Field Testing of an Advanced Low-Charge Supermarket Refrigeration System

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Abstract

A field test was conducted to compare the performance of multiplex and distributed supermarket refrigeration systems. Two supermarkets in the vicinity of Worcester, Massachusetts were the sites utilized for the field test. One store was equipped with a multiplex refrigeration system that had 3 compressor racks and air-cooled condensers. At the second store, a low-refrigerant-charge distributed refrigeration system was installed that consisted of 10 compressor cabinets. Heat rejection for the compressor cabinets was accomplished through water-cooled condensers piped to a fluid loop that used dry fluid coolers. The second store also had water-source heat pumps for space heating and cooling that were piped into the fluid loops. Both sites were instrumented to determine energy consumption, refrigeration and heating, ventilating and air conditioning (HVAC) loads, and numerous system state points to characterize operation. Data were collected for approximately 18 months. A comparison of the performance of the two refrigeration systems was made based on the information gathered. The results showed that the multiplex refrigeration system used less energy and showed better EER than the distributed system. A TEWI comparison between the two refrigeration systems showed significant reductions in atmospheric CO_2 generation by the distributed system despite added energy use. Test results also showed that the water-source heat pumps produced energy cost savings for store space heating by recovery of the refrigeration reject heat.

1. Introduction/Background

Present supermarket systems employ direct expansion of refrigerant in the display cases and walk-in coolers with remotely operated multiplexed compressor racks and condensers. Because of the large floor area of modern supermarkets, the amount of refrigerant needed for operation is very large, on the order of 3,000 – 5,000 lb. With growing concern over the environmental impact of refrigerant leakage from these systems, several new system configurations have been designed and constructed that require less refrigerant for their operation. Examples of such systems include:

- Distributed – Display cases and walk-in's are piped to compressors located in cabinets on the sales floor or around the perimeter of the store. Heat rejection may be done through the use of water-cooled condensers and a fluid loop or with air-cooled condensers on the store roof above each compressor cabinet.
- Secondary loop – Refrigeration to the display cases and walk-in's is provided by a secondary fluid loop that is refrigerated by a central chiller system. Heat rejection is done through air-cooled or evaporative condensers, or a fluid loop can be employed to minimize the refrigerant charge.
- Advanced self-contained – each display case or group of several cases has a condensing unit installed in the case. Heat rejection is done through the use of a fluid loop.

While each of these systems uses significantly less refrigerant than standard multiplex compressor systems, the energy consumption of these systems is also very important in determining their impact on the environment. A complete assessment of the environmental impact of a refrigeration system can be made through a TEWI (Total Equivalent Warming Impact) evaluation that takes into account both primary and secondary global warming caused by direct refrigerant leakage and electric generation, respectively [1].

The energy consumption of low-charge refrigeration systems can be greater than that of multiplex systems because of the methods used to limit their refrigerant charges. For distributed refrigeration, low noise-level scroll compressors must be employed in order to locate the compressor cabinets in the sales area. Scroll compressors generally have lower EER values than reciprocating compressors today. If a fluid loop is used for heat rejection, this will add an extra temperature difference that will increase the operating condensing temperature, and it will also result in adding pump energy consumption for fluid circulation in the loop. For the secondary loop system, the pumping energy associated with the secondary loops adds significantly to the total refrigeration energy.

System design methods and enhancements exist that help offset the added energy associated with low-charge operation. Both distributed and secondary loop systems can be close-coupled to their evaporator loads which helps maintain higher suction pressure and tends to lower the return gas temperature. Scroll compressors can be operated at very low head pressures, because they have no suction or discharge valves. Refrigerant subcooling can be employed by scroll compressors through mid-scroll vapor injection. For the secondary loop system, refrigerant subcooling can be obtained through the use of hot brine defrost where the brine used for defrost is heated by subcooling the liquid refrigerant. The use of evaporative heat rejection helps to lower the condenser temperature penalty seen with fluid loops.

An analytical investigation of low-charge refrigeration systems showed that with proper design and operation both distributed and secondary loop refrigeration systems could use less energy than a baseline air-cooled multiplex system. [2,3]. The estimated energy savings ranged from 6.3 - 12.4 % for distributed and 3.7 - 10.2 % for secondary loop refrigeration, respectively. A TEWI analysis of the distributed and secondary loop systems showed significant reductions (up to 60%) in CO_2 generation versus the baseline air-cooled multiplex refrigeration system.

Another energy saving method investigated was the use of water-source heat pumps in conjunction with fluid loops for heat rejection. In this approach, the heat of rejection from the refrigeration system can be reclaimed by the water-source heat pumps to provide space heating for the store. The advantage of this method over standard refrigeration heat reclaim is that the condensing temperature of the refrigeration system can be floated to minimum levels allowable without limiting heat recovery by the heat pump. A larger fraction of the reject heat can be utilized with the water-source heat pumps than can be recovered with conventional heat reclaim.

In order to verify these analysis results, it was decided that a field test of a low-charge refrigeration system coupled with water-source heat pumps for HVAC should be undertaken. Through the cooperation and assistance of Price Chopper Supermarkets (Golub Corporation, Schenectady, NY) and Massachusetts Electric, such a field test was conducted involving two Price Chopper supermarkets in the Worcester, MA area. The test supermarket located in Marlborough, MA. was equipped with multiplex compressor racks with air-cooled condensers. The store consisted of 52,000 ft^2 of floor space and was a new construction site that opened in 1997. The second test supermarket was located in Webster, MA and also had approximately 52,000 ft^2 of floor space. During renovation and remodeling of the Webster store, the distributed refrigeration system was installed along with 3 water-source heat pumps for store HVAC. Both stores were thoroughly instrumented and were monitored for approximately 2 years. During that time period energy and performance data were gathered for the refrigeration and HVAC systems at both sites.

2. Description of the Test Refrigeration Systems

Table 1 provides a listing of the refrigerated fixtures that were connected to the multiplex refrigeration system. Three racks are employed with each rack having two suction groups. Rack 1 is the low temperature refrigeration system that employs R-404A as the refrigerant. Racks 2 and 3 are used for medium temperature refrigeration and both racks use R-22 as the refrigerant. The high temperature suction group of Rack 3 supplies mechanical subcooling for both Racks 1 and 2. Mechanical subcooling is normally used on low temperature refrigeration only, making the subcooling for the medium temperature Rack 2 somewhat unique. Separate air-cooled condensers were used for each compressor rack. The low temperature condenser was sized with a design temperature difference of 10^oF while the medium temperature condensers were sized to operate with a 15^oF temperature difference. During low ambient temperature operation, fan cycling was used to maintain condensing temperature at approximately 70^oF. This same condenser temperature set point was employed with all 3 rack systems. The design refrigeration loads were 422,825 and 899,953 Btu/h for the low and medium temperature refrigeration, respectively. The design subcooling load for Racks 1 and 2 was 361,670 Btu/h.

The details of the distributed refrigeration system are given in Table 2. The refrigeration load is divided among 10 compressor cabinets (A – J). Cabinets A, B, C, and D are dedicated to low temperature refrigeration, while Cabinets E, G, and I are medium temperature refrigeration. Cabinets F, H, and J each have two suction groups, one for low temperature refrigeration and the other for medium temperature. The low temperature cabinets A, B, C, and, D were equipped with subcooling by mid-scroll vapor injection. The design refrigeration loads for the distributed system were 404,845 and 1,010,936 Btu/h for low and medium temperature refrigeration, respectively.

Heat rejection for the distributed refrigeration system was accomplished by fluid-cooled condensers located at each compressor cabinet. Two fluid loops using a propylene glycol/water solution were connected to these condensers and heat was rejected by fluid coolers located on the roof of the supermarket. The fluid coolers were dry units, rather than evaporative. The choice of dry coolers was made by Price Chopper, because the store is located a considerable distance from their headquarters and the added maintenance of evaporative units could not be addressed by their in-house maintenance people. Price Chopper felt that improperly maintained evaporative fluid coolers could result in refrigeration system reliability issues for the store.

The use of dry fluid coolers was detrimental to the performance of the distributed system for several reasons. The added temperature difference associated with the operation of the fluid loops for heat rejection could not be offset by the dry coolers, since the coolers reject heat to the ambient dry-bulb temperature. This factor was particularly evident during summer operation when the condensing temperatures of the distributed cabinets were considerably higher than those of the multiplex racks. The fan power needed for the dry fluid coolers was also considerably larger than that associated with the multiplex air-cooled condensers, primarily because of the inclusion of the water-source heat pumps in the fluid loops. The fluid coolers had to be sized to provide heat rejection for the refrigeration and the water-source heat pumps when operating in space cooling mode.

Water-source heat pump operation also impacted the sizing of the fluid loop pumps since added fluid flow was needed for space cooling heat rejection. Each fluid loop had a circulating pump. The power of each pump was measured at 6.5 kW. Pumping power for these loops was higher than expected because the loops were filled with a 50/50 mixture of propylene glycol and water for freeze protection. The concentration of glycol was higher than needed for this particular location, but the concentration was chosen to insure reliable operation of the fluid loops during winter.

Three water-source heat pumps were installed at the distributed system test store. The heat pumps were integrated into the fluid loops so that the refrigeration heat of rejection could be used by the heat pumps for store space heating and the fluid loops could be used for heat rejection from the heat pumps during store space cooling. Table 3 provides descriptions of the water-source heat pump units that were installed. The 2 large heat pumps were constructed to utilize a dual-path approach for space cooling operation. The heat pump was equipped with 2 sets of compressors and evaporator coils; one set of compressors and a coil were used for cooling return air from the store, while the other set of compressors and coil provided cooling and dehumidification to outside ventilation air. The cooling capacity of the heat pump was divided such that the outside air system provided approximately 25 tons of cooling while the return air system provided approximately 20 tons. In space heating mode, only the return air system was used. The third heat pump was considerably smaller than the 2 dual-path units and was operated as a single-path system where outside air was mixed with return air prior to passing through the heat pump coil.

Operating data gathered from the heat pumps showed that they were very effective in recovering the rejected heat from the refrigeration system for store space heating. The dual-path approach for space cooling was found not to be appropriate for this particular site for space cooling. The dehumidification requirements for the site were much less than the dual-path system was designed for, and it was found that both the return and outside air systems had to be run in order to meet the total space cooling load of the supermarket. The outside air system required considerably more energy for cooling because of the higher temperature and humidity of the outside air. Energy consumption for space cooling was found to be very high for this particular site. A conventional single-path arrangement for all 3 heat pumps would have been more appropriate.

Space heating operation of the water-source heat pumps also affected the minimum operating temperature of the fluid loops, which, in turn, impacted the minimum condensing temperature that could be achieved by the distributed refrigeration system. Originally, it was decided that the minimum condensing temperature for the distributed system should be set at 50^{o}F, which meant that the fluid loops had to be operated at a minimum temperature of 40^{o}F. During the first winter of operation it was found that the heat pumps required a fluid temperature of 50^{o}F in order to meet the largest space heating load at lowest ambient temperature. At the largest space heating load, the heat pumps needed both heat recovery from the fluid loop and auxiliary heating in order to meet this load. For this reason, the fluid loops were maintained at a minimum temperature of 50°F during winter operation, which resulted in a minimum condensing temperature for the distributed refrigeration system on the order of 60°F.

Table 3 - Operating Characteristics of the Water-Source Heat Pumps

Heat Pumps WSHP-1 and WSHP-2 Space Cooling			
Entering Air Temp (°F) Entering Water Temp(°F) - 100 Dry- Bulb 73 Wet-Bulb 60			
	Cooling Capacity (Btu/h)	Compressor Power (kW)	Water Flow (gpm)
Outside Air	293,173	26.4	68
Return Air	252,202	19.4	66
Space Heating Entering Air Temp (°F) - 70 Entering Water Temp (°F) - 70			
	Heat Absorbed (Btu/h)	Compressor Power (kW)	Water Flow (gpm)
Return Air	278,672	17.2	66
Heat Pump WSHP-3 Space Cooling			
Entering Air Temp (°F) Entering Water Temp(°F) - 100 Dry- Bulb 73 Wet-Bulb 60			
	Cooling Capacity (Btu/h)	Compressor Power (kW)	Water Flow (gpm)
	65,634	6.5	20
Space Heating Entering Air Temp (°F) - 70 Entering Water Temp (°F) - 70			
	Heat Absorbed (Btu/h)	Compressor Power (kW)	Water Flow (gpm)
	87,282	6.3	20

Both sites were equipped with similar energy management systems (EMS) that were used for control of the refrigeration. The display cases and walk-in coolers at both sites employed case controllers for control of evaporator refrigerant flow and discharge air temperature. The case controllers employ 4 temperature readings in their control algorithms that are located at the refrigerant inlet and outlet of the coil and at the air discharge and return. Refrigerant flow was regulated at the evaporator by either an electronic expansion valve or by a combination of solenoid valve and electronic suction pressure regulator.

2.1 Test Instrumentation and Data Collection

Both the multiplex and distributed refrigeration systems were thoroughly instrumented in order to assess and compare their operating performances. Table 4 lists the instrumentation associated with the multiplex refrigeration system. Each rack and each condenser was equipped with a watt transducer for power measurement; the watt transducer readings were used for all energy consumption comparisons. Data were collected at 5-minute intervals and stored in the EMS for later retrieval.

The refrigeration load of each suction group was calculated based upon the suction and discharge pressures measured at the rack, the compressor on-off digital signals, and through the capacity curves for each compressor as specified by the manufacturer [reference Copeland Compressor data]. The procedure to determine the refrigeration load consisted of first finding the rated capacity of each compressor at the measured saturated suction and discharge temperatures (determined from the suction and discharge pressure readings). The rated capacities were then adjusted to take into account the actual return gas and liquid refrigerant temperatures. The refrigeration load for the rack suction group was then determined from

$$Q_{sg} = \sum RF_i Cap_i$$

where

Q_{sg} is the refrigeration load of the rack suction group (Btu/h)

RF_i is the on/off status of the compressor (on=1, off=0)

Cap_i is the refrigeration capacity of the compressor for the measured operating conditions

The load calculation was performed using the 5-minute data. The results were later averaged to determine the hourly and daily refrigeration loads.

The refrigerant liquid subcooling loads for racks 1 and 2 were found by calculating a refrigerant mass flow rate from the load calculations and the enthalpy difference between the liquid and suction manifolds of each rack. The amount of subcooling could then be calculated using the liquid temperatures at the condenser outlet and the liquid manifold. The compressor power for subcooling was included in the power measurement of Rack 3. The power used for subcooling was estimated from the compressor performance curves based upon the subcooling load and the state points of the compressors.

Table 4 – Multiplex Refrigeration Instrumentation (Each Compressor Rack)

Rack Suction Pressure (one per suction group)
Rack Discharge Pressure
Return Gas Temperature (one per suction group)
Liquid Temperature at Condenser Outlet
Liquid Manifold Temperature (after subcooler)
Rack Power (kW)
Compressor On/Off Digital (one per compressor)
Condenser Fan Power

Table 5 describes the instrumentation installed on the distributed refrigeration system for the field test. Watt transducers were installed on all compressor cabinets and on the fluid coolers. Refrigeration state point measurements consisted of suction and discharge pressures and return gas temperature. For cabinets with two suction groups separate measurements were made for each suction pressure and return gas temperature

Table 5 – Distributed Refrigeration Instrumentation (Each Compressor Cabinet)

Cabinet Suction Pressure (one per suction group)
Cabinet Discharge Pressure
Return Gas Temperature (one per suction group)
Liquid Manifold Temperature
Cabinet Power (kW)
Fluid Cooler Fan Power (one per fluid cooler)

The large number of compressors made it too costly to install digital readings on each compressor. The refrigeration load for each cabinet was estimated based upon the rated capacity of compressors at the measured operating conditions. The rated power for each compressor was also estimated and the refrigeration load was adjusted by the ratio of the measured to the rated compressor power. The refrigeration loads of the compressor cabinets estimated using the instrument reading measured at 5-minute intervals. The refrigeration loads were calculated for each 5-minute interval and later averaging of these load estimates determined the hourly and daily refrigeration load values.

Table 6 lists the instrumentation used for the evaluation of the water-source heat pumps. The operating mode of the heat pumps (heating or cooling) was determined by monitoring the heat pump power, the return air coil temperatures, and the water outlet temperature. Space heating or cooling was indicated by the coil temperature, with a temperature above 150°F indicating heating and a temperature below 50°F indicating cooling. Compressor cycling could also be detected by a change in coil temperature accompanied by a change in total heat pump power. Heat recovered from the fluid loop was determined by the change in fluid temperature and assuming a constant flow rate through the heat exchanger of 66 gpm.

Table 6 – Instrumentation Associated with the Water-Source Heat Pumps

Inlet Fluid Temperature to Water-Refrigerant HX
Outlet Fluid Temperature leaving Heat Pump Water-Refrigerant HX
Outlet Fluid Temperature leaving Water-Refrigerant HX for Outside Air Cooling
Refrigerant-Air Coil Temperature - Return Air Compressor 1
Refrigerant - Air Coil Temperature - Return Air Compressor 2
Return Air Temperature
Supply Air Temperature
Heat Pump Power

Outside ambient temperature and relative humidity readings were installed at both sites. After several months, it became apparent that the ambient temperature reading at the distributed refrigeration site was not reading correctly. The ambient temperature probe was located on the supermarket roof and the readings were strongly influenced by the sun. Also, the relative humidity readings at both sites stopped functioning after approximately 6 months of operation. It was decided that a better approach for ambient data was to use hourly temperature and humidity readings that were obtained from NOAA for the Worcester Airport. Comparison of the airport temperature readings with readings from the multiplex refrigeration site showed good agreement and the airport relative humidity measurements were considered to be more representative. The NOAA ambient readings were used for analysis of refrigeration performance data for both sites. Both stores were equipped with inside dry-bulb temperature and relative humidity measurements that were used in conjunction with the EMS for HVAC control.

All display cases and walk-in coolers at both sites were equipped with discharge air temperature sensors that were used by the EMS for refrigeration and fixture temperature control. Discharge air temperature readings for all cases and walk-in coolers were collected regularly as part of the field test data.

All instrumentation at both sites was connected to the store EMS and readings were taken at 5-minute intervals. The EMS at both sites had adequate data storage capability so that they could be used as data acquisition systems for the field testing.

2.2 Discussion of Field Test Results

Data collection was begun at both test stores in November 1999 and is continuing through the present. Test results for the first year of operation and comparison showed that the water-cooled condensers of the distributed system were under-sized, particularly for the low temperature compressor cabinets. The water-cooled condensers on all compressor cabinets were replaced with larger heat exchangers prior to May 2001. The results shown below are based upon data collected after the condenser replacement.

Comparison between the two systems was done on the basis of low temperature (LT) and medium temperature (MT) refrigeration. For the multiplex system, the low temperature refrigeration data consisted of measurements taken from the low temperature compressor rack. For the distributed system, data for the low temperature refrigeration consisted of measurements taken on 4 compressor cabinets (Cabinets A-D) and on the 3 low-temperature suction groups associated with Cabinets F, J, and H. The medium temperature data for the multiplex system were taken from the remaining 2 compressor racks. For the medium temperature refrigeration load of the multiplex system, the refrigeration provided for mechanical subcooling was estimated and removed, since this cooling was not used directly by the display cases or walk-in coolers. The compressor power associated with mechanical subcooling was included with the total power for the medium temperature refrigeration. The medium temperature refrigeration data for the distributed system consisted of measurements taken for the remaining compressor cabinets and the medium temperature suctions groups of the 3 split cabinets. The power and energy data for heat rejection of either system were combined for both medium and low temperature refrigeration, because each fluid loops of the distributed system service both the low and medium temperature compressor cabinets.

Table 7 and Figure 1 show the energy consumption of the two refrigeration systems for two time periods, the first consisting of May through August 2001, and second from November 2001 through February 2002. Data for the months of September and October 2001 were not included, because problems incurred with the EMS at both sites during that period did not allow adequate data to be collected for representative comparison.

Table 7 - Energy Consumption Comparison between Multiplex and Distributed Refrigeration

Energy Consumption (kWh/day)				
	Distributed	Multiplex	Difference	% Difference
May-Aug				
LT Compressor	1306.4	1290.2	16.2	1.2
MT Compressor	1594	1201.2	392.8	24.6
Heat Reject	702.7	608.4	94.3	13.4
Total	3603.2	3100.5	502.7	14.0
Nov - Feb				
LT Compressor	863.2	957.9	-94.7	-11.0
MT Compressor	951.3	635.9	315.4	33.2
Heat Reject	316.1	364.4	-48.3	-15.3
Total	2130.5	1958.2	172.3	8.1

The energy data show that the multiplex compressors consumed less energy for both low and medium temperature refrigeration during summer operation. For the winter, the distributed system showed lower energy consumption for low temperature refrigeration, and higher energy consumption for medium temperature refrigeration. For heat rejection, the multiplex system had lower energy consumption during summer and higher during winter. This finding is significant in that the energy for heat rejection for the distributed system included both fan and pump energy, and the minimum rejection temperature for the fluid loops was 50^{o}F, versus 70^{o}F for the multiplex condensers. It is likely that the reduction in heat rejection energy can be attributed to the operation of the water-source heat pumps for space heating. In general, the energy associated with heat rejection accounted for 15 - 20% of the total refrigeration energy for both the multiplex and distributed refrigeration systems.

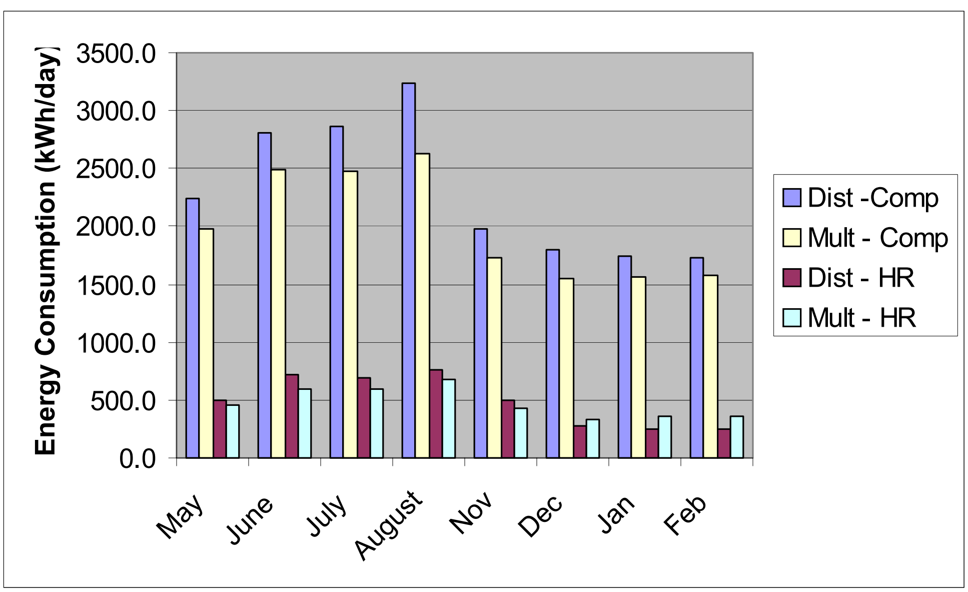

Figure 1 - Average daily energy consumption for the Multiplex and Distributed refrigeration systems

Table 8 provides a description of the operating state points, refrigeration loads, and energy efficiency ratios (EER's) measured for each refrigeration system for summer and winter operation. The operating state points (i.e.

saturated suction and discharge temperatures, and return gas and liquid refrigerant temperatures) have been combined for each system in terms of low and medium temperature refrigeration. The average state point values were calculated on the basis of refrigeration load by

$$SP_{avg} = \frac{\sum SP_i Q_i}{\sum Q_i}$$

where

SP_{avg} = the average state point value
SP_i = the state point value for a particular suction group
Q_i = the refrigeration load associated with the suction group

The refrigeration load measurements indicated that the low temperature refrigeration loads are significantly different for the multiplex and distributed systems and the load for the distributed system is lower than that of the multiplex system. The medium temperature load for the distributed system is higher than that measured for the multiplex system.

Normalization of the test results was done by comparing EER values. This comparison shows that the multiplex system had significantly higher EER values for low and medium temperature refrigeration for both summer and winter operation. For summer operation the multiplex system EER values were 34.7 and 18.5% higher than the EER values of the distributed system for low and medium temperature refrigeration, respectively. For winter operation the multiplex system EER values were higher than those of the distributed system by 12.1 and 22.2% for low and medium temperature refrigeration, respectively. Figure 2 shows the EER values for both systems as a function of average daily dry-bulb temperature. The results in this plot show that the EER values for the multiplex system were consistently higher than those of the distributed system.

Table 8 - Average State Points , Refrigeration Loads, and EER's for the Multiplex and Distributed Refrigeration Systems

	May - Aug		Nov - Feb	
Low Temp	Distributed	Multiplex	Distributed	Multiplex
SST (°F)	-15.8	-19.2	-17.1	-20.2
SDT (°F)	90.0	82.8	61.0	71.0
Return Temp (°F)	11.5	2.2	16.3	11.0
Liquid Temp (°F)	79.1	57.8	56.9	50.4
Ref Load (Btu/h)	340,800	455,600	309,000	385,900
Comp Power (kW)	54.4	53.8	36.0	39.9
EER (Btu/W-h)	6.34	8.54	8.63	9.67
Med Temp	Distributed	Multiplex	Distributed	Multiplex
SST (°F)	16.2	22.9	13.4	20.2
SDT (°F)	92.5	83.3	72.1	67.0
Return Temp (°F)	46.4	48.9	44.3	49.3
Liquid Temp (°F)	85.0	68.1	67.1	58.3
Ref Load (Btu/h)	729,900	653,200	541,000	442,800
Comp Power (kW)	66.4	50.1	39.6	26.5
EER (Btu/W-h)	11.16	13.22	13.71	16.75

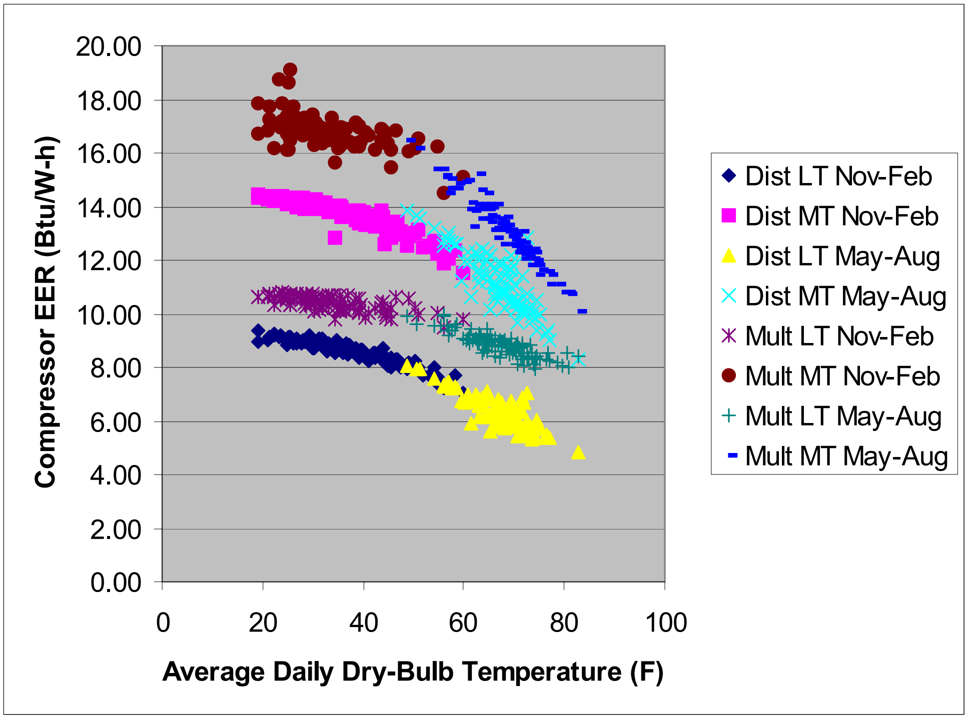

Figure 2 - EER Comparison between the Multiplex and Distributed Refrigeration

A TEWI analysis of each test system was conducted and the results of this analysis are given in Table 9. The charge sizes for the two systems were estimated at 3,000 lb. of refrigerant for the multiplex system and 1,000 lb. for the distributed system. Actual refrigerant leakage rates for the systems are not known. The estimated leakage rates employed in the analysis are 20 and 5 % for the multiplex and distributed systems, respectively. The analysis showed that the distributed system had a lower TEWI for both the summer and winter periods despite the higher indirect value due to the higher energy consumption.

Table 9 - TEWI Analysis for the Field Test Results

	Refrigerant Leakage (lb)		Energy (kWh)	TEWI (kg CO2)		
	R-404A	R-22		Direct	Indirect	Total
May - August						
Multiplex	67	133	381,300	202,055	247,845	449,900
Distributed	17		443,169	24,895	288,060	312,954
Difference						136,046
Nov - Feb						
Multiplex	67	133	234,960	202,055	152,724	354,779
Distributed	17		255,600	24,895	164,034	188,929
Difference						165,850
Multiplex leak rate estimated at 20%/yr Distributed leak rate estimated at 5%/yr						

The field test results for the water-source heat pumps are given in Table 10 for the winter period from November 2001 through February 2002. The table lists the average recovery rate of heat from the fluid loops. The amount of heat recovered was estimated at 25.9% of the total heat rejection of the refrigeration system. This amount is considerably less than the capability of the heat pumps, which can recover as much as 663,200 Btu/h, or 59.8% of the rejected heat from the refrigeration system. The ambient conditions during this time period were very mild so that the amount of space heat needed for the store was much less than normally seen. For much of the time, the large heat pumps had only one compressor operating. The small heat pump had a run fraction of approximately 30% during this time period.

Table 10 - Field Test Results for the Water-Source Heat Pumps Space Heating Performance (November 2001 - February 2002)

	Average Recovered Heat (Btu/h)	Average Space Heat (Btu/h)	Heat Pump Energy (kWh)	Gas Displaced (therms)
November 01	248,400	326,100	17,236	2,966
December 01	293,500	381,000	20,531	3,602
January 02	301,900	394,700	21,004	3,702
February 02	307,200	404,700	20,295	3,446
Total			79,066	13,716

Despite limited operation, the water-source heat pumps were able to displace approximately 13,716 therms of natural gas. The value of this displacement is dependent upon the utility rates for electric and gas. Figure 3 shows the estimated energy cost savings for a range of utility rates. For this particular site, the average commercial electric rate is approximately $.09/kWh and the gas rate is $.75/therm. The estimated energy cost savings seen over this 4-month period were $3,171.

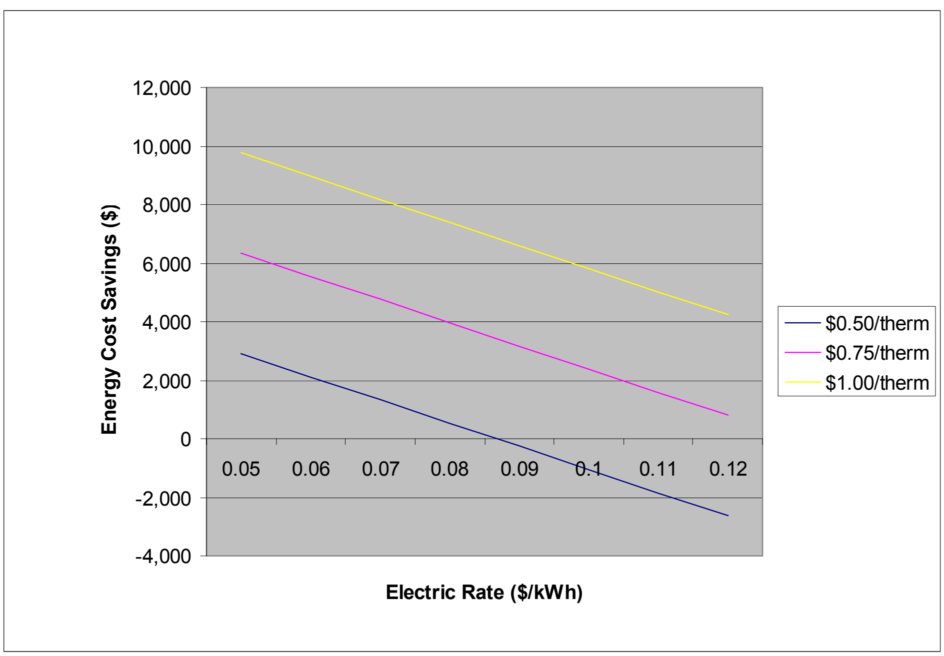

Figure 3 - Water-Source Heat Pump Energy Cost Savings (Nov '01 - Feb '02)

2.3 Conclusions and Recommendations

The distributed refrigeration system consumed much more energy and operated at much lower EER values relative to the baseline system than expected based on the projections from the earlier analytical studies [2,3]. This increased energy use can be attributed to several factors:

- Use of dry fluid coolers for heat rejection - Lower energy consumption should be expected if evaporative heat reject was used for the fluid loops. This is particularly true for summer operation. With evaporative heat rejection the fluid loop temperature could be dropped below the ambient dry-bulb temperature and would help to lower the condensing temperatures seen at the compressor cabinets. Evaporative heat rejection would also require less fan power than dry heat rejection, which would help to lower energy consumption. It should be noted that evaporative heat rejection can be used with any supermarket refrigeration system but usually is not due primarily to the increased maintenance requirements.
- The performance of the scroll compressors - Comparison of manufacturer's data for the scroll and reciprocating compressors suggests that the EER of the scroll compressors is less than that of the reciprocating compressors at the same saturated suction and discharge temperatures. This observation is true for both low and medium temperature compressors. The methods needed to overcome this performance difference include operation of the scroll compressors at lower saturated discharge temperature and at higher saturated suction temperature. The differences in these values obtained by the distributed system tested were not adequate to overcome the performance difference between these two compressor types. It is also desirable to maintain as low a return gas temperature as possible to eliminate excess refrigeration load. The return gas temperature values were about the same for both the multiplex and distributed refrigeration systems despite the shorter piping runs used by the distributed system.
- Extensive mechanical subcooling of the multiplex systems coupled with ineffective vapor injection subcooling for the distributed system compressors - The multiplex system used mechanical subcooling extensively for both the low and medium temperature compressor racks to maintain low liquid refrigerant temperature. This added subcooling produced more of a performance enhancement for the multiplex system than had been originally expected. Subcooling by mid-scroll vapor injection was installed on the scroll compressors in the low temperature compressor cabinets of the distributed system. The subcooling obtained was found to be very limited because the injection ports on the scroll compressors were sized for liquid injection and were too small to allow adequate vapor flow.

Some of the performance difference between the two systems can also be attributed to the use of R-22 for medium temperature refrigeration on the multiplex system and R-404A on the distributed system. This difference will disappear in later installations after the phase-out of R-22.

TEWI analysis for the two systems indicates that the distributed system has less environmental impact with lower total CO_2 production than the multiplex system. This result was obtained even though the indirect CO_2 production of the distributed system was greater due to higher energy consumption.

The distributed system that was tested did not have compressor cabinets located in the sales area of the supermarket. The compressor cabinets were located either in backrooms around the perimeter of the store, or above several of the walk-in coolers. This fact suggests that other compressor types, such as reciprocating could have been used in the cabinets in this installation without concern about excessive noise in the sales area. Small rooftop refrigeration units may also be an effective alternative for a distributed refrigeration system. The use of more efficient reciprocating compressors could help to reduce the energy difference between the distributed and multiplex systems.

Operation of the water-source heat pumps in conjunction with the distributed refrigeration system was shown to be effective at recovering reject heat from the refrigeration system for space heating. Energy cost savings were found for the test site due to this heat recovery by the heat pumps. It should be noted that this recover approach could be applied to any supermarket refrigeration system that employs a fluid loop for heat rejection and is not limited to the distributed system.

Future project work calls for more analysis of the field test data to further quantify the differences between the multiplex and distributed systems. Field test data will be used to validate system models for both refrigeration systems. The models will then be used to determine ways to change operation or design of the distributed system to improve its energy performance.

3. References

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4. Acknowledgements

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Table 1 - Description of the Multiplex Refrigeration Test Store

Circuit	Case Length, No. of Doors, or Walk-in Floor Area	Design Refrigeration Load (Btu/hr)
Rack 1 Low Temperature - Sat. Suction Temp -20°F Refrigerant - R404A		
Reach-in Frozen Food	76 Doors	114,000
Multi-Deck Frozen Meat	40 ft	60,520
Multi-Deck Frozen Fish	18 ft	27,234
Multi-Deck Bakery	12 ft	18,156
Walk-in Freezers	925 ft^2	69,375
Ice Maker		36,000
Rack 1 Low Temperature - Sat. Suction Temp -30°F		
Coffin Ice Cream	104 ft	74,600
Coffin Shrimp	16 ft	5,840
Walk-in Ice Cream	180 ft^2	17,100
Rack 2 Medium Temperature - Sat. Suction Temp 20°F Refrigerant - R22		
Reach-in Dairy	46 Doors	62,560
Single-Deck Produce	56 ft	53,200
Multi-Deck Produce	24 ft	32,160
Service Meat & Deli	64 ft	23,692
Multi-Deck Deli	48 ft	69,920
Coffin Cheese & Deli	16 ft	24,800
Tables Fish	20 ft	10,000
Cooler Floral	140 ft^2	10,500
Floral Display	8 ft	10,336
Rack 2 Medium Temperature - Sat. Suction Temp 15°F		
Bakery Display	15 ft.	12,550
Multi-Deck Meat	68 ft	91,800
Isle Cheese & Deli	56 ft	63,360
Isle Ready Meals	52 ft	57,200
Rack 3 Medium Temperature - Sat. Suction Temp 20°F Refrigerant - R22		
Walk-in Coolers	2,195 ft^2	197,475
Rack 3 Medium Temperature - Sat. Suction Temp 35°F		
Meat Prep Room	1,010 ft^2	141,400
Produce Prep Room	520 ft^2	39,000
Mechanical Subcooling		361,672

Table 2 - Description of the Distributed Refrigeration Test Store

Circuit	Case Length, No. of Doors, or Walk-in Floor Area	Design Refrigeration Load (Btu/hr)
Cabinet A Sat. Suction Temp -25°F		
Walk-in Freezer	1,013 ft^2	76,000
Cabinet B Sat. Suction Temp -25°F		
Walk-in Freezer	506 ft^2	38,000
Cabinet B Sat. Suction Temp -15°F		
Reach-in Frozen Food	23 Doors	34,500
Cabinet C Sat. Suction Temp -15°F		
Reach-in Frozen Food	52 Doors	67,500
Multi-Deck Frozen Food	8 ft	11,880
Cabinet D Sat. Suction Temp -25°F		
Multi-Deck Frozen Meat	56 ft	57,120
Multi-Deck Frozen Fish	28 ft	23,740
Walk-in Frozen Fish	144 ft^2	12,000
Cabinet E Sat. Suction Temp 20°F		
Multi-Deck Dairy	120 ft	160,800
Multi-Deck Cheese	14 ft	18,760
Walk-in Dairy	857 ft^2	62,000
Cabinet F Sat. Suction Temp 17°F		
Service Deli	84 ft	25,920
Multi-Deck Cheese	20 ft	28,800
Isle Deli	16 ft	38,388
Walk-in Deli	153 ft^2	15,300
Walk-in Raw	48 ft^2	4,800
Cabinet F Sat. Suction Temp -20°F		
Walk-in Deli Freezer	81 ft^2	6,075
Cabinet G Sat. Suction Temp 17°F		
Single-Deck Meat	24 ft	10,680
Multi-Deck Meat	68 ft	91,800
Floral Display	24 ft	31,008
Multi-Deck Deli	28 ft	42,420
Single-Deck Fish	16 ft	17,600
Walk-in Fish	110 ft^2	11,000
Cabinet H Sat. Suction Temp 35°F		
Meat Prep Room	6,144 ft^2	153,600
Cabinet H Sat. Suction Temp 5°F		
Ice Maker		36,000
Cabinet I Sat. Suction Temp 20°F		
Multi-Deck Produce	116 ft	122,680
Table Fish	15	7,500
Walk-in Produce	984 ft^2	66,520
Cabinet J Sat. Suction Temp 15°F		
Walk-in Meat	1,056 ft^2	79,200
Walk-in Bakery	120 ft^2	19,800
Cabinet J Sat. Suction Temp -20°F		
Walk-in Freezer Meat	100 ft^2	7,650
Walk-in Freezer Bakery	144 ft^2	10,800
Multi-Deck Bakery	14 ft	15,780
All Compressor Cabinets employ R-404A as the refrigerant		