrigeration system

Figure 4 shows a piping diagram of a secondary loop refrigeration system.
Brine loops are run between the display cases and central chiller systems. The brine is refrigerated at the chiller and is then circulated through coils in the display cases where it is used to chill the air in the case.

Lowest energy consumption for secondary loop systems is achieved when the display case evaporators are designed specifically for the use of brine, so that the temperature difference between the brine and air is minimized. Brine selection is also of importance, because energy consumption for pumping is a large component of overall energy consumption. The use of brines, such as those employing potassium formate, with high heat capacity and low viscosity at low temperature is desirable.

The number of brine loops employed can also impact energy consumption. Typically, 2 loop temperatures are used, such as –20 and +20°F. If significant portions of the refrigeration load can be addressed by higher temperature loops, energy savings can be obtained. For example, refrigeration loads at 10 or 15°F could be addressed by a loop at 0°F, rather than including this portion of the load with the –20°F. For the analysis given here, 4 loops are considered, operating at temperatures of -20, 10, 20, and 30°F.

Central chiller systems are constructed similarly to the multiplex parallel racks, using multiple parallel compressors for capacity control. The use of high-efficiency compressors, such as reciprocating or scroll units is highly desirable to help offset some of the added energy consumption associated with brine pumping. Because of the location of the evaporator on the chiller skid, the compressors for the secondary loop system are considered close-coupled to the evaporator. The pressure drop and return gas heat gain are minimized in this configuration. Both these factors help to reduce compressor energy consumption. These chiller systems can also be equipped with hot brine defrost where brine is heated by subcooling of the chiller refrigerant.

Heat rejection can be accomplished with air-, water-, or evaporatively cooled condensers. Lowest condensing temperatures are achieved with evaporative condensers, which help reduce energy consumption, particularly when the minimum condensing temperature is set as low as possible. The system refrigerant charge will be on the order of 500-700 lb. with either air-cooled or evaporative condensers and 200 lb. when water-cooled condensers and a fluid loop are used. Like distributed refrigeration, the use of evaporative heat rejection for the fluid loop is recommended to reduce energy consumption.

Low-Charge Multiplex Refrigeration

Several refrigeration system manufacturers now offer control systems for condensers that limit the amount of refrigerant charge needed for the operation of multiplex refrigeration. Figure 5 shows an example of such a control approach. A control valve is used to operate a bypass from the condenser liquid line in order to maintain a constant differential between the high and low pressures of the system. The refrigerant liquid charge is limited to that needed to supply all display case evaporators. No added liquid is needed for the receiver, which is included in the system, primarily for pump-down during servicing. All refrigerant liquid bypassed is expanded and evaporated through heat exchange with the discharge manifold. The resulting vapor is piped to the suction manifold for recompression and return to the condenser. The use of this control approach reduces the charge needed by the refrigeration system by approximately 1/3.

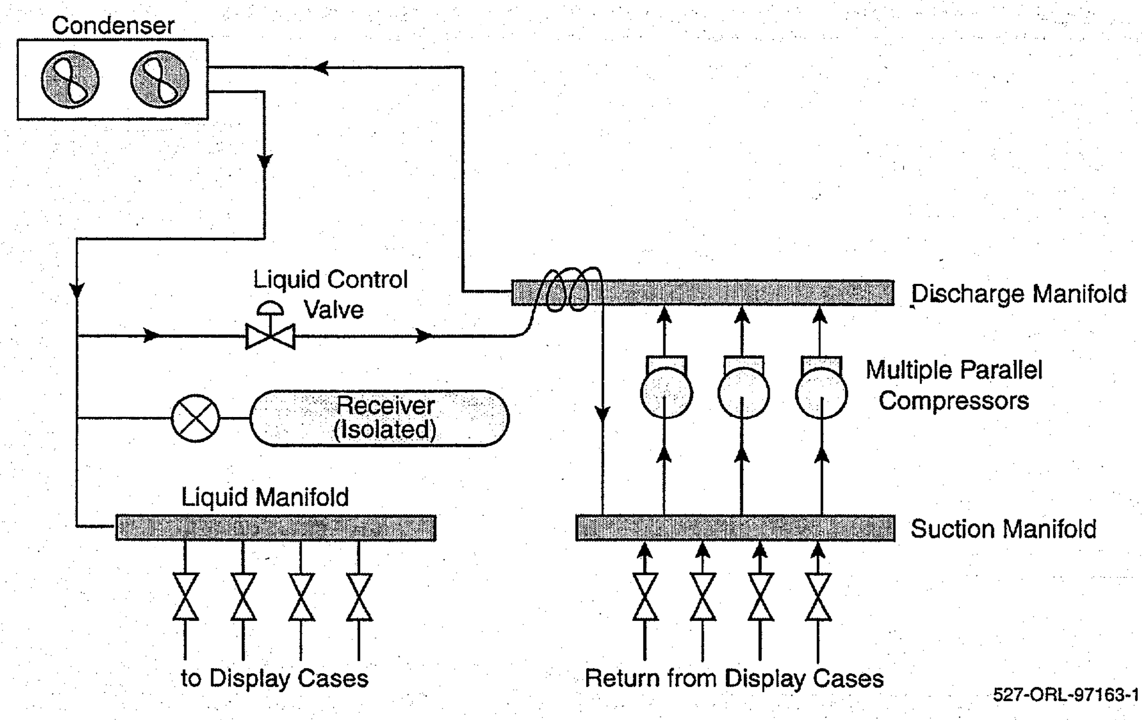

Figure 5 – Piping Diagram for the Low-Charge Multiplex System

The control of the liquid charge by this method offers some energy-saving potential, because it has been found that compressors can be operated at very low head pressures when this control method is employed. The minimum condensing temperature values suggested for this low-charge system are 40 and 60°F for low and medium temperature refrigeration, respectively.

It was found that the fan control strategy is very significant to the energy savings achieved for this particular system, since it is possible for the condenser fans to consume all compressor energy saved in order to maintain the low head pressure. A control strategy, such as variable-speed condenser fans tends to result in the lowest fan energy consumption while achieving the desired low head pressure values.

Modeling of low-charge multiplex refrigeration was performed for both air-cooled and evaporative condensing. Evaporative condensing requires significantly less fan power to achieve the desired low head pressure values {3}.

Advanced Self-contained Refrigeration

An advanced self-contained system is a low refrigerant charge configuration in which the refrigeration compressors and water-cooled condensers are located in the display cases. A glycol loop is used to reject heat from the display cases to the exterior of the store. Several problems have prevented the implementation of this configuration previously. Scroll compressors operate at a low enough noise level to allow their placement in the sales area. Until recently, scroll compressors were available only in a vertical configuration, which was not suitable for placement in display cases. Now, horizontal scroll compressors have been introduced, which could be employed for this purpose.

For energy-efficient operation of a self-contained system, capacity control of the compressor is needed. With a fixed compressor capacity, the condensing temperature of the self-contained system must be maintained within a limited range to insure that the capacity does not greatly exceed the required refrigeration load. Otherwise, excessive compressor cycling will occur, making it difficult to control case temperature. The use of compressor unloading allows the condensing temperature to vary with ambient temperature, since the unloading reduces compressor capacity and helps to match the capacity to the refrigeration load.

Modeling of the scroll compressor included unloading for capacity control in order to maintain the suction pressure set point. The unloading is modeled as a continuous process; and the compressor power is modeled using the standard relationship for power change with compressor unloading. Figure 6 shows the relation between capacity control and compressor power required.

For analysis the minimum condensing temperature was set at 40 and 60°F for low and medium temperature refrigeration, respectively. This may or may not be practical, since 2 glycol loops are needed in order to have 2 different minimum condensing temperature values.

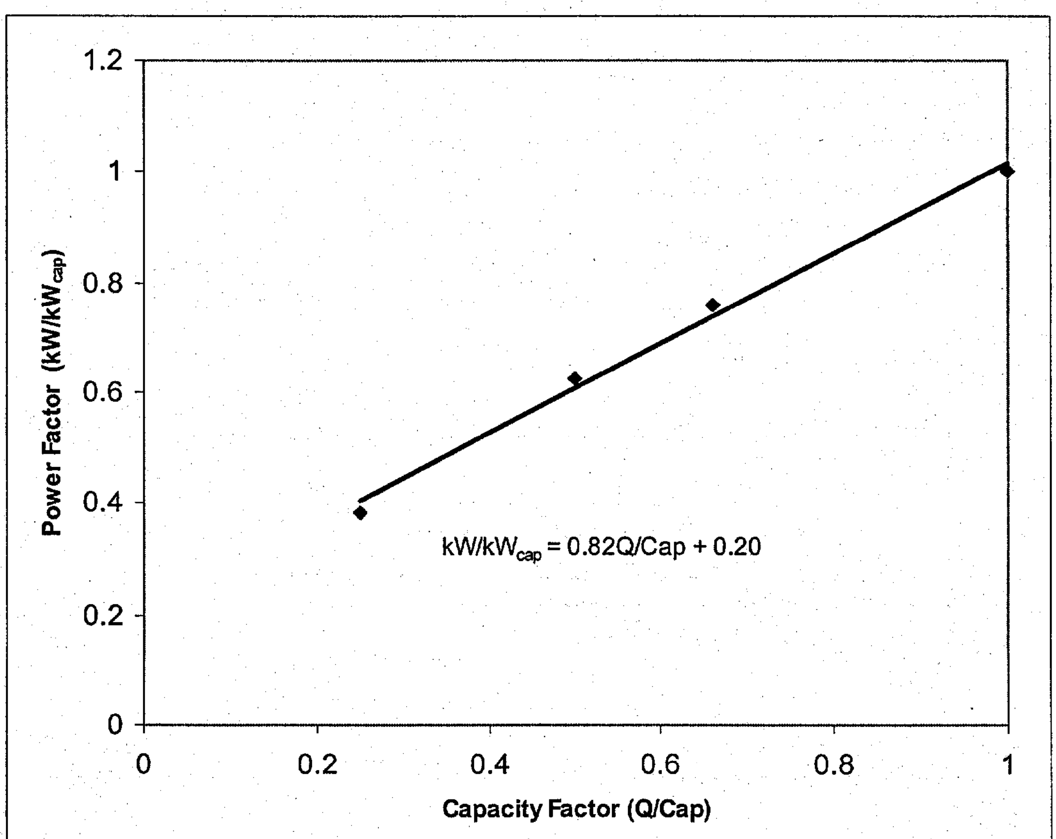

Figure 6 – Relation between Capacity and Power used for modeling unloading scroll compressors.

The close coupling of the compressor to the case evaporator seen in self-contained systems reduces the pressure drop at the compressor suction and also minimizes the heat gain to the suction gas. Both of these effects will result in more efficient operation and were included in the analysis.

Performance Predictions

A model supermarket was simulated and energy consumption estimates were made for present multiplex refrigeration and the advanced, low charge systems. The results of this analysis for a supermarket located in Washington, DC are given in Table 1. Both the distributed system with either air-cooled condensing or water-cooled condensing with evaporative heat rejection, and secondary loop system with evaporative condensing achieved similar results compared to the baseline multiplex system with air-cooled condensing, with annual energy savings on the order of 10.4 –11.9%. The low-charge multiplex refrigeration system showed annual energy savings of 4.3 and 11.6% for air-cooled and evaporative condensing, respectively. No energy savings were estimated for the advanced, self-contained system.

For the three systems modeled with air-cooled condensing, the distributed system showed the lowest energy consumption. The energy savings seen over the baseline multiplex system can be attributed to the close-coupling of the compressors to the display case evaporators and operation of the scroll compressors at lower head pressures than those of the reciprocating compressors employed with the baseline system. The distributed system with air-cooled condensing did not incur energy use for pumping associated with the use of water-cooled condensers and glycol loops. The low-charge multiplex system with air-cooled condensing also showed savings versus the baseline multiplex system. These saving are attributable to operation at lower head pressures.

The analysis results also show the importance of evaporative heat rejection for energy savings. The energy consumption of the multiplex baseline system was reduced by 107,400 kWh, or 11.0% when an evaporative condenser was substituted for the air-cooled condenser. A significant improvement was also seen with the low-charge multiplex system when evaporative condensing was employed. The energy consumption of the secondary loop system was less than that of the baseline multiplex system primarily because of the use of evaporative condensing. Secondary loop system energy savings can also be attributed to the close-coupling of the compressors to the chiller evaporators and savings obtained through subcooling associated with the warm brine defrost employed by this system.

Three systems were modeled with water-cooled condensers and glycol loops for heat rejection. These systems had the lowest refrigerant charge and could be coupled with water-source heat pumps for heat reclaim for space heating. Of these three systems, the distributed system showed the lowest energy consumption.

Table 1
Predicted Energy Consumption for Low-Charge Refrigeration Systems

System	Heat Rejection	Annual Energy (kWh)	Energy Savings vs Multiplex baseline (kWh)	% Savings vs Multiplex baseline
Multiplex (baseline)	Air-Cooled Condenser	976,800	-	-
Multiplex	Evaporative Condenser	869,400	107,400	11.0
Low-Charge Multiplex	Air-Cooled Condenser	935,200	41,600	4.3
Low-Charge Multiplex	Evaporative Condenser	863,600	113,100	11.6
Distributed	Air-Cooled Condenser	860,500	116,300	11.9
Distributed	Water-Cooled Condenser, Evap Rejection	866,100	110,700	11.3
Secondary Loop	Evaporative Condenser	875,200	101,600	10.4
Advanced Self-Contained	Water-Cooled Condenser, Evap Rejection	1,048,300	-	-
Secondary Loop	Water-Cooled Condenser, Evap Rejection	987,900	-	-
Results for supermarket at Washington, DC location				

An environmental assessment of these refrigeration systems was also made through a Total Equivalent Warming Impact (TEWI) analysis for operation of these systems over a 15-year life. Table 2 gives the results of this investigation along with assumptions used for the analysis. The multiplex systems were modeled using 2 refrigerants, which were R-404A for low temperature and R-22 for medium temperature refrigeration. The remaining systems employed only one refrigerant, either R-404A or R-507, for both low and medium temperature refrigeration. Annual leak rates were assigned to each system based on discussions with system manufacturers and supermarket end-users {5}. Lowest refrigerant charge sizes and leak rates were assigned to systems employing water-cooled condensers and glycol loops for heat rejection. The lowest TEWI was achieved by the distributed refrigeration system when water-cooled condensing and evaporative heat rejection was employed. The reduction in CO_2 emission achieved by the operation of this system compared to the baseline multiplex system amounts to approximately 13.7 million kg, or 59.2%. Similar reductions in emissions were also seen for the secondary loop refrigeration system.

Table 2
Total Equivalent Warming Impact (TEWI)
for Supermarket Refrigeration

System	Condensing	Charge (lb)	Refrigerant	Leak (%)	Annual Energy (kWh)	TEWI (million kg of CO_2)		
						Direct	Indirect	Total
Multiplex	Air-Cooled	3,000	R404A/R-22	30	976,800	13.62	9.52	23.15
	Evaporative	3,000		30	869,400	13.62	8.48	22.10
Low-Charge Multiplex	Air-Cooled	2,000	R404A/R-22	15	935,200	4.54	9.12	13.66
	Evaporative	2,000		15	863,600	4.54	8.42	12.96
Distributed	Air-Cooled	1,500	R404A	10	860,500	3.33	8.38	11.71
Distributed	Water-Cooled, Evap	900	R404A	5	866,100	1.00	8.44	9.44
Secondary Loop	Evaporative	500	R507	10	875,200	1.13	8.54	9.67
Secondary Loop	Water-Cooled, Evap	200	R507	5	987,900	0.23	9.63	9.86
Advanced Self-Contained	Water-Cooled, Evap	100	R404A	1	1,048,300	0.02	10.22	10.24

Results for site in Washington , DC – 15 year service life
Conversion factor = 0.65 kg CO_2/kWh
Multiplex - 33.3% R404A (low temp.), GWP = 3260; 66.7% R22 (medium temp.), GWP = 1700
Distributed and Adv. Self-Contained - 100% R404A, GWP = 3260
Secondary Loop – 100% R507, GWP=3300

Supermarket HVAC

HVAC also represents a large portion of the energy use of a supermarket, on the order of 10 to 20% of the store total, depending upon geographic location. The large amount of refrigerated fixtures installed in a supermarket has a major impact on the design and operation of the store HVAC system. For space cooling, the refrigeration removes both sensible and latent heat from the store, such that the sensible-to-latent load ratio is much smaller than is seen in most commercial buildings. Because of the installed refrigeration, space heating is the dominant HVAC load. Rejection heat from the refrigeration system is also available for reclaim and can be used for both water and space heating in the store.

The use of water-source heat pumps represents a better way to utilize refrigeration reject heat for space heating. The heat pumps can be installed in the glycol/water loop for refrigeration heat rejection, and use the refrigeration heat to provide space heating. This method offers several advantages, such as reclamation of a much larger portion of the reject heat, and the condensing temperature and head pressure of the refrigeration system does not have to be elevated for the heat pumps to use the reject heat. Refrigeration system energy savings achieved by low head pressure operation can be realized along with the energy benefits seen through heat reclaim.

An analysis was performed for a supermarket HVAC system where conventional roof top units, refrigeration heat reclaim, and water-source heat pumps were examined and compared. The results of the HVAC analysis are shown in Table 3 for a supermarket located in Washington, DC. Since both gas and electric energy are used, the comparison of systems is given in terms of annual operating cost for both refrigeration and HVAC. Local utility rates for gas and electricity were used for these cost calculations. The lowest operating cost was achieved by the combination of distributed refrigeration and water-source heat pumps, which saved $12,997, or 12.6%, when compared to the baseline multiplex refrigeration system with conventional rooftop HVAC units.

Table 3 – Operating Cost Results for Combined Refrigeration and HVAC Systems

System		Operating Costs($)				
Refrigeration	HVAC	Electric	Gas	Water	Total	Savings
Multiplex (baseline)	Convent.	85,565	17,653		103,218	
Multiplex	Reclaim	90,519	8,967		99,486	3,732
Distributed –Air cooled	Convent.	77,218	17,653		94,871	8,346
Distributed – WC, Evap	WSHP	88,091	0	2,130	90,221	12,997
Sec Loop – Evap Cond	Convent.	77,804	17,653	1,784	97,241	5,977
Sec Loop - WC, Evap	WSHP	95,258	0	2,130	97,388	5,830

Results for Washington, DC Site

Conclusions

The results of this analysis showed that the greatest refrigeration energy savings for a supermarket application were achieved by: a distributed refrigeration system with air-cooled condensing; a low-charge multiplex system with evaporative condensing; and secondary loop also with evaporative condensing. An advanced self-contained system approach showed higher energy consumption than the baseline multiplex system. The power penalty associated with compressor unloading along with energy required for glycol loop pumping contributed to this increase.

The supermarket refrigeration systems showing lowest TEWI were the distributed compressor and secondary loop systems employing the glycol loop and evaporative heat rejection. The lowest direct warming impact was demonstrated by the advanced self-contained system, but these reductions were offset by the higher indirect impact due to increased energy use.

The use of water-source heat pumps with refrigeration systems employing glycol loops for heat rejection was found to produce operating cost savings due to combined savings for refrigeration and HVAC. Distributed and secondary loop systems in this configuration showed significantly higher savings than those obtained by multiplex refrigeration with heat reclaim.

Further energy savings could be realized by refrigeration systems employing scroll compressors if mid-scroll injection for subcooling were employed. Unfortunately, the savings that can be obtained could not be quantified here due to a lack of available design or operating data for compressors of this type.

Because of the energy and cost savings potential that low charge refrigeration systems have, the US DOE has extended the efforts in this investigation to include field testing of a distributed refrigeration system employing a glycol loop and WSHP for HVAC. This particular system is now installed in a supermarket operating in the suburbs of Worcester, MA. This store was instrumented to gather energy and operating data for the refrigeration and heat pump systems. At the same time, a second store in close proximity to the distributed store was also instrumented. The second store employs a state-of-the-art multiplex refrigeration system and conventional, rooftop HVAC. Both sites are now being monitored.

In addition, a second field test had been started in the Los Angeles, CA area by Southern California Edison for the California Energy Commission (CEC) {4} that will involve the design and field testing of a secondary loop refrigeration system. Test results obtained from this field demonstration will compliment those obtained in the DOE low refrigerant charge system field testing.

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