The GreenChill Advanced Refrigeration Partnership

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Theoretical Analysis of Alternative Supermarket Refrigeration Technologies

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1. INTRODUCTION

EPA is developing a voluntary partnership with the supermarket industry to facilitate the transition from ozone-depleting substances to ozone-friendly alternatives. Known as the GreenChill Advanced Refrigeration Partnership, the overall goal of this program is to promote the adoption of technologies, strategies, and practices that lower emissions of ozone-depleting substances (ODS) and greenhouse gases (GHGs) through both the reduction of refrigerant emissions and the increase of refrigeration systems' energy efficiency. Specific partnership goals are to provide supermarkets and organizations that support the supermarket industry with information and assistance to:

- Transition to non-ODS refrigerants
- Reduce both ODS and non-ODS refrigerant emissions
- Promote supermarkets' adoption of alternative refrigeration technologies that offer qualities such as:
    - Reduced ODS/GHG emissions (e.g., through reduced refrigerant charges and leak rates)
    - Potential for improved energy efficiency
    - Reduced maintenance and refrigerant costs
    - Extended shelf life of perishable food products
    - Improved system design, operations, and maintenance
- Reduce the total impact of supermarkets on ozone depletion and global warming

A key component of the GreenChill Partnership is to facilitate technological research and information-sharing to assist partners in meeting these goals. EPA, in conjunction with the Food Marketing Institute (FMI), determined that one area where information is currently limited involves assessment of the energy efficiencies and energy consumption of currently available, alternative supermarket refrigeration systems. Consequently, EPA commissioned this study to compare the energy consumption of alternative supermarket refrigeration technologies. The study is based on theoretical analyses of the energy efficiency of the three most common refrigeration technologies:

- *Direct-expansion (DX) centralized systems*. In a direct expansion system, the compressors of one suction group are mounted together on a rack and share suction and discharge refrigeration lines. Liquid and suction lines run throughout the store, feeding refrigerant to the cases and coolers and returning refrigerant vapor to the suction manifold. The compressor racks are located in a separate machine room, either in the back of the store inside or outside of the building, or on its roof, to reduce noise and prevent customer access. Condensers are usually air-cooled and hence are placed outside to reject heat. These multiple compressor racks operate at various suction pressures to support refrigerated fixtures (i.e., display cases, coolers, freezers, and some other small consumers) operating at different evaporating temperatures. The hot gas from the compressors is piped to the condenser and converted to liquid. The liquid refrigerant is then piped to the receiver and distributed to the fixtures by the liquid supply lines. After evaporating in the fixtures, the refrigerant returns in suction lines to the suction manifold and the compressors.

- *Secondary-loop, secondary-coolant, centralized systems (SC)*. Two fluids are used in secondary loop systems: the first is a secondary coolant, which is pumped throughout the store to remove heat from the refrigerated fixtures, and the second fluid is a refrigerant used to cool the cold fluid. Secondary loop systems can operate with two to four separate loops and chiller systems depending on the temperatures needed for the display cases. Secondary loop systems use a much smaller refrigerant charge than traditional direct expansion refrigeration systems.
- *Distributed systems (DS).* Unlike traditional direct expansion refrigeration systems, which have a central refrigeration room containing multiple compressor racks, distributed systems use multiple smaller rooftop units that connect to cases and coolers, using considerably less piping. The compressors in a distributed system are located near the display cases they serve – on the roof above the cases, behind a nearby wall, or even on top of or next to the case in the sales area. Thus, distributed systems typically use a smaller refrigerant charge than DX systems.[1]

The analysis uses primarily existing thermo-physical data for refrigerants and secondary-coolant fluids, as well as performance characteristics from existing laboratory and field measurements, and manufacturers' data. A significant attempt was made to reach beyond traditional theoretical/academic studies and to reflect current best practices of the supermarket industry.

[1] GreenChill Advanced Refrigeration Partnership Web Site. Advanced Refrigeration Technology. http://www.epa.gov/greenchill/alttechnology.html

2. STUDY APPROACH

This study was conducted with input from EPA and a Technical Review Committee, convened by EPA, that includes GreenChill partners and EPA representatives (see Appendix A for a list of GreenChill Technical Review Committee members). Cryotherm developed a study work plan that outlined an approach for conducting a theoretical study comparing the energy usage for six supermarket refrigeration scenarios. EPA and Cryotherm presented the initial work plan to the Technical Review Committee during a conference call held on July 13, 2007, with a follow-up call on August 9, 2007. Based on these discussions, the baseline and alternative scenarios were redefined, three cities were chosen to represent different climates to be investigated, and a detailed set of parameters that could affect the performance of supermarket refrigeration systems was developed. Cryotherm and EPA presented initial results and conclusions of the theoretical study at FMI's Energy and Technical Services conference held September 9-12, 2007 in Denver, Colorado.

The general approach for conducting this study involved the following steps:

1. **Define Baseline and Alternative Scenarios.**
   Based on input from the Technical Review Committee, the following baseline and alternatives were defined:

Baseline:	New supermarket with a DX refrigeration system using an HFC refrigerant (DX).
Alternative A:	New supermarket with a low temperature (LT) DX and medium temperature (MT) glycol secondary loop refrigeration system using an HFC refrigerant (MTS).
Alternative B:	New supermarket with a LT secondary loop refrigeration system and a MT secondary loop refrigeration system, each using an HFC refrigerant (SC).
Alternative C:	New supermarket with a distributed refrigeration system using an HFC refrigerant (DS).

2. **Identify geographic locations for study analysis.**
   The Technical Review Committee, EPA, and Cryotherm selected three cities on which to conduct the analysis: Atlanta, Georgia; Boulder, Colorado; and Philadelphia, Pennsylvania. These cities were selected to represent both different climates in the U.S. and locations that are near the GreenChill partners' stores.

3. **Identify general parameters affecting the performance and energy efficiency of supermarket refrigeration systems (Section 3).**
   Cryotherm developed a list of parameters affecting alternative supermarket refrigeration systems, based on a literature review and experience in designing and analyzing advanced refrigeration systems. Three groups of parameters were identified:

   - Parameters determined by the ambient conditions at the location of the store,

- Parameters determined by the indoor conditions in the store, and
- Parameters defined by the type of refrigeration system, its design features, and its interaction with the outdoor and indoor ambient conditions.

4. **Specify the design and operational features of each refrigeration system (Section 4 and Appendix B).**
   This step involved considerable input from the Technical Review Committee. Based on an existing store layout (including piping and refrigerated fixtures, such as display cases, coolers, and freezers), the specific design and operational features also reflect the variety of technical and design approaches, geographic locations, store sizes, and other experiences represented by the committee members and their supermarket chains. The list of parameters developed through this consensus process with the Technical Review Committee was presented in a Phase 1 report submitted to EPA on August 6, 2007 and is provided in Appendix B.

   The level of detail described for these parameters was appropriate for a detailed engineering analysis of the baseline and alternative scenarios. It was, therefore, necessary to use these specifications as the basis for defining a more simplified set of parameters that realistically reflect currently-designed supermarket refrigeration systems that could be analyzed from a more theoretical perspective. The temperature levels and refrigeration loads are based on actual store(s) recently or soon to be constructed. While the detailed set of parameters defined multiple temperature levels for the baseline and each alternative, the theoretical study assesses a single temperature level for the medium and low-temperature refrigeration systems (i.e., the Baseline and Alternatives A and B) and two temperature levels for the medium-temperature distributed system (i.e., Alternative C). The medium-temperature and the low-temperature refrigeration loads in the theoretical study are similar to the corresponding loads defined for a detailed engineering analysis. The key conditions assumed for the theoretical study are described below and a more detailed description of these parameters is provided in Section 4.

   - Baseline: DX system consisting of a medium-temperature suction group with a saturation suction temperature at +20°F corresponding to evaporating temperature at the MT refrigerated fixtures at +22°F and a low-temperature suction group with a saturation suction temperature at -20°F corresponding to evaporating temperature at the LT refrigerated fixtures -18°F.
   - Alternative A: Secondary-coolant medium-temperature system with SST = 17°F providing +22°F supply temperature of the secondary coolant and a DX low-temperature suction group with a saturation suction temperature at -20°F.
   - Alternative B: Secondary-coolant system consisting of a medium-temperature circuit with a secondary-coolant supply temperature at +22°F and a low-temperature circuit with a secondary-coolant supply temperature at -18°F.
   - Alternative C: Distributed system consisting of two medium-temperature suction groups, at 20°F and 25°F, and one low-temperature suction group at -20°F.

5. **Develop Energy Analysis Methodology (Section 5).**
Cryotherm developed a methodology for estimating the annual energy consumption for the baseline and alternative scenarios at each of the geographic locations (i.e., Atlanta, Boulder, and Philadelphia). This involved estimating the power input into compressors and circulation pumps for each refrigeration circuit/system in the store differentiated by suction groups and supply temperatures of secondary coolants. The power input for a given suction group was determined as a function of the ambient temperature and the cooling capacity. The ambient temperatures were divided into 5°R groups (bins). The power input of each system/circuit was determined for the average temperature in each bin. The "WYEC2 Weather Year for Energy Calculations 2" software of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) was used as a source of the weather data and hourly frequency of occurrence of each ambient temperature for the three locations. Manufacturers' data were used as the source of compressor performance data and system energy efficiency ratios (EER), used in calculating the compressors' power input. The EER was determined as a function of the saturated suction temperature (SST) in the analyzed system/circuit, useful superheat in the refrigerated fixture, return-gas temperature to the compressors, liquid-refrigerant subcooling into the refrigerated fixture, and the saturation discharge temperature (SDT). The saturated discharge temperatures were approximated with condensing temperatures. The condensing temperatures were correlated with the ambient temperature in each bin by adding a temperature difference of 10°R. In the range of ambient temperatures for which the compressor SDT would fall below the set minimum, the EER at the minimum allowable SDT was used. The energy consumption at the average ambient temperatures in each bin was determined as the product of the corresponding power input in the bin and the number of hours in the bin for each location. The annual energy consumption was then calculated as the sum of the energy consumption in all bins.

6. **Conduct Analysis and Present Results (Section 6).**
Cryotherm conducted the energy analysis and described the study findings. The results compare energy consumption by type of system, by baseline vs. alternatives, and by location. For the baseline and each alternative, Cryotherm developed a set of three tables showing the energy consumption per bin and annual energy consumption at each location: Atlanta, GA; Boulder, CO, and Philadelphia, PA. Cryotherm summarized the results in a table by suction groups, technologies, and locations. The summary results are also presented graphically in a bar chart showing the annual energy consumption for each of the analyzed technologies.

7. **Analyze Results (Section 7).**
Cryotherm analyzed the annual energy consumption results, comparing the energy consumption of each alternative with the baseline system, by geographical location. Factors that affect the energy efficiency and energy consumption of each alternative are discussed.

8. **Present Conclusions and Recommendations for Next Steps (Section 8).**
This section presents final conclusions and suggests next steps for future and/or more detailed analyses of the energy efficiency and energy consumption of alternative supermarket refrigeration systems.

3. PARAMETERS AFFECTING THE PERFORMANCE AND ENERGY EFFICIENCY OF A SUPERMARKET REFRIGERATION SYSTEM

The major parameters affecting the performance and energy efficiency of a supermarket refrigeration system reflect the ambient conditions, indoor conditions, and system design features. The system operational parameters are a consequence of the system interaction with the ambient and indoor conditions. The general parameters under consideration are:

- Ambient conditions
    - Store location
    - Ambient temperature
- Indoor data
    - Indoor temperature
    - Humidity
- System Design Features
    - Refrigeration loads
    - Suction saturation temperature
    - Discharge saturation temperature
    - Liquid refrigerant subcooling
    - Refrigerant vapor superheat
    - Type of system (e.g., DX, SC or DS)
    - Refrigerant selection
    - Secondary coolant selection
    - Components selection

4. DESIGN AND OPERATIONAL FEATURES AFFECTING THE PERFORMANCE AND ENERGY EFFICIENCY OF REFRIGERATION SYSTEMS

The theoretical study was performed based on the parameters, assumptions, and conditions that affect refrigeration system performance and energy, described below. As described in the study approach, these study parameters, assumptions, and conditions were developed based on input, experience, and review from EPA and the Technical Review Committee. A summary of the key conditions is presented in Table 1, and Table 2 describes the parameters organized by the baseline and each alternative. Figures 1, 2, and 3 illustrate the piping layout for the DX baseline and Alternative C (DS) systems, Alternative A (MTS), and Alternative B (SC), respectively.

4.1 Systems to be investigated

Baseline: Supermarket with a DX refrigeration system with HFC-404A as the refrigerant (DX).

Alternative A: Supermarket with a low-temperature DX and medium-temperature propylene glycol secondary-coolant refrigeration system using HFC-404A as the refrigerant (MTS).

Alternative B: Supermarket with both MT and LT secondary-coolant refrigeration systems using HFC-404A as the refrigerant in the primary systems (SC).

Alternative C: Supermarket with distributed refrigeration systems with HFC-404A as the refrigerant (DS)

4.2 Store size, location, and assumptions

1. The baseline and alternative stores are each 45,000 sq. ft.
2. Stores consist of a medium-temperature (MT) refrigeration system with a refrigerating load of 856,079 Btu/h at a saturated suction temperature of +20°F and a low-temperature system (LT) with a refrigerating load of 300,000 Btu/h at a saturated suction temperature of -20°F. These loads were chosen to closely match the total load and approximate distribution in an actual store (i.e., recently or soon to be constructed).
3. The refrigeration loads are from the refrigerated fixtures only. The load from the mechanical subcooling of the LT liquid refrigerant is added to the MT load.
4. All systems use HFC-404A as the refrigerant.
5. Locations for the analysis are Atlanta, GA; Boulder, CO; and Philadelphia, PA.
6. Heat reclaim and defrost method are excluded from the analysis.
7. Heating and air-conditioning loads, building fire and safety code, store lighting, plug loads and other loads, and the HVAC annual consumption are excluded from this study.

8. The analysis for the baseline and all alternatives use the energy efficiency ratio (EER) of a representative compressor based on manufacturer's data calculated at the specified operating conditions for each alternative technology.

Nomenclature

DX	Direct expansion
IHX	Intermediate heat exchanger (evaporator/chiller)
LT	Low-temperature
MT	Medium-temperature
MTS	Medium-temperature secondary
MSC	Mechanical subcooling, °R
NSC	Natural subcooling, °R
RGT	Return-gas temperature, °F
SC	Secondary coolant
SCST	Secondary-coolant supply temperature, °F
SDT	Saturation discharge temperature, °F
SST	Saturation suction temperature, °F
TD	Temperature difference, °R

4.3. Conditions for the analysis

1. Number of suction groups, secondary-coolant circuits and refrigeration loads:
   a. Baseline: one LT DX suction group with a saturation suction temperature of -20°F, yielding an evaporating temperature of -18°F at the refrigerated fixtures and one MT DX suction group with a saturation suction temperature of +20°F, yielding an evaporating temperature of 22°F at the refrigerated fixture.
   b. Alternative A: one LT DX suction group with a saturation suction temperature of -20°F, yielding an evaporating temperature of -18°F, and one secondary-coolant circuit with SST 17 yielding a +22°F secondary-coolant supply temperature. The refrigeration loads from the refrigerated fixtures in the MT and LT circuits are the same as in the baseline.
   c. Alternative B: one MT and one LT SC circuit with +22°F and -18°F secondary-coolant supply temperature, respectively. The corresponding SST are 17°F in the MT and -23°F in the LT circuit. The refrigeration loads from the refrigerated fixtures in the MT and LT circuits are the same as in the baseline.
   d. Alternative C: one LT DX suction group with saturation suction temperature -20°F and two MT suction groups with saturation suction temperatures of +25°F and +20°F. The LT refrigeration load from the refrigerated fixtures is the same as in the baseline. The MT refrigeration load is distributed as follows: 450,000 Btu/hr at 20°F and 406,079 Btu/hr at SST at 25°F. The load from the mechanical subcooling of the LT liquid refrigerant is added to the load of the group with SST of 25°F.
2. Compressor return gas temperature: 45°F
3. Useful superheat in the DX refrigerated fixtures, mechanical sub-cooler, and intermediate heat exchanger (IHX):
   a. MT: 5°R
   b. LT: 15°R
   c. Mechanical sub-cooler and IHX: 10°F
4. Mechanical subcooling (MSC) of the LT liquid refrigerant by the MT refrigerant:
   a. Baseline: to 50°F

b. Alternative A (MTS): to 50°F.

c. Alternative B (SC): to 50°F, 40°F, and 30°F.

d. Alternative C (DS): to 50°F.

5. Impact of heat gains/losses in the liquid refrigerant lines on subcooling at the display cases and intermediate heat exchanger (IHX): neglected.

6. Heat gains in DX return lines and in SC supply and return lines: neglected.

7. Condenser temperature difference: 10°R for both MT and LT in all technologies.

8. Natural subcooling in the condensers: 0°R for all systems.

9. Condenser fan control:

   a. Baseline (DX): float SDT to 70°F for MT and LT condensers.

   b. Alternative A (MTS): float SDT to 50°F for MT and to 70°F for LT condensers.

   c. Alternative B (SC): float SDT to 50°F for MT and to 40°F for LT condensers.

   d. Alternative C (DS): float SDT to 70°F for both MT and LT condensers.

   While some supermarket DX systems operate at 50°F SDT, this study assumes floating the condensing temperature to 70°F for the DX systems and 50°F or 40°F for the SC systems. This accounts for the long refrigerant lines in DX systems and the possibility of the liquid refrigerant reaching saturation point at the expansion valves, resulting in malfunction. The shorter liquid refrigerant lines in an SC system allow floating the condensing temperature to lower temperatures without causing problems at the expansion valves.

   The MT SST in the Baseline DX and in Alternative C is assumed to be 20°F. Accounting for a 2°R equivalent pressure drop in the suction line for oil return, this yields an evaporating temperature of 22°F in the evaporator. The MT SST in Alternative A and Alternative B are 17°F. The LT SST in the Baseline DX and in Alternative C is assumed to be -20°F yielding an evaporating temperature of -18°F in the evaporator.

   Assuming a 5°R temperature difference in the MT IHX, a SST of 17°F yields 22°F secondary fluid going to the refrigerated fixtures. Thus, the MT SST in Alt A and B are 3°R lower than the corresponding MT SST in the Baseline DX. Similarly, the LT SST is -20°F for the baseline DX and -23°F for Alternative B.

10. Compressor inlet pressure:

    a. Pressure drop in DX baseline, DX LT, and DS MT and LT suction lines: 2°R equivalent for oil return

    b. Pressure drop in the Alternatives A (MTS) MT and B (SC) MT and LT suction lines: neglected because of the short return lines and the downstream movement of oil.

11. Compressor inlet temperature (Return Gas Temperature): 45°F in DX and DS, 10°R superheat in SC.

12. Secondary-coolant supply/return temperature difference: 7°R
13. Circulation pumps:
    a. The power input into the SC circulation pumps is added to the power input of the compressor racks.
    b. 90% of the heat from the pumps is added to the cooling load from the fixtures.
    c. Pressure head of the LT and MT SC circulation pumps is assumed to be 70 ft. H_2O.
    d. Assumed efficiency (including electric motor efficiency) of the LT and MT SC circulation pumps is 60%.
14. Analysis with Dynalene in the LT SC and Propylene Glycol in the MT SC.
15. Indoor temperature and relative humidity for the study: 75°F /55% year around.

Table 1: Conditions for the theoretical analysis

Technology	System Type	Temp. Level	Notes	SST °F	Max SDT °F	Min SDT °F	Liquid Temp. °F	Refrigerant/Sec. Coolant Temp. at Case/Chiller Outlet °F	RGT °F	Cooling load Btu/hr	Power kW
Baseline	DX	LT	Subcooled by MT	-20	110	70	50	-5	45	300,000	
	DX	MT		20	110	70	SDT	25	45	856,079 + MSC	
Alternative A	DX	LT	Same as Baseline	-20	110	70	50	-5	45	300,000	
	SC	MT		17	110	50	SDT	27	27	856,079 + MSC + PH	add PP
Alternative B	SC	LT	Subcooled by MT	-23	110	40	50, 40, 30	-13	-13	300,000 + PH	add PP
	SC	MT		17	110	50	SDT	27	27	856,079 + MSC + PH	add PP
Alternative C	DS	LT	Subcooled by MT	-20	110	70	50	-5	45	300,000	
	DS	MT		25	110	70	SDT	30	45	406,079 + MSC	
	DS	MT		20	110	70	SDT	25	45	450,000	

MSC = Mechanical subcooling, PH = Pump heat, PP = Pump power, SDT = Saturation discharge temperature, RGT = Return gas temperature

In the theoretical analysis, 2°R of equivalent pressure drop for oil return in the LTDX and MTDX lines has been assumed.

Table 2: Descriptive conditions for the theoretical analysis
Analysis is based on a supermarket refrigeration system with cooling loads close to that in a real store.
Baseline DX system
One MT system 856,079 Btu/h designed for SST/SDT = +20/110°F One LT system 300,000 Btu/h designed for SST/SDT = -20/110°F, subcooled by MT Condenser TD = 10.0°R for both MT and LT Floating condensing pressure to: 70°F in both MT and LT condensers. Equivalent pressure drop in suction lines: 2°R in both MT and LT systems Natural subcooling NSC=0°R in both MT and LT systems Mechanical subcooling of LT liquid refrigerant in MT system to 50°F Return gas temperature 45°F in both MT and LT systems resulting from: • 25°R MT compressor superheat of which 5°R useful superheat in MT evaporators/display cases and 20°R estimated superheat in the return lines • 65°R LT compressor superheat of which 15°R useful superheat in LT evaporators/display cases and 50°R estimated superheat in the return lines
Alternative A, MTS System
One SC MT system 856,079 Btu/h designed for SCST = +22°F, SST/SDT = +17/110°F One DX LT system 300,000 Btu/h designed for SST/SDT -20/110°F, subcooled by MT Condenser TD = 10.0°R for both MT and LT Floating condensing pressure to: 50°F in MT and 70°F in LT condensers. Equivalent pressure drop in suction lines: 0°R in MT SC and 2°R in LT DX Natural subcooling NSC=0°R in both MT and LT systems Mechanical subcooling of LT liquid refrigerant in MT system to 50°F Return gas temperature +27°F in the MT refrigerating circuit resulting from: • 10°R compressor superheat of which 10°R useful superheat in the intermediate heat exchanger and 0°R superheat in refrigerant return lines (assumed no heat gains because of short lengths.) Return gas temperature +45°F in LT resulting from: • 65°R LT compressor superheat of which 15°R useful superheat in LT evaporators/display cases and 50°R estimated superheat in the return lines
Alternative B, SC system
One SC MT system 856,079 Btu/h designed for SCST = +22°F, SST/SDT = +17/110°F One SC LT system 300,000 Btu/h designed for SCST/SDT -18/110°F, SST/SDT = -23/110°F, subcooled by MT Condenser TD =10.0°R for both MT and LT Floating condensing pressure to: 50°F in both MT and LT condensers. Equivalent pressure drop in suction lines: 0°R in both MT and LT Natural subcooling NSC=0°R Mechanical subcooling of LT liquid refrigerant in MT system to 50, 40, and 30°F Return gas temperature +27°F in MT and -13°F in LT resulting from: 10°R useful superheat in both MT and LT IHX Pump Design Head both in MT and LT: 70 ft. H2O Evaporator Design Temp. Difference both in MT and LT: 7°R LT Secondary Coolant: Dynalene HC-30 MT Secondary Coolant: 30% Propylene Glycol Pump Efficiency, both MT and LT: 0.6 (including electric motor efficiency) Pump Heat (% of Pump Work) both in MT and LT: 90%
Alternative C, DS system
One MT system 450,000 Btu/h designed for SST/SDT = +20/110°F One MT system 406,000 Btu/h designed for SST/SDT = +25/110°F One LT system 300,000 Btu/h designed for SST/SDT = -20/110°F Condenser TD = 10.0°R for both MT and LT Floating condensing pressure to: 70°F in both MT and LT condensers. Equivalent pressure drop in suction lines: 2°R in both MT and LT Natural subcooling NSC=0°R Mechanical subcooling of LT liquid refrigerant in MT system with SST +25°F: to 50, 40, and 30°F Return gas temperature 45°F in both MT and LT systems resulting from: • 25°R MT compressor superheat of which 5°R useful superheat in MT evaporators/display cases and 20°R estimated superheat in the return lines • 65°R LT compressor superheat of which 15°R useful superheat in LT evaporators/display cases and 50°R estimated superheat in the return lines
Summary of the general assumptions:
Refrigeration load from the refrigerated fixtures is independent of operating conditions (except for LT subcooling load) Condenser TD is 10.0°R DX Systems designed with 2°R equivalent pressure drop in suction lines, SC with 0°R

Figure 1: Piping diagram of Baseline (DX) and Alternative C (DS)

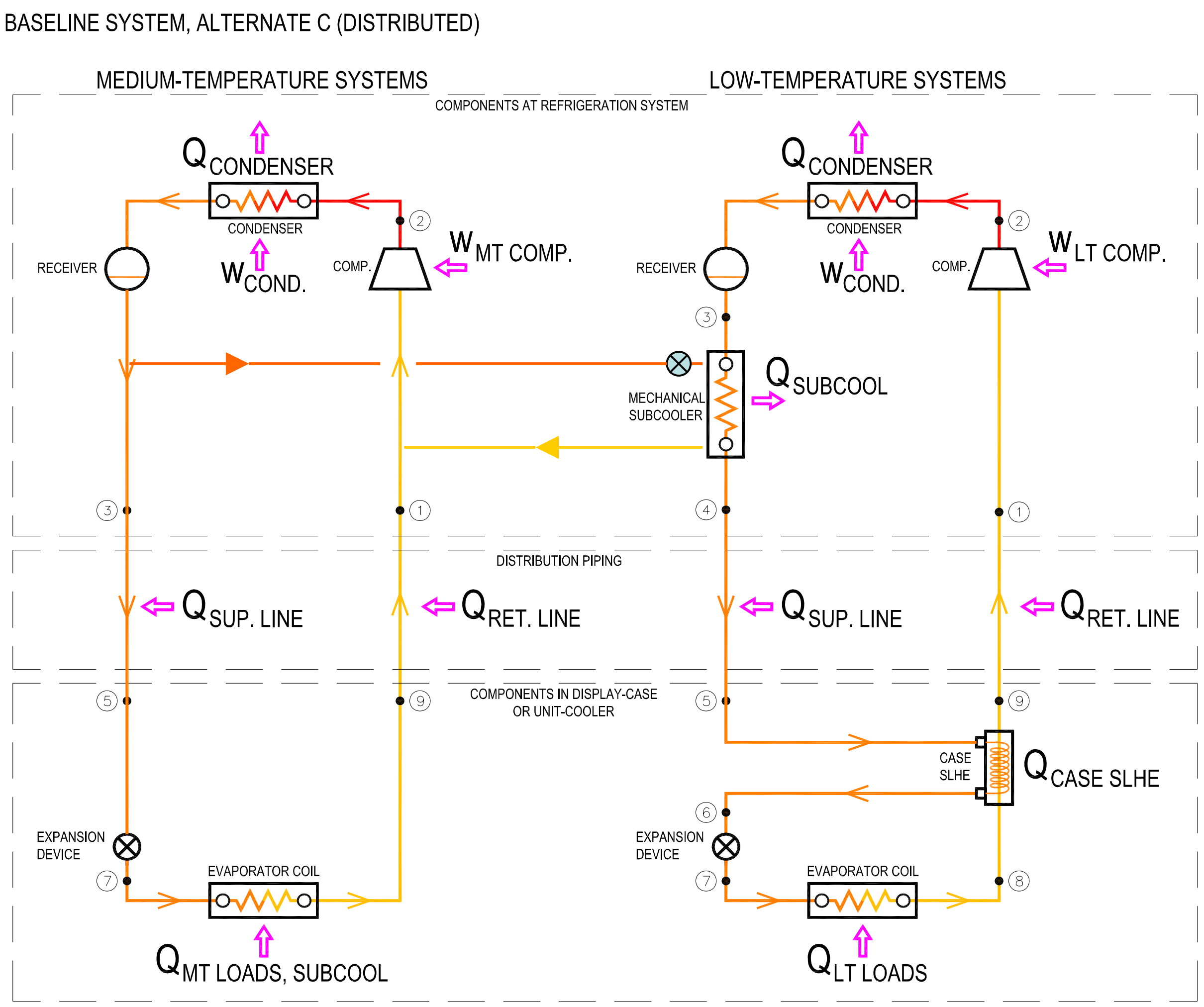


Figure 2: Piping diagram of Alternative A (MTS)

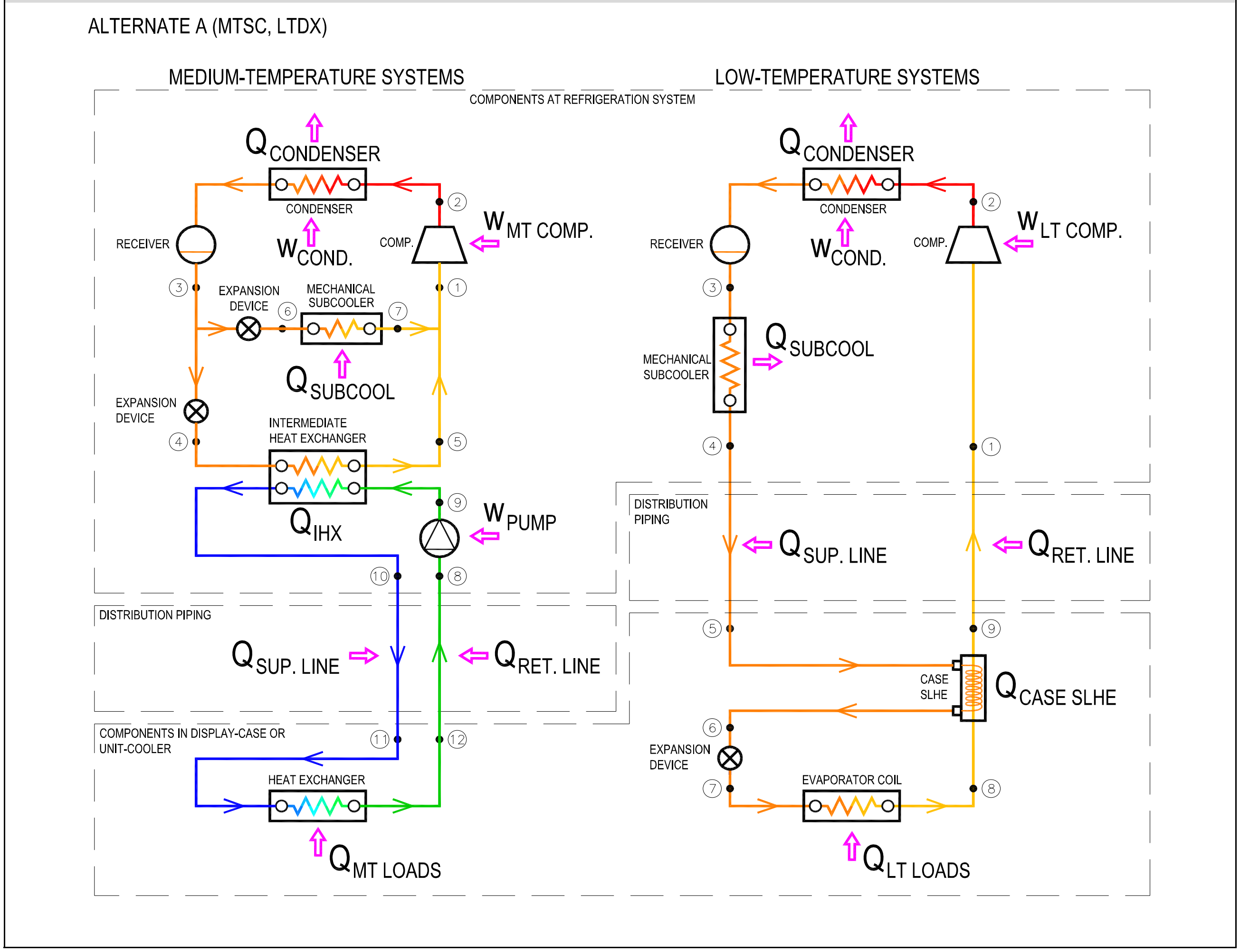
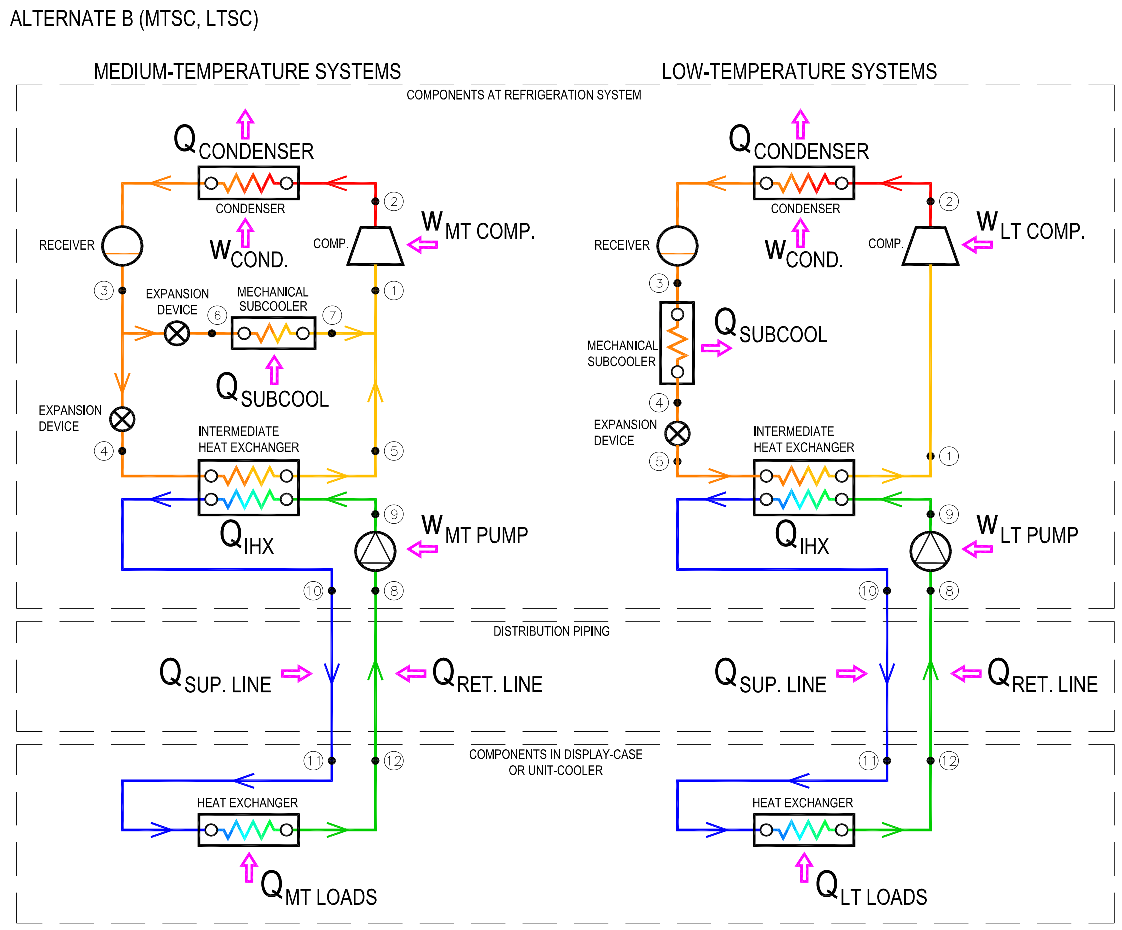


Figure 3: Piping diagram of Alternative B (SC)



5. ENERGY ANALYSIS METHODOLOGY

The energy analysis and comparison of the three alternative technologies with the baseline DX technology was performed based on an estimation of the annual energy consumption at three geographic locations. The annual energy consumption of the baseline and the alternative technologies was determined from the power input into the MT and LT refrigeration systems and the number of operating hours. Since both of these factors vary with the ambient temperature, the calculation was performed across the range of ambient temperatures for each of the three geographic locations during a year. For practical purposes, the range of ambient temperatures was divided into temperature intervals, or “bins.”

5.1. Number of bin hours

Weather statistical data for the three analyzed geographic locations provided the number of hours the temperatures in each bin occur in a year. The source of these weather data was ASHRAE’s WYEC2 Weather Year for Energy Calculations 2. The bin hours for the three locations analyzed in this study are shown in Table 3.

Table 3: Weather Bin Data for Atlanta, GA; Boulder, CO; and Philadelphia, PA

Ambient Temperature Bin °F	Weather Bin Data Atlanta, GA Hours	Weather Bin Data Boulder, CO Hours	Weather Bin Data Philadelphia, PA Hours
95-100	9	22	3
90-95	56	96	52
85-90	196	115	104
80-85	758	382	477
75-80	768	440	656
70-75	1314	489	907
65-70	885	503	619
60-65	1027	907	983
55-60	790	698	625
50-55	673	754	540
45-50	641	762	576
40-45	436	633	552
35-40	560	834	1067
30-35	323	717	685
25-30	181	611	442
20-25	72	251	248
15-20	64	201	184
10-15	7	130	40
5-10	0	89	0
0-5	0	126[a]	0
Total Hours	**8760**	**8760**	**8760**

[a] The number of hours in this bin is the cumulative number of hours of all temperatures within and below the 5°F-bin, thus integrating the 5°F bin and the next lower-temperature bins. The reason is that all these temperatures affect the performance of the refrigeration system in a similar way and can be processed together.

5.2. System power input

The power input into the refrigeration system consists of the power input into the refrigerating compressors, condenser fans, secondary-coolant circulation pumps, refrigerated fixture lights and fans, anti-sweat heaters, and defrost heaters. This study assessed only the power input into compressors and circulation pumps. The power input into refrigerated fixture lights and fans, anti-sweat and defrost heaters were omitted since they affect all technologies equally. In this study, the power input into condenser fans is assumed to be equal among refrigeration technologies; however, in reality there are slight differences. An exact engineering analysis could account for theses differences.

When determining the system power input, it was assumed that it depends only on the ambient temperature and not on the specific time when this ambient temperature occurs. Thus, the energy consumption in each bin reflects the number of hours in the bin and the system power input at the average temperature in the bin. The annual energy consumption is a sum of the energy consumption of all bins.

5.2.1 Power input into compressors

The compressor power input was determined from the system cooling load and the net EER from equation (1). The system cooling load is discussed in Section 5.3. The net EER is determined from the compressor performance characteristics by the SST, SDT, return gas temperature, liquid subcooling, and useful superheat. (These parameters are specified in detail for each technology in Table 2 and illustrated in Figure 4.) The condensing temperature in all technologies was determined by adding the specified temperature difference of 10°R to the ambient temperature. This applies in the range from the highest to the lowest ambient temperature bins at which the condensing temperature has reached its specified minimum value, designated as lowest floating condensing pressure/temperature (see Table 2).

Compressor manufacturers provide compressor performance data in a variety of formats, including tables, curves, equations, and software packages. For this study, performance data for Copeland brand compressors were used. The performance data for LT compressors were derived from the compressor 3DRHF46KE-TFC and are shown in Table 4. and the performance data for MT compressors were derived from the compressor 3DS3R17ML-TFC and are shown in Table 5.

The power input at the average ambient temperature in each bin was determined from the system cooling capacity and the net EER by applying the following equation:

$$\text{Compressor Power Input [W]} = \frac{\text{System cooling capacity [Btu/h]}}{\text{Net Energy Efficiency Ratio [Btu/Wh]}} \qquad (1)$$

5.2.2 Power Input into Circulation Pumps

The power input into the secondary-coolant circulation pumps was determined from the following equation:

$$P_{circ.pump}\,\text{[W]} = \frac{\text{Volumetric Flow Rate [m}^3\text{/s]. Pressure difference [Pa]}}{\text{Efficiency}} \qquad (2)$$

Figure 4: HFC-404A pressure-enthalpy diagram with definitions of key parameters

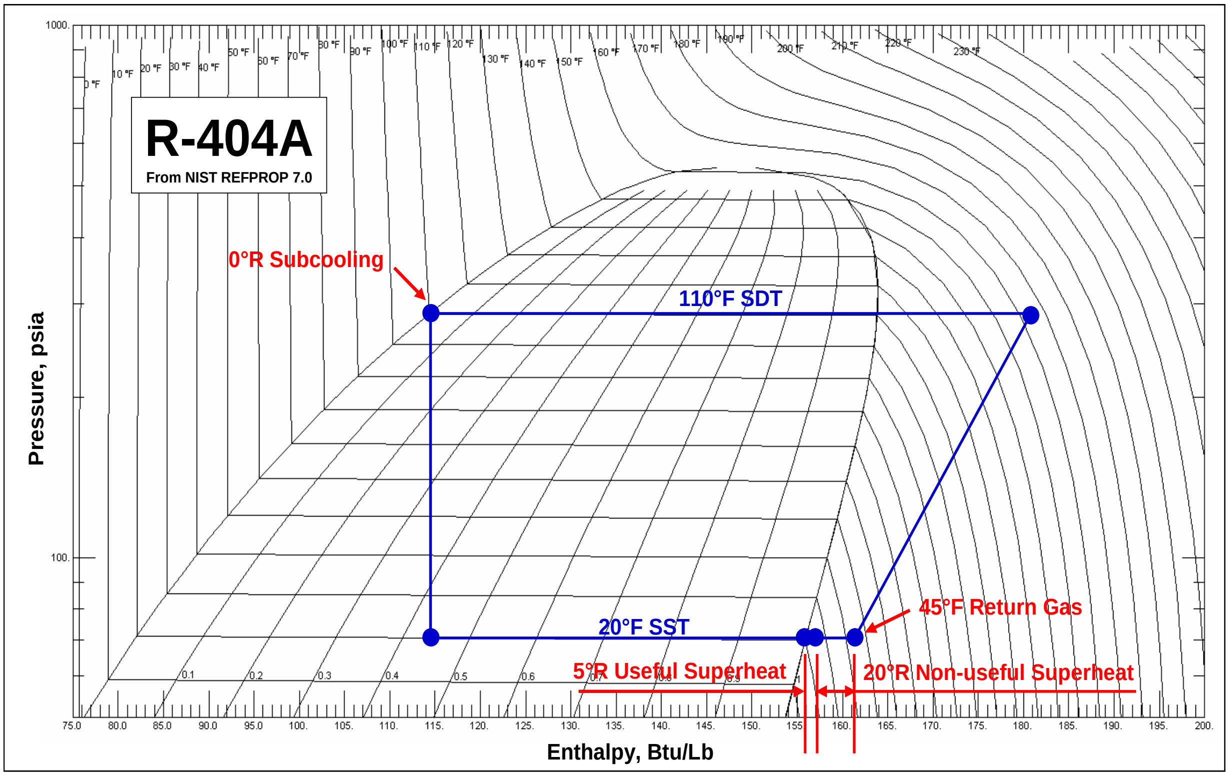


Figure definitions of parameters used in calculating the performance of the refrigeration system and refrigerating compressors: saturation suction temperature (SST), saturation discharge temperature (SDT), subcooling, useful superheat, non-useful superheat (in the return lines), and return gas temperature. Compressor performance characteristics refer to compressor superheat which is the sum of the non-useful and useful superheat. Non-useful superheat is used here only for illustration purposes. In the first order of simplification, suction saturation temperature is used interchangeably with evaporating temperature and saturation discharge temperature is used interchangeably with condensing temperature. In this picture: SST = 20°F, SDT = 110°F, subcooling = 0°R, useful superheat = 5°R, compressor superheat = 25°R, and return gas temperature = 45°F. The source of refrigerant properties is NIST Refprop 7.0.

Table 4: Performance table of a low-temperature compressor at return gas temperature 45°F and zero liquid-refrigerant subcooling. C = Capacity, Btu/hr; P = Power, W; A = Current, A; M = Refrigerant mass flow rate, lb/hr; E = EER, Btu/W-hr; % = Isentropic Efficiency, %. *Note: This table is used as an illustration by permission from Emerson Climate Technologies.*

RATING CONDITIONS
45°F Return Gas
0°F Subcooling
95°F Ambient Air Over

60 Hz Operation

LOW TEMPERATURE

3DRHF46KE-TFC
COPELAMETIC® HFC-404A
DISCUS® COMPRESSOR
TFC 208/230-3-60

Condensing Temperature °F (Sat Dew Pt Pressure, psig)		Evaporating Temperature °F (Sat Dew Pt Pressure, psig)								
		-40(4.5)	**-35**(7.1)	**-30**(9.9)	**-25**(13)	**-20**(16)	**-15**(20)	**-10**(24)	**-5**(28)	**0**(33)
130 (354)	**C**	17900	22500	27300	32300	37600	43200	49100	55500	62500
	P	6350	7250	8150	9000	9850	10700	11500	12300	13000
	A	21.9	23.9	25.9	28	30	32	34	36	38
	M	432	545	665	790	925	1060	1220	1380	1560
	E	2.8	3.1	3.4	3.6	3.8	4	4.3	4.5	4.8
	%	64.7	67.8	69.5	70.5	71	71.2	71.2	71.1	71
120 (310)	**C**	22700	27400	32300	37600	43200	49300	56000	63000	70500
	P	6550	7350	8100	8900	9650	10400	11200	11900	12500
	A	22.3	24.1	25.9	27.7	29.5	31.3	33.1	34.9	36.7
	M	495	600	710	830	955	1100	1250	1410	1590
	E	3.5	3.7	4	4.2	4.5	4.7	5	5.3	5.6
	%	68.9	70.2	70.8	71.1	71.1	71.1	70.9	70.7	70.5
110 (271)	**C**	26600	31400	36600	42200	48300	55000	62000	69500	78000
	P	6600	7300	8000	8700	9400	10100	10700	11400	12000
	A	22.4	23.9	25.6	27.2	28.8	30.4	32	33.7	35.3
	M	530	630	735	850	980	1120	1270	1430	1610
	E	4	4.3	4.6	4.8	5.1	5.4	5.8	6.1	6.5
	%	69.8	70.4	70.7	70.7	70.6	70.5	70.3	70.1	69.8
105 (252)	**C**	28200	33100	38500	44300	50500	57500	65000	73000	82000
	P	6550	7250	7900	8600	9250	9850	10500	11100	11700
	A	22.3	23.8	25.3	26.8	28.4	29.9	31.5	33	34.6
	M	545	640	745	860	985	1120	1270	1440	1620
	E	4.3	4.6	4.9	5.2	5.5	5.8	6.2	6.6	7
	%	69.6	70	70.2	70.3	70.2	70.1	69.9	69.7	69.4
100 (235)	**C**	29700	34800	40300	46300	53000	60000	68000	76500	85500
	P	6550	7150	7800	8450	9050	9650	10200	10800	11400
	A	22.2	23.6	25	26.5	27.9	29.4	30.9	32.3	33.8
	M	550	645	750	865	990	1130	1280	1450	1630
	E	4.5	4.8	5.2	5.5	5.8	6.2	6.6	7.1	7.5
	%	69.1	69.5	69.7	69.8	69.7	69.6	69.5	69.2	68.9
80 (174)	**C**	34500	40100	46400	53500	61000	69500	79000	89500	101000
	P	6200	6650	7150	7650	8150	8650	9150	9650	10100
	A	21.6	22.6	23.7	24.8	26	27.2	28.4	29.6	30.8
	M	560	655	760	875	1000	1150	1310	1480	1680
	E	5.6	6	6.5	7	7.5	8	8.6	9.3	10
	%	65.9	66.6	67	67.3	67.5	67.4	67.2	66.7	66.1
70 (148)	**C**	36400	42500	49300	57000	65500	74500	85000	96000	108000
	P	5900	6350	6750	7250	7700	8150	8600	9000	9450
	A	21.2	22.1	23	24	25	26.1	27.2	28.3	29.5
	M	560	655	760	880	1010	1160	1320	1510	1700
	E	6.2	6.7	7.3	7.9	8.5	9.2	9.9	10.7	11.5
	%	64.2	65.1	65.7	66.1	66.2	66.1	65.7	65	63.9
50 (104)	**C**	40200	47300	55000	64000	74000	85500	97500	111000	125000
	P	5200	5550	5900	6250	6650	7050	7450	7800	8200
	A	20.5	21.2	21.9	22.7	23.5	24.4	25.3	26.3	27.3
	M	555	655	770	895	1040	1200	1370	1560	1780
	E	7.8	8.5	9.4	10.2	11.2	12.1	13.1	14.2	15.3
	%	61.5	62.6	63.2	63.4	63.1	62.3	61	59.2	56.9
40 (86)	**C**	42400	50000	58500	68500	79500				
	P	4790	5100	5450	5800	6150				
	A	20.4	21	21.6	22.3	23.1				
	M	560	665	780	910	1060				
	E	8.8	9.8	10.8	11.8	12.9				
	%	60.5	61.5	61.9	61.6	60.8				

NON-STANDARD CONDITIONS: Nominal Performance Values (±10%) based on 72 hours run-in. Subject to change without notice. Current @ 230 V
C:Capacity(Btu/hr), P:Power(Watts), A:Current(Amps), M:Mass Flow(lbs/hr), E:EER(Btu/Watt-hr), %:Isentropic Efficiency(%)

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Table 5: Performance table of a medium-temperature compressor at return gas temperature 45°F and zero liquid-refrigerant subcooling. C = Capacity, Btu/hr; P = Power, W; A = Current, A; M = Refrigerant mass flow rate, lb/hr; E = EER, Btu/W-hr; % = Isentropic Efficiency, %. *Note: This table is used as an illustration by permission from Emerson Climate Technologies.*

RATING CONDITIONS
45°F Return Gas
0°F Subcooling
95°F Ambient Air Over

60 Hz Operation

MEDIUM TEMPERATURE

HFCs Require Use of Polyol Ester Lubricant Approved by Bulletin AE-1248

3DS3R17ML-TFC
COPELAMETIC® HFC-404A
DISCUS® COMPRESSOR
TFC 208/230-3-60

Condensing Temperature °F (Sat Dew Pt Pressure, psig) / **Evaporating Temperature °F (Sat Dew Pt Pressure, psig)**

Condensing		**-10**(24)	**0**(33)	**5**(38)	**10**(44)	**15**(49)	**20**(56)	**25**(63)	**30**(70)	**35**(78)	**40**(0)	**45**(0)
140 (402)	C	42300	54500	61500	68500	76000	84000	92500	102000	112000	0	0
	P	12000	13600	14400	15200	16000	16800	17600	18500	19200	0	0
	A	38.8	42.6	44.6	46.6	48.6	50.6	52.7	54.7	56.7	0	0
	M	1190	1560	1770	1990	2230	2500	2800	3130	3500	0	0
	E	3.5	4	4.3	4.5	4.7	5	5.3	5.5	5.8	0	0
	%	70.5	72.2	72.3	72	71.6	71.1	70.5	69.8	69.1	0	0
130 (354)	C	49000	63000	70500	79000	87500	97500	108000	119000	131000	0	0
	P	11600	13100	13900	14600	15300	16100	16800	17500	18200	0	0
	A	38	41.6	43.4	45.2	47	48.8	50.6	52.4	54.2	0	0
	M	1220	1580	1790	2010	2260	2530	2840	3170	3550	0	0
	E	4.2	4.8	5.1	5.4	5.7	6	6.4	6.8	7.2	0	0
	%	70.2	71.3	71.4	71.2	70.8	70.4	69.9	69.2	68.5	0	0
120 (310)	C	55500	71000	79500	89000	99500	111000	123000	136000	150000	0	0
	P	11200	12600	13300	14000	14600	15300	15900	16500	17100	0	0
	A	37.2	40.4	42	43.7	45.3	46.9	48.4	49.9	51.4	0	0
	M	1240	1610	1810	2040	2300	2580	2890	3240	3630	0	0
	E	4.9	5.6	6	6.4	6.8	7.2	7.7	8.2	8.8	0	0
	%	70	70.7	70.7	70.5	70.3	69.9	69.4	68.8	68	0	0
110 (271)	C	62000	79000	89000	99500	111000	124000	138000	153000	169000	0	0
	P	10800	12100	12700	13300	13900	14400	14900	15400	15900	0	0
	A	36.2	39.1	40.6	42	43.4	44.8	46.1	47.3	48.5	0	0
	M	1270	1640	1850	2080	2340	2630	2960	3310	3710	0	0
	E	5.7	6.6	7	7.5	8	8.6	9.2	9.9	10.6	0	0
	%	69.7	70.2	70.1	70	69.7	69.3	68.8	68.1	67.3	0	0
100 (235)	C	68500	87500	98000	110000	123000	137000	153000	170000	188000	0	0
	P	10400	11500	12000	12600	13100	13500	13900	14300	14700	0	0
	A	35.1	37.7	39	40.2	41.4	42.5	43.6	44.5	45.4	0	0
	M	1290	1670	1880	2130	2390	2690	3020	3390	3800	0	0
	E	6.6	7.6	8.2	8.8	9.4	10.2	11	11.9	12.8	0	0
	%	69.4	69.5	69.4	69.3	69	68.5	67.9	67.1	66	0	0
90 (203)	C	75000	95500	108000	121000	135000	151000	168000	187000	208000	0	0
	P	9900	10900	11400	11800	12200	12600	12900	13200	13400	0	0
	A	34.1	36.3	37.4	38.4	39.4	40.3	41	41.7	42.3	0	0
	M	1320	1700	1920	2170	2450	2750	3090	3470	3890	0	0
	E	7.6	8.8	9.5	10.3	11.1	12	13.1	14.2	15.5	0	0
	%	68.8	68.7	68.5	68.3	67.9	67.3	66.5	65.4	64	0	0
70 (148)	C	89000	113000	127000	143000	160000	179000	200000	223000	247000	0	0
	P	8800	9550	9850	10100	10300	10500	10600	10600	10600	0	0
	A	32	33.6	34.2	34.8	35.3	35.6	35.8	35.9	35.9	0	0
	M	1390	1780	2010	2270	2560	2880	3240	3630	4070	0	0
	E	10.1	11.9	12.9	14.2	15.6	17.1	18.9	21	23.3	0	0
	%	67.1	66.1	65.6	64.9	63.9	62.7	61	58.9	56.1	0	0
60 (125)	C	96000	122000	138000	155000	173000	194000	217000	241000	268000		
	P	8200	8800	9000	9200	9300	9350	9350	9250	9100		
	A	31.1	32.3	32.7	33.1	33.3	33.4	33.3	33.1	32.7		
	M	1420	1820	2060	2330	2620	2950	3320	3720	4180		
	E	11.7	13.9	15.3	16.9	18.7	20.8	23.2	26.1	29.4		
	%	65.7	64.1	63.2	62	60.5	58.6	56.1	52.9	48.9		
50 (104)	C	104000	132000	148000	167000	187000	209000	234000	261000			
	P	7550	8000	8150	8200	8200	8150	8050	7850			
	A	30.3	31.1	31.3	31.4	31.4	31.2	30.8	30.3			
	M	1460	1870	2110	2380	2690	3030	3400	3820			
	E	13.7	16.5	18.2	20.3	22.7	25.6	29.1	33.3			
	%	63.8	61.2	59.7	57.9	55.5	52.6	48.7	43.6			

NON-STANDARD CONDITIONS: Nominal Performance Values (±10%) based on 72 hours run-in. Subject to change without notice. Current @ 230 V
C:Capacity(Btu/hr), P:Power(Watts), A:Current(Amps), M:Mass Flow(lbs/hr), E:EER(Btu/Watt-hr), %:Isentropic Efficiency(%)

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The secondary-coolant volumetric flow-rates in both the MT and LT systems were determined from the following equation:

$$V\,[ft^3/hr] = \frac{\text{Cooling Capacity [Btu/hr]}}{\text{Density }[lb/ft^3].\text{Specific heat [Btu/lb-°R]}.\text{Delta T [°R]}} \quad (3)$$

The density and the specific heat for propylene glycol, the secondary coolant in the MT system, are shown in Table 6. The density and specific heat for Dynalene HC-30, the secondary coolant in the LT system, are shown in Table 7. The pressure difference is derived from the pressure head in the secondary-coolant systems. The pressure head in both MT and LT systems is 70 ft. H_2O. The delta T is the temperature difference between the secondary coolant supply and return temperatures and is 7°R in both MT and LT systems. The efficiency of the circulation pumps in both MT and LT circuits is 60%.

Table 6: Properties of inhibited Propylene Glycol 30% by weight, freezing point 9.2°F

Fluid Temp. [°F]	Density [lb/ft3]	Specific Heat [Btu/lb°R]
10	64.96	0.901
15	64.91	0.902
20	64.86	0.904
25	64.81	0.906
30	64.75	0.908
35	64.69	0.910
40	64.63	0.911
45	64.57	0.913
50	64.50	0.915
55	64.43	0.917
60	64.36	0.919
65	64.28	0.921
70	64.21	0.922

Table 7: Properties of Dynalene HC-30

Temperature °F	Density lb/ft3	Specific Heat Btu/lb°R
425	73.29	0.8447
70	79.56	0.7360
60	79.74	0.7329
50	79.91	0.7298
40	80.09	0.7268
30	80.27	0.7238
20	80.44	0.7206
10	80.62	0.7176
0	80.80	0.7145
-10	80.97	0.7114
-20	81.15	0.7084
-30	81.33	0.7054

5.3. Cooling load

The major portion of the required cooling capacity used in the calculation of the compressor power input is the cooling load from the display cases, coolers, and freezers. This cooling load is often referred to as a net refrigerating load and is used to determine the system net refrigerating capacity or net refrigerating effect. Additional cooling loads come from small local air-conditioning units and from the mechanical subcooling of the LT liquid refrigerant in the MT refrigeration system.

For efficient operation, the cooling loads are distributed into suction groups. Not all of the above listed load components are present in each suction group. For instance, the mechanical subcooling is piped into the MT with the highest SST. For this study, all MT net cooling loads in the Baseline (DX system) have been combined into one suction group at SST of +20°F and all LT net loads have been combined into one suction group at SST -20°F. The combined MT net cooling load is 856,079 Btu/hr. The combined LT net cooling load is 300,000 Btu/hr. The load from the mechanical subcooler is an additional load to the MT circuit.

The net cooling loads in Alternative A (MTS) have been serviced by one MT secondary-coolant circuit with SCST of +22°F with an associated refrigerant SST of +17°F in the intermediate heat exchanger (evaporator/chiller) and a net load of 856,079 Btu/hr. The load from the mechanical subcooler is added to this circuit. The heat gains from the SC circulation pumps are also added to this circuit. Similar to the baseline, all LT net loads have been combined into one suction group at SST -20°F. The combined LT net cooling load is 300,000 Btu/hr.

The cooling loads in Alternative B (SC) have been serviced by one MT secondary-coolant circuit with SCST (secondary-coolant supply temperature) of +22°F with corresponding refrigerant SST of +17°F and one LT secondary-coolant circuit with SCST -18°F with corresponding refrigerant SST of -23°F. The MT circuit also includes the load from the mechanical subcooling of the LT liquid refrigerant and the heat gains from the MT SC circulation pump. The LT circuit includes the heat gains from the LT SC circulation pump. The net refrigeration loads from the fixtures are the same as in the baseline system: MT 856,079 Btu/hr and LT 300,000 Btu/hr.

The MT refrigeration load in Alternative C (DS) is distributed between two suction groups: with SST of +25°F and SST of +20°F to illustrate and assess the benefit of the distributed technology. The net loads are 406,079 Btu/hr and 450,000 Btu/hr respectively. The first suction group also assumes the load from the mechanical subcooling of the LT liquid refrigerant. Similar to the baseline, all LT net loads in Alternative C have been combined into one suction group at SST -20°F. The combined LT net cooling load is 300,000 Btu/hr.

The net loads described above closely match the cooling loads in the supermarket store that was selected as a reference store for this study. The analysis of combined MT and LT suction groups in this study provides an objective tool for energy comparison of the alternative and baseline technologies. A detailed engineering analysis would be required to assess a refrigeration system with multiple suction/supply groups in all technologies.

The heat gains into the refrigerant return lines create additional load. In this study, the heat gains into return lines are accounted for by an estimated vapor superheat between the outlet of the display cases/evaporators and compressor inlet. This superheat is designated as a non-useful superheat in Figure 4. A more detailed investigation of the impact of the heat gains and other parasitic losses in various parts and components of the baseline refrigeration system and alternative technologies would require a more detailed engineering study.

The study analysis was conducted under the assumption that the net refrigeration loads do not vary with the outdoor ambient conditions, since they perform in an air-conditioned indoor environment. In reality, the refrigeration load in the display cases, coolers, and freezers can vary significantly during the year as a result of changes in the indoor dry-bulb temperature and the relative humidity. Capturing these variations and implementing them into the energy analysis requires adequate performance data (mainly refrigerating load and evaporating temperature) from the manufacturers of refrigerated fixtures. These data are generally not available and require a large number of additional tests from the original equipment manufacturer. Obtaining each data point by the currently used test method (ANSI/ASHRAE Standard 72 Method of Testing Commercial Refrigerators and Freezers) is time-consuming. With the variety of refrigerated fixtures and the pace of developing new models and improving the existing ones, it is unrealistic to expect data on refrigerating loads and evaporating temperatures as a function of the dry and wet bulb indoor temperatures to become available in the near future. Yet, such data would contribute substantially to improving the design and operation of efficient supermarket refrigeration systems and to finding optimum design conditions minimizing the energy consumption of the refrigeration and air-conditioning systems.

The load from the mechanical subcooling varies with outdoor ambient conditions. This variation is expressed through the enthalpy of the liquid refrigerant entering the mechanical subcooler and was analyzed separately for each bin temperature and number of hours in the bin. Thus, the cooling load from the mechanical subcooler was determined from the following equation:

$$Q_{MSC} = m_{LTR} \left(h_{cd,out} - h_{MSCout}\right), \qquad (4)$$

where:

Q_{MSC} = Cooling capacity of the mechanical subcooler, Btu/hr

m_{LTR} = LT refrigerant mass flow rate, lb/hr

$h_{cd,out}$ = Specific enthalpy of the refrigerant at the condenser outlet, Btu/lb

h_{MSCout} = Specific enthalpy of the refrigerant at the mechanical sub-cooler outlet, Btu/lb

The LT refrigerant mass flow rate was calculated from the LT net cooling capacity and the specific refrigeration capacity of the refrigerant at the outlet and inlet of the refrigerated fixtures/evaporators applying the following equation:

$$m_{LTR} = \frac{Q_{LT}}{h_{LTEvapOut} - h_{LTMSCout}}, \qquad (5)$$

where:

m_{LTR} = LT refrigerant mass flow rate, lb/hr

Q_{LT} = Cooling load in the LT system, Btu/hr

$h_{LTEvapOut}$ = Specific enthalpy of the refrigerant exiting the refrigerated fixture, Btu/lb.

$h_{LTMSCout}$ = Specific enthalpy of the LT refrigerant leaving the mechanical sub-cooler, Btu/lb.

An assumption was made that there will be no heat gains or losses in the liquid refrigerant lines between the mechanical sub-cooler and refrigerated fixtures. (A detailed engineering study could account for these heat gains or losses.)

The specific enthalpy of the refrigerant exiting a refrigerated fixture is determined at the evaporating pressure and the temperature of the superheat vapor at the outlet of the LT fixture, specified for each technology in Table 2.

The specific enthalpy of the refrigerant leaving the mechanical sub-cooler is determined at the liquid refrigerant sub-cooled temperature for each technology (see Table 2).

The heat gains in the secondary-coolant supply and return lines and the heat gain from the circulation pumps are additional loads added to the loads from the refrigerated fixtures. In this study, the heat gains in the secondary-coolant supply and return lines have not been accounted for but could be the subject of a future detailed engineering study. The heat load from the circulation pumps is taken into consideration by adding to the particular secondary-coolant circuit load, MT or LT, an estimated 90% of the power input into the electric motor of the circulation pumps.

5.4. Bin energy consumption

The energy consumption (in kWh) in each bin was determined by multiplying the system power per bin (in kW) times the number of hours in that bin.

5.5. Annual energy consumption

The total annual energy consumption is the total of the energy consumption in all bins.

6. RESULTS: BIN AND ANNUAL ENERGY CONSUMPTION OF THE BASELINE AND ALTERATIVE TECHNOLOGIES

The results from the theoretical analysis are summarized in Table 8 and as a bar graph in Figure 5. Table 8 shows the annual energy consumption of the baseline and alternatives organized by geographic location, technology, and system temperature level. Figure 5 shows the patterns in the annual energy consumption by refrigeration technology and geographic location. Appendix C provides a more detailed set of results tables. Presented for the baseline and each alternative within each geographical location, these tables illustrate how annual energy consumption was calculated based on the power input and weather bin data.

As shown in Table 8, the results indicate that both secondary-coolant and distributed systems are viable alternatives to the current centralized DX systems. All systems analyzed in the three regions are within a few percent of the baseline in terms of energy use. Many other factors regarding the actual operation of the systems are likely to lead to at least this amount of fluctuation in energy use.

Boulder. In areas with a large number of hours with low ambient temperatures, secondary-coolant systems have the lowest annual energy consumption when liquid refrigerant is subcooled to 30°F. In Boulder, the annual energy consumption for Alternative B (SC) was 4.1% lower than the DX baseline for systems with liquid refrigerants subcooled to 30°F, 3.2% lower for systems subcooled to 40°F, and 2.4% lower for systems subcooled to 50°F. Distributed systems show similar results as the secondary-coolant systems, with energy consumption for Alternative C (DS) 3.3% lower than baseline energy consumption. As shown in the table, for Alternative A (MTS), which has a low-temperature DX and medium-temperature secondary-coolant refrigeration system, annual energy consumption is 0.9% lower than Baseline (DX) system energy use.

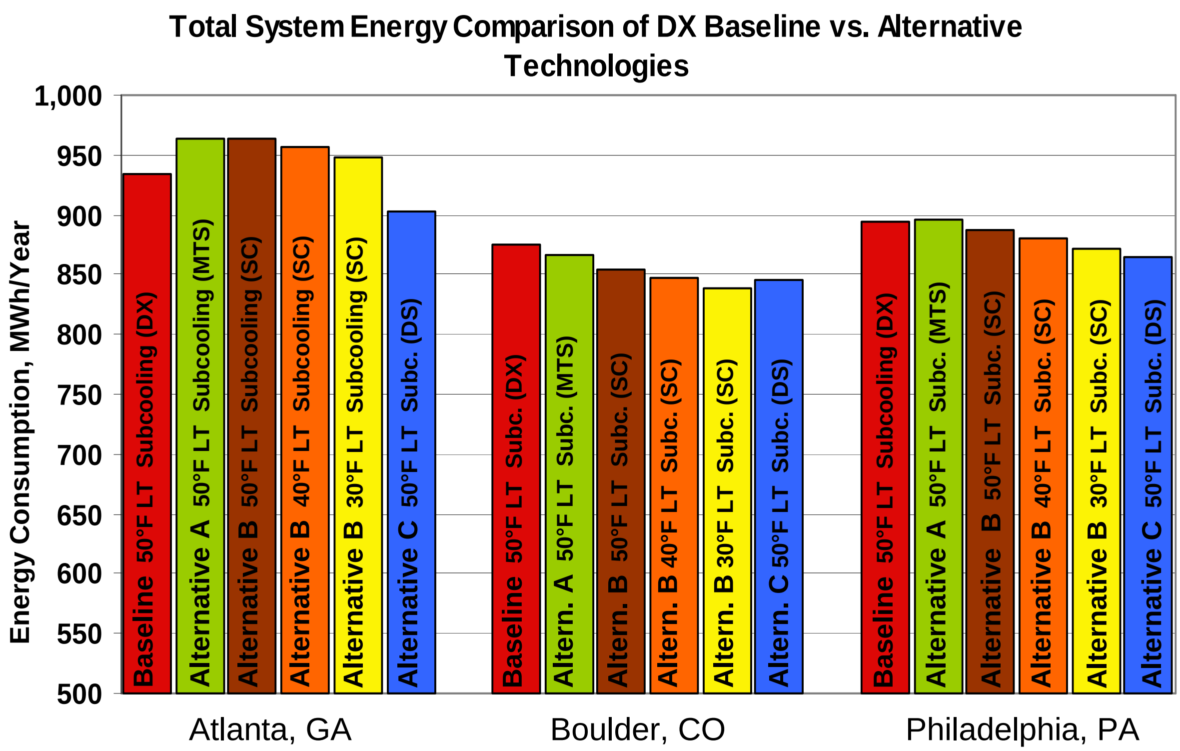
Philadelphia. In Philadelphia, annual energy consumption was lowest for Alternative C (DS), at 3.3% less than Baseline (DX) energy consumption. The Alternative B secondary-coolant systems also resulted in reduced energy consumption, ranging from 0.8% to 2.5% lower than the baseline system. In Philadelphia, annual energy consumption for Alternative A (MTS) is 0.2% higher than for the baseline system.

Atlanta. In areas with fewer hours of low temperatures, distributed systems show the lowest annual energy consumption. In Atlanta, Alternative C (DS) consumed 3.4% less energy than the DX baseline system, while the secondary-coolant systems consumed between 1.5% and 3.2% more than the baseline. Annual energy consumption for Alternative A (MTS) is 3.1% higher than for the baseline system.

Table 8: Annual energy consumption of supermarket refrigeration technologies at three geographic locations				
System Type	LT System Energy kWh/Year	MT System Energy kWh/Year	Combined Total System Energy kWh/Year	Combined Total Compared to DX %
Atlanta, GA Results				
Baseline DX with 50°F(10°C) LT Liquid	339,627	594,186	933,813	-
Alternative A MTSC/LTDX with 50°F(10°C) LT Liquid	339,627	623,416	963,043	3.1%
Alternative B SC with 50°F(10°C) LT Liquid	339,838	624,258	964,096	3.2%
Alternative B SC with 40°F(4°C) LT Liquid	323,473	632,425	955,899	2.4%
Alternative B SC with 30°F(-1°C) LT Liquid	307,964	639,967	947,931	1.5%
Alternative C DS with 50°F(10°C) LT Liquid	339,627	562,871	902,499	-3.4%
Boulder, CO Results				
Baseline DX with 50°F(10°C) LT Liquid	330,651	544,427	875,078	-
Alternative A MTSC/LTDX with 50°F(10°C) LT Liquid	330,651	536,371	867,022	-0.9%
Alternative B SC with 50°F(10°C) LT Liquid	316,919	536,903	853,822	-2.4%
Alternative B SC with 40°F(4°C) LT Liquid	303,532	543,368	846,900	-3.2%
Alternative B SC with 30°F(-1°C) LT Liquid	289,049	550,253	839,302	-4.1%
Alternative C DS with 50°F(10°C) LT Liquid	330,651	515,452	846,102	-3.3%
Philadelphia, PA Results				
Baseline DX with 50°F(10°C) LT Liquid	333,877	561,044	894,921	-
Alternative A MTSC/LTDX with 50°F(10°C) LT Liquid	333,877	562,628	896,505	0.2%
Alternative B SC with 50°F(10°C) LT Liquid	324,243	563,253	887,496	-0.8%
Alternative B SC with 40°F(4°C) LT Liquid	309,817	570,318	880,135	-1.7%
Alternative B SC with 30°F(-1°C) LT Liquid	294,959	577,396	872,355	-2.5%
Alternative C DS with 50°F(10°C) LT Liquid	333,877	531,317	865,194	-3.3%

LEGEND:
Baseline DX with 50°F (10°C) Liquid: Baseline direct-expansion (DX) refrigeration system with min condensing temperature 70°F in both medium-temperature (MT) and low-temperature (LT) circuits, and LT liquid refrigerant subcooled to 50°F by mechanical subcooling in MT circuit.
Refrigeration load from refrigerated fixtures: MT with saturation suction temperature (SST) 20°F 856,079 BTU/h, LT with SST -20°F 300,000 BTU/h. Alternative A MTSC/LTDX with 50°F (10°C) LT liquid: Refrigeration system with MT secondary-coolant (SC) circuit with 17°F SST and 22°F secondary-coolant supply temperature (SCST) and DX LT circuit with SST -20°F. Min condensing temperature 70°F in LT DX and 50°F in MT SC circuits. LT liquid refrigerant subcooled to 50°F by mechanical subcooling in MT circuit. Refrigeration load from refrigerated fixtures: MT 856,079 BTU/h, LT 300,000 BTU/h.
Alternative B SC with 50°F (10°C) LT Liquid: SC refrigeration system with min condensing temperature 50°F in MT and 40°F in LT circuits. LT liquid refrigerant subcooled to 50°F (when condensing temperature is above 50°F) by mechanical subcooling in MT circuit. SCST 22°F in MT and -18°F in LT circuits. Refrigeration load from refrigerated fixtures: MT 856,079 BTU/h, LT 300,000 BTU/h.
Alternative B SC with 40°F (10°C) LT Liquid: SC refrigeration system with min condensing temperature 50°F in MT and 40°F in LT circuits. LT liquid refrigerant subcooled to 40°F (when condensing temperature is above 40°F) by mechanical subcooling in MT circuit. SCST 22°F in MT and -18°F in LT circuits. Refrigeration load from refrigerated fixtures: MT 856,079 BTU/h, LT 300,000 BTU/h.
Alternative B SC with 30°F (10°C) LT Liquid: SC refrigeration system with min condensing temperature 50°F in MT and 40°F in LT circuits. LT liquid refrigerant subcooled to 30°F by mechanical subcooling in MT circuit. SCST 22°F in MT and -18°F in LT circuits.
Refrigeration load from refrigerated fixtures: MT 856,079 BTU/h, LT 300,000 BTU/h. Alternative C DS with 50°F (10°C): Distributed DX refrigeration systems with min condensing temperature 70°F in both MT and LT. Refrigeration load from refrigerated fixtures: MT with SST 25°F 450,000 BTU/h, MT with SST 20°F 406,079 BTU/h, LT with SST -20°F 300,000 BTU/h. LT liquid refrigerant subcooled to 50°F by mechanical subcooling in the adjacent MT circuit with SST 25°F.

Figure 5: Annual energy consumption of supermarket refrigeration technologies in three geographic locations



(Note that the scale for energy consumption starts at 500 MWh/year, not zero.)

7. ANALYSIS OF THE RESULTS

The focus of this theoretical study primarily involved an energy analysis of alternative supermarket refrigeration technologies as compared to a baseline DX technology. Some previous studies of the energy efficiency of alternative supermarket refrigeration systems, particularly secondary-coolant technologies, have shown secondary-coolant refrigeration systems to be associated with up to 30% higher annual energy consumption compared to DX systems. However, these studies have involved a limited number of secondary coolants with poor thermo-physical properties, a lack of a good design practice, and in some instances, design errors. For this reason, this study represents an attempt to conduct an analysis based on the most advanced design practices and using secondary coolants with improved performance properties.

Annual energy consumption is a reliable indicator of the design and operational efficiency of a supermarket refrigeration system. When comparing the three alternative technologies with the baseline, it becomes apparent that no one technology will be superior in all geographic locations in terms of energy efficiency.

In climates with fewer hours of low annual ambient temperatures, such as Atlanta, GA, Alternative C (DS) distributed systems have the lowest annual energy consumption by 3.4%. In comparison, Alternative B (SC) systems have between 1.5% and 3.2% higher energy consumption than the baseline. The design features of the distributed systems lead to the conclusion that the two temperature levels in the MT load (+20°F and +25°F) have contributed to the high efficiency of Alternative C. Because of the prevailing size of the MT load, which is approximately three times the size of the LT load, any efficiency-improving measure in MT will have a noticeable impact on the annual energy consumption of the whole system. The same efficiency-enhancing effect can be achieved in the other technologies by using multiple suction groups, which are analogous to the multiple temperature levels in Alternative C.

A second conclusion is that multiple suction groups in any technology have the most significant impact in geographic locations with warmer climates. In such climates, the special features of the secondary-coolant technologies, such as the lower limit of the floating condensing temperature and the deeper mechanical subcooling of the LT liquid refrigerant cannot make up for the benefits from the multiple MT suction groups because in milder climates these special features cannot materialize their full potential. In warmer climates, the use of a complete secondary-coolant technology (Alternative B) can be counterproductive from an energy point of view. This situation can be exacerbated when a secondary-coolant technology is applied only to the MT system (e.g., Alternative A), preventing the implementation of multiple suction groups or distributed systems.

The benefits from the special features of the alternative technologies have a different relative impact in geographic locations with a larger number of hours of low ambient temperatures. In Boulder, CO, the version of the secondary-coolant technology (Alternative B - SC) with a level of liquid refrigerant subcooling of 30°F has the lowest energy consumption. Since subcooling to 30°F has become the norm in the design practice of at least one major original equipment manufacturer, the secondary-coolant technology can be expected to have low energy consumption in geographic locations with climates similar to Boulder, Co.. Apparently, the lower annual energy consumption in the secondary-coolant technology in climates with a larger number of low ambient temperatures results from the lower limit of floating condensing pressure/temperature and lower subcooling. Because of the larger number of hours with low ambient temperatures, both LT and MT compressors operate longer at low discharge pressures and consume less energy. Some supermarket industry experts have suggested that the baseline DX systems can be operated at the same low limit of the condensing pressures as Alternative B (SC), and energy savings could be expected if DX systems were operated in this manner. However, secondary-coolant systems are especially suitable for low condensing pressures and low liquid subcooling because of the short liquid refrigerant lines upstream from the expansion valves.

The energy benefits of Alternative C (DS) in low-ambient climates are similar to the benefits from Alternative B (SC), due to the multiple temperature levels or multiple suction groups in the DS system. Thus, the decision of which system to select may depend on consideration of other issues, such as ease and cost of operation and maintenance, the supermarket's established practices and preferences, and installed cost.

In the climate conditions of Philadelphia, PA, the only alternative technology that did not use less energy than the baseline system was Alternative A. The comparable annual energy consumption between Alternative B with 30°F subcooled liquid and Alternative C indicates that the decision of which technology to choose will depend on additional considerations.

The interpretation of the results becomes even more evident from the number of hours the MT and LT compressors operate at their minimum SDT at the three geographic locations (see Table 9). The MT compressors will operate at their minimum SDT (50°F) 2.3 times longer in Boulder, CO and 2.2 times longer in Philadelphia, PA as compared to Atlanta, GA. The LT compressors will operate at their minimum SDT (40°F) 4.0 times longer in Boulder, CO and 2.8 times longer in Philadelphia, PA as compared to Atlanta, GA. Therefore, technologies that can operate the compressors at the lowest SDT are expected to have a prevailing energy efficiency benefit in geographic areas with climates similar to or colder than Boulder, CO and Philadelphia, PA. Their energy efficiency advantage is expected to be negligible or non-existent in geographic locations with climates similar to or warmer than Atlanta, GA.

To summarize, the results of the analysis of alternative supermarket refrigeration technologies at the three geographic locations indicate that two of the three analyzed alternative technologies have lower energy requirements than the baseline DX technology in these climates. Multi-temperature distributed systems (Alternative C) are the best choice in climate conditions such as Atlanta, GA or warmer. Secondary-coolant technologies (Alternative B) and distributed systems (Alternative C) provide energy benefits in climate conditions such as Philadelphia, Boulder, or colder. The third technology, Alternative A, only showed energy advantages compared to the baseline DX system in Boulder, CO. In all three locations, Alternative A showed energy penalties of up to a few percent compared to the other alternative technologies.

Table 9: Number of hours of MT and LT compressors at their minimum operating SDT (50°F for MT and 40°F for LT) at the three geographic locations

Ambient Temp. Bin °F	MT Compr. Min. Cond. Temp. °F	Weather Bin Data Atlanta, GA Hours	Weather Bin Data Boulder, CO Hours	Weather Bin Data Philadel-phia, PA Hours	Ambient Temp. Bin °F	LT Compr. Min. Cond. Temp. °F	Weather Bin Data Atlanta, GA Hours	Weather Bin Data Boulder, CO Hours	Weather Bin Data Philadel-phia, PA Hours
35-40	50	560	834	1067	35-40				
30-35	50	323	717	685	30-35				
25-30	50	181	611	442	25-30	40	181	611	442
20-25	50	72	251	248	20-25	40	72	251	248
15-20	50	64	201	184	15-20	40	64	201	184
10-15	50	7	130	40	10-15	40	7	130	40
5-10	50	0	89	0	5-10	40	0	89	0
0-5	50	0	126 [a]	0	0-5	40	0	126[a]	0
Subtotal	(hours):	1207	2833	2666			324	1282	914
Relative to Atlanta	(ratio):	1	2.3	2.2			1	4.0	2.8

[a] The number of hours in this bin is the cumulative number of hours of all temperatures within and below the 5°F-bin, thus integrating the 5°F bin and the next lower-temperature bins. The reason is that all these temperatures affect the performance of the refrigeration system in a similar way and can be processed together.

8. SUMMARY OF CONCLUSIONS AND RECOMMENDATIONS FOR NEXT STEPS

8.1. Summary of conclusions

A general conclusion from this analysis is there are viable alternative supermarket technologies with equal or better energy efficiency to the baseline DX technology. Depending on geographic location, the alternative technology of choice is either a secondary-coolant (Alternative B) or a distributed (Alternative C) system.

In geographic areas with a large number of hours with ambient temperatures below 40°F, the MT compressors will operate at their lowest allowable SDT (50°F) with reduced energy consumption. At ambient temperatures below 30°F, the LT compressors will also operate at their minimum allowable SDT (40°F) with reduced energy consumption. The prolonged operation of both MT and LT systems with low energy consumptions in geographic areas with such ambient conditions will lead to a lower annual energy consumption of the SC refrigeration systems compared to the baseline. The distributed systems show a similar level of energy performance in these cold climates.

In geographic areas with a limited number of hours below 40°F, the secondary coolant systems do not have competitive annual energy consumption. The most advantageous technology for these conditions is Alternative C (distributed refrigeration systems), with as many SST levels as feasible with respect to installed cost.

In geographic areas with ambient conditions falling between the two climate extremes studied here, both alternative technologies, Alternative B (secondary-coolant) and Alternative C (distributed), offer about equal energy efficiency and the choice between these technologies will reflect additional considerations (such as ease and cost of operation and maintenance, the supermarket's established practices and preferences and installed cost).

The conclusions in this study are supported by the practices in some of the major supermarket chains operating in the northeastern and southeastern states. Secondary-coolant systems have become the exclusive technology for a large supermarket chain in the northeast. In addition to the measurable lower annual energy use compared to other alternatives, lower operating costs have been reported, due to low or no maintenance, low or no loss of refrigerant, lower shrinkage, and better product quality.[2]

Another large supermarket chain operating in the southeast achieves favorable annual energy consumption by using multiple suction groups in its DX systems.[3] In this case, distributed systems would reduce the amount of refrigerant charge while maintaining the same energy efficiency. In addition, a large national chain has initiated aggressive cost- and energy-cutting

[2] FMI Energy and Technical Services Conference, Miami, FL, September 2002.

[3] Confidential materials submitted by a supermarket chain.

measures through the deployment of optimized distributed systems. Whenever justified, this chain also deploys secondary systems.[4]

An important conclusion of this study is that no one technology has competitive annual energy consumption in all climate conditions. When planning a new store in a different location, it is important to estimate the annual energy consumption for all technologies under consideration. Some of the other factors to consider are:

- Cost of equipment
- Cost and ease of installation
- Refrigerant and secondary coolant costs
- Cost and ease of operation and maintenance
- Other performance issues (e.g., food quality and shrink).

8.2. Recommendations for next steps

A large number of factors affect the performance and annual energy consumption of a refrigeration system. This theoretical study was performed based on a number of simplifying assumptions in order to provide a preliminary assessment of the feasibility of alternative supermarket refrigeration technologies based on conditions that reflect some of the recent advancements in the alternative and baseline technologies, and to determine if a more detailed engineering study, involving a higher level of effort, is needed to more fully analyze the alternative systems. The results from this study indicate that two of the investigated alternative technologies, Alternative B (secondary-coolant) and Alternative C (distributed), are viable DX alternatives and that a more detailed engineering study could provide data that are more accurate and more closely reflect the real systems and practices, including recent advancements, for both the baseline and the alternative technologies.

An expanded engineering study could include some or all of the following approaches:

- Conduct an engineering-based study that incorporates additional parameters and conditions that more accurately define currently available DX, SC, and DS supermarket refrigeration systems. This theoretical study was based on several simplified assumptions: 1) a limited number of suction groups, temperature levels, and secondary-coolant supply temperatures, 2) omission of power input into condensing fans, 3) omission of heat gains and losses into refrigerant supply and return lines, and 4) omission of heat gains into secondary-coolant supply and return lines. These factors should be included in a detailed engineering study. Table 10 presents a summary of the key parameters and conditions to include in a more detailed engineering study. Appendix B contains a more detailed list of these factors.

- Evaluate the energy impact of the lower limit of floating condensing temperatures in a DX system.

- Consider the seasonal variation in fixture refrigeration loads.

[4] Based on confidential conversations with a supermarket chain.

- Conduct an investigation of a hybrid distributed/secondary-coolant technology, which could prove to be a successful combination of the benefits of the distributed and secondary-coolant systems.

- Include a secondary-coolant technology with a phase-change secondary fluid, in particular CO_2.

- Assess CO_2 as a primary refrigerant in a low-temperature cascade system.

- The dependency of the annual energy consumption on climate conditions justifies the expansion of a study that investigates additional geographic locations. A larger number of analyzed locations can become a building block for a technology map that will provide preliminary information on the suitability of each technology. Supermarkets could use this information during the planning process for building a new supermarket or remodeling an existing one to assess the viability of different technologies.

Proposals for parameters to study in a detailed engineering analysis are provided in Appendix B.

Table 10: Conditions for a detailed engineering analysis

Technology	System Type	Temp. Level	Unit # [a]	Notes	SST[b] °F	Max SDT [c] °F	Min SDT [d] °F	Liquid Temp. °F	Case/Chiller Outlet[g] °F	RGT °F	Cooling load Btu/hr	Power kW
Baseline	DX	LT	1	Subcool Load to Unit 3	-25	110	70	50	-6	Add Heat Gain		
	DX	LT	2	Subcool Load to Unit 3	-14	110	70	50	5	Add Heat Gain		
	DX	MT	3		24	115	70	SDT – 5 [e]	39	Add Heat Gain	Add MSC of units 1 & 2	
	DX	MT	4		20	115	70	SDT - 5	35	Add Heat Gain		
	DX	MT	4		15	115	70	SDT - 5	30	Add Heat Gain		
Alternative A MTSC, LTDX	DX	LT	1	Subcool Load to Unit 3	-25	110	70	50	-6	Add Heat Gain		
	DX	LT	2	Subcool Load to Unit 3	-14	110	70	50	5	Add Heat Gain		
	SC	MT	3		21	115	50	SDT - 5	26	SLHE	Add MSC u's 1&2 & PH	add PP
	SC	MT	4		17	115	50	SDT - 5	22	SLHE	Add PH	add PP
	SC	MT	4		12	115	50	SDT - 5	17	SLHE	Add PH	add PP
Alternative B MTSC, LTSC with Dynalene	SC	LT	1	Subcool Load to Unit 3	-27	110	40	50 (SCT-5)[e],30	-22	SLHE	Add PH	add PP
	SC	LT	2	Subcool Load to Unit 3	-16	110	40	50 (SCT-5)[e],30	-11	SLHE	Add PH	add PP
	SC	MT	3	Same as Alternative A	21	115	50	SDT - 5	26	SLHE	Add MSC u's 1&2 & PH	add PP
	SC	MT	4	Same as Alternative A	17	115	50	SDT - 5	22	SLHE	Add PH	add PP
	SC	MT	4	Same as Alternative A	12	115	50	SDT - 5	17	SLHE	Add PH	add PP
Alternative C DISTRIBUTED	DS	LT	1	Subcool Load to Unit 3	-24	110	50	50,45[f]	-6	Add Heat Gain		
	DS	LT	6b	Subcool Load to Unit 6a	-20	110	50	50,45[f]	-2	Add Heat Gain		
	DS	LT	2	Subcool Load to Unit 3	-13	110	50	50,45[f]	5	Add Heat Gain		
	DS	MT	3		24	115	50	SDT - 5	39	Add Heat Gain	Add MSC of units 1 & 2	
	DS	MT	5		24	115	50	SDT - 5	39	Add Heat Gain		
	DS	MT	4a		20	115	50	SDT - 5	35	Add Heat Gain		
	DS	MT	6a		20	115	50	SDT - 5	35	Add Heat Gain	Add MSC of unit 6b	
	DS	MT	4b		15	115	50	SDT - 5	30	Add Heat Gain		

MSC = Mechanical subcooling, PH = Pump heat, PP = Pump power, SDT = Saturation discharge temperature, RGT = Return gas temperature

[a] See Appendix B (Tables 1-10 and Figures 2-3) for a more detailed illustration of how these systems are configured.

[b] Clarify/confirm with GreenChill Technical Review Committee members the pressure drops for oil return in LTDX and MTDX. While 2°R for both LT&MT DX were assumed in the theoretical analysis, in the current table for a detailed engineering analysis 2°R in MTDX and 3°R in LTDX equivalent pressure drop has been assumed.

[c] Clarify/confirm with GreenChill Technical Review Committee members condenser sizing. While the theoretical analysis was performed for temperature difference 10.0°R for both LT & MT condensers, the current table for a detailed engineering analysis assumes 10°R for LT and 15°R for MT condensers.

[d] Clarify/confirm with GreenChill Technical Review Committee members the minimum SDT for Alternative C. While the theoretical analysis was performed for minimum 70°F, the current table for a detailed engineering analysis assumes 50°F for both MT and LT.

[e] Clarify/confirm with GreenChill Technical Review Committee members the natural subcooling in the condensers. While the theoretical analysis was performed with no subcooling, the current table for a detailed engineering analysis assumes 5°R natural subcooling in both LT and MT condensers.

[f] 50°F out of mechanical subcooler or SCT - 5 = min cond. - 5°R natural SC

[g] Clarify/confirm with GreenChill Technical Review Committee members the superheat out of MT and LT display cases and intermediate heat exchangers (evaporator/chillers). While the theoretical analysis was performed at 15°R superheat out of LT display cases, 5°R out of MT display cases, and 10°R out of both LT and MT intermediate heat exchangers, the current table for a detailed engineering analysis assumes 19°R superheat out of LT DX display cases (3°R in the coil and 16°R in the suction/liquid heat exchanger), 15°R out of the MT DX display cases; 18°R out of LT DS display cases, 15°R out of the MT DS display cases; and 5°R superheat in both LT and MT intermediate heat exchangers (evaporator/chillers).

Theoretical Analysis of Alternative Supermarket Refrigeration Technologies: Technical Review Committee Members

Theoretical Analysis of Alternative Supermarket Refrigeration Technologies:

Technical Review Committee Members

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EPA Supermarket Alternatives Study Report (August 6, 2007)

Phase 1: Proposal for a detailed engineering analysis—description of a baseline store and alternative configurations

EPA Supermarket Alternatives Study

Phase 1: Proposal for a detailed engineering analysis—description of a baseline store and alternative configurations

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August 06, 2007
(Introduction Revised December 19, 2007)

CONTENTS:

1. INTRODUCTION

EPA is developing a voluntary partnership with the supermarket industry to facilitate the transition from ozone-depleting substances to ozone-friendly alternatives. Known as the GreenChill Advanced Refrigeration Partnership, the overall goal of this activity is to promote the adoption of technologies, strategies, and practices that lower emissions of ozone-depleting substances (ODS) and greenhouse gases (GHGs) through both the reduction of refrigerant emissions and the increase of refrigeration systems' energy efficiency. One aspect of the partnership is to conduct technological research and share information that will aid partners in meeting the GreenChill goals.

To meet this goal, EPA commissioned a study to compare the energy efficiency of alternative supermarket refrigeration technologies. The study, *Theoretical Analysis of Alternative Supermarket Refrigeration Technologies*, is based on a theoretical analysis of the energy efficiency of the three most common technologies:

- Direct-expansion (DX) centralized systems,
- Secondary-loop, secondary-coolant, centralized systems, and
- Distributed systems.

The analysis is based primarily upon existing thermodynamic and heat transfer data for refrigerants and secondary-coolant fluids, and performance characteristics from existing laboratory and/or field measurements, manufacturer data, or other available information. The study assesses the following four supermarket refrigeration scenarios:

Baseline:	New supermarket with a DX refrigeration system using an HFC refrigerant (DX).
Alternative A:	New supermarket with a Low Temp DX and Medium Temp glycol secondary loop refrigeration system using an HFC refrigerant (MTS).
Alternative B:	New supermarket with a secondary loop refrigeration system using an HFC refrigerant (SC).
Alternative C:	New supermarket with a distributed refrigeration system using an HFC refrigerant (DS).

This Phase 1 report represents the first phase of the theoretical study. It involved a series of conference calls with the GreenChill Technical Review Committee and EPA to scope out the parameters and methodologies that could be used to estimate annual energy use of various types of supermarket refrigeration systems. The resulting Phase 1 report describes parameters and methodologies that were developed from this process. Upon consideration, it was determined that these parameters were appropriate for conducting a *detailed engineering analysis* of the annual energy use of the baseline and alternative systems, rather than a simplified theoretical study that reflects currently-designed supermarket refrigeration systems. Consequently, the proposed parameters and assumptions were simplified for the theoretical study (for example, the theoretical study is based on fewer suction groups than suggested in this Phase 1 report - see Chapter 4 of the main report).

This Phase 1 report describes the proposed engineering study that was initially developed. This could provide the basis for follow-on work to the existing theoretical study.

2. PARAMETERS AFFECTING THE PERFORMANCE AND ENERGY EFFICIENCY OF A SUPERMARKET REFRIGERATION SYSTEM

The major parameters are:

- Store location
- Indoor data
- Refrigeration loads
- Suction saturation temperature
- Discharge saturation temperature
- Liquid refrigerant subcooling
- Refrigerant vapor superheat
- System design:
  - Type of system
  - Refrigerant selection
  - Secondary coolant selection
  - Components selection
  - Tailoring the system to the refrigerant properties

2.1. Summary of the parameters for energy comparison:

2.1.1. Systems to be investigated:

Baseline:	New supermarket with a DX refrigeration system using an HFC refrigerant (DX).
Alternative A:	New supermarket with a Low Temp DX and Medium Temp glycol secondary loop refrigeration system using an HFC refrigerant (MTS).
Alternative B:	New supermarket with a secondary loop refrigeration system using an HFC refrigerant (SC).
Alternative C:	New supermarket with a distributed refrigeration system using an HFC refrigerant (DS).

2.1.2. Store size, location, and assumptions:

1. Baseline store will be 45,000 sq. ft. with HFC-404A.
2. Locations will be Atlanta, Philadelphia and Boulder, CO.
3. Heat reclaim and defrost method will be excluded from the analysis.
4. Heating and air-conditioning loads, building fire and safety code, store lighting, plug loads and other loads, HVAC annual consumption will be excluded from this study.
5. Note: To avoid the effects of compressor designs, models, cycling, and control strategies, the analysis for the base line and all alternatives will use the energy efficiency ratio (EER) of a representative compressor based on manufacturer's data calculated at the required operating conditions of each alternative technology rather than selecting individual compressors for each alternative technology.
6. Note: to avoid the effect of the compressor design on the technology comparison, the use of scroll compressors with EVI needs to be a subject of another study. Since scroll compressors with EVI can be used in the baseline and in all alternatives, their potential use will equally impact all technologies.

2.1.3. Conditions for the analysis:

The analysis will be performed at the following conditions:

1. Number of the distributed groups for Alternative C:

a. Three saturation suction temperatures for LT (-25, -20, -15°F) and three saturation suction temperatures for MT (+24, +20, and +15°F) located strategically on the roof above the associated line-ups.
b. The 6 suction saturation temperatures will be distributed among 8 groups in 6 locations.

2. Use of suction-line-liquid-line heat exchanger (SLHX) in the display cases. Since both, presence and absence of SLHX, are observed, and the SLHX size and efficiency vary by case manufacturers, the analysis will be performed with superheat out of the display cases equal to the superheat at the coil exit plus additional 5R for MT and additional 10R for LT regardless whether this has resulted from SLHX or through direct heat transfer between the air inside the display case and the suction line between the coil outlets and the case outlet.
3. Use of SLHX on the rack in Alternative B (SC) – optional.
4. Exit superheat from the evaporators and display cases:
   a. Exit superheat for MT evaporators: 8R
   b. Exit superheat for LT evaporators: 6R
   c. Exit superheat for MT display cases: 13R
   d. Exit superheat for LT display cases: 16R

   The superheat increase of 5R in the MT and 10R in the LT display cases are to account for possible use of SLHE or other similar useful superheat between the evaporator and the case outlet.
5. Mechanical subcooling (MS) of the LT liquid refrigerant by the MT refrigerant:
   a. In Baseline, to 50°F
   b. In Alternative A (MTS), to 50°F.
   c. In Alternative B (SC), to 50°F and 30°F.
   d. In Alternative C (DS), to 50°F.
6. Impact of heat gains/losses in the liquid refrigerant lines on subcooling at the display cases and intermediate heat exchanger (IHX):
   a. In Baseline and LT line of Alternative A, the liquid temperature will increase as a result of the heat gains. The increase will be calculated from the diameters, lengths, and insulation of the liquid lines.
   b. In Alternative C (DS), the heat losses will be calculated from the diameters, lengths, and insulation of the liquid lines.
   c. In Alternative B (SC) and MT line of Alternative A, the increase of the liquid refrigerant temperature can be neglected because of the short liquid lines.
7. Heat gains in SC supply and return lines in Alternative B (SC) and MT line of Alternative A (MTS) will be calculated from the SC properties, temperatures, and geometry (diameters, lengths, and insulation) in the MT and LT circuits. The heat gains will be added to the cooling load of the display cases.
8. Temperature difference (TD) between ambient-air temperature and condensing temperature will be used rather than type of condensers (air-cooled, evaporative, or water-cooled), manufacturers and model numbers. Condenser TD:
   a. Medium-temperature system 15R
   b. Low-temperature system 10R
9. Natural subcooling in the condensers: 5R for all systems.
10. Condenser fan control:
    a. In Baseline, float SDT to 70°F for MT and LT condensers.
    b. In Alternative A (MTS), float SDT to 50°F for MT and to 70°F for LT condensers.
    c. In Alternative B (SC), float SDT to 50°F for MT and to 40°F for LT condensers.
    d. In Alternative C (DS), float SDT to 50°F for both MT and LT condensers.
11. Condenser fan consumption: consider it by fan kW/THR for all technologies.

    <u>Note:</u> THR = Total Heat Rejection, BTU/hr

12. MT saturation suction temperature (SST) in Alternative A (MTS) and both MT and LT SST in Alternative B (SC) to be 3R lower than the corresponding SST in the DX suction groups in Baseline.

<u>Note:</u> This results from the assumed 5R temperature difference in the MT and LT intermediate heat exchangers (IHX) and the absence of 2R equivalent pressure drop in the suction lines for oil return.

14. Compressor inlet pressure:
    a. Pressure drop in DX MT suction lines: 2R equivalent
    b. Pressure drop in DX LT suction lines: 2R equivalent
    c. Pressure drop in DS MT suction lines: 2R equivalent
    d. Pressure drop in DS LT suction lines: 2R equivalent
    e. Pressure drop in Alternatives A (MTS) and B (SC) suction lines: equivalent of less than 0.5R lower than the IHX evaporating temperature because of the short return lines and the downstream movement of oil.
15. Compressor inlet temperature:
    a. In Baseline, Alternatives A (LT line), and C (DS), the compressor inlet temperature will be equal to the temperature at the outlet of the display cases plus temperature increase from the heat gains in the return lines. These will be calculated.
    b. In Alternative B, the temperature increase from heat gains will be neglected because of the short lines.
16. Secondary-coolant supply/return temperature difference: 6, 8, and 10R
17. Circulation pumps:
    a. The power input into the SC circulation pumps will be added to the power input of the compressor racks.
    b. The heat from the pumps will be added to the cooling load of the racks.
18. Compressors: in the report the compressor manufacturer and compressor models will be blanked out. The same applies for any information that may be perceived as biased.
19. Refrigerant R-404A will be used in the study.
20. Analysis with both Dynalene and CO2 as a secondary coolant in Alternative B LT loop.
21. In Alternative A (MTS) and Alternative B (SC), glycol will be used in the MT loop.
22. Indoor temperature and relative humidity for the study: 75/55% year around.
23. Insulation – Rubatex with thickness:
    a. MT DX: liquid ½”, suction ¾”
    b. LT DX: liquid ¾”, suction 1”
    c. MT SC supply and return: 1”
    d. LT SC supply and return: 1½”
    e. MT DS: liquid ½”, suction ¾”
    f. LT DS liquid ¾”, suction 1

<u>2.2. Piping diagrams for the baseline and alternative configurations</u>

Schematics of the baseline and alternative configurations are presented in Figure 1.

3. <u>DEFINITION OF THE BASELINE STORE:</u>

3.1. Floor plan and location of the refrigeration loads – Figure 2.
3.2. Load distribution, load components, and piping – Table 1 to 4.

<u>4. DEFINITION OF ALTERNTATIVE A</u>

4.1. Location of the refrigeration loads – same as for the baseline.

4.2. Load distribution, load components, and piping – Load distribution and components are the same as for the baseline. The piping for the LT system is the same as for the baseline. The piping for the MT system will be determined in the second phase of the project.

5. DEFINITION OF ALTERNATIVE B

5.1. Location of the refrigeration loads – same as for the baseline.

5.2. Load distribution, load components, and piping – Load distribution and components are the same as for the baseline. The piping for the LT and MT systems will be determined in the second phase of the project.

6. DEFINITION OF ALTERNATIVE C:

6.1. Location of the loads and units – Figure 3.

6.2. Load distribution and load components - Table 5 to 10. The piping for the LT and MT distributed systems will be determined in the second phase of the project.

7. AMBIENT DRY-BULB TEMPERATURES

Ambient dry-bulb temperatures that will be used in the analysis for Atlanta, Boulder, CO, and Philadelphia are presented in Tables 11 to 13.

Table 1: DX Baseline LT Unit #1, -25°F, Refrigerant HFC-404A

DX Header Loads						Load Line Sizes		
# Loads	**ID**	**Load Description**	**Model**	**MBTU**	**Evap,°F**	**Run**	**Supply**	**Return**
		Unit #1 Circuit Manifold	Remote	129.6	-22	40	7/8"	2-5/8"
1	SP	Spare						
2	54	8'x12'x10' Bakery Frzr, R=7/8		8.0	-18	377	5/8"	1-1/8"
3	52	10'x12'x10' Bakery/Deli Frzr, R=7/8		9.3	-18	322	5/8"	1-1/8"
4	6	12'+(1)E Fz Island Case, R=7/8		10.5	-12	170	5/8"	1-1/8"
5	5	12' Frozen Island Case, R=7/8		7.6	-12	190	5/8"	7/8"
6	4	12'+(1)E Fz Island Case, R=7/8		10.5	-12	202	5/8"	1-1/8"
7	21	10 Drs Ice Cream Cases, R=1		14.1	-20	210	5/8"	1-3/8"
8	20	10 Drs Ice Cream Cases, R=1		14.1	-20	230	5/8"	1-3/8"
9	19	10 Drs Ice Cream Cases, R=1		14.1	-20	270	5/8"	1-3/8"
10	18	5 Drs Ice Cream Cases, R=7/8		7.0	-20	290	5/8"	1-1/8"
11	29	16'x24'x10' IC Freezer, R=1-3/8		20.8	-22	172	5/8"	1-5/8"
12	30	12'x18'x10' Meet Freezer, R=1-1/8		13.6	-18	79	5/8"	1-1/8"
Total Load #1 -25°F			**MBTU**	**129.6**				

DX Compressor Rack #1, Design conditions -25/110°F, Subcooled liquid temp. is 50°F.					
Pos.	Compr. Model	Capacity, MBTU	% Cap.	Rej.MBTU	Rej. %
1		26.5	13%	31.9	120%
2		44.5	21%	53.6	120%
3		54.2	26%	64.9	120%
4		85.3	41%	102.3	120%
	Total Compressors Capacity	**210.5**		252.7	120%
	Rack Capacity to Load Ratio		**162%**		

Table 2: DX Baseline LT Unit #2, -14°F, Refrigerant HFC-404A

DX Header Loads						Load Line Sizes		
# Loads	**ID**	**Load Description**	**Model**	**MBTU**	**Evap,°F**	**Run**	**Supply**	**Return**
		Unit #2 Circuit Manifold	Remote	149.7	-11	40	7/8"	2-5/8"
1	SP	Spare						
2	40	3 Drs Frozen Fd Cases, R=5/8		4.0	-11	120	5/8"	7/8"
3	59	3 Drs Frozen Fd Cases, R=5/8		4.0	-11	322	5/8"	7/8"
4	10	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	230	5/8"	1-3/8"
5	11	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	220	5/8"	1-3/8"
6	9	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	220	5/8"	1-3/8"
7	8	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	170	5/8"	1-3/8"
8	12	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	170	5/8"	1-3/8"
9	13	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	150	5/8"	1-3/8"
10	7	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	150	5/8"	1-3/8"
11	17	5 Drs Frozen Fd Cases, R=7/8"		6.7	-11	245	5/8"	7/8"
12	16	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	235	5/8"	1-3/8"
13	15	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	185	5/8"	1-3/8"
14	14	10 Drs Frozen Fd Cases, R=1-1/8"		13.5	-11	165	5/8"	1-3/8"
Total Load #2 -14°F			**MBTU**	**149.7**				

DX Compressor Rack #2, Design conditions -14/110°F, Subcooled liquid temp. is 50°F.

Pos.	Compr. Model	Capacity, MBTU	% Cap.	Rej.MBTU	Rej. %
1		42.0	18%	47.5	113%
2		50.7	22%	57.4	113%
3		60.0	26%	68.1	114%
4		82.5	35%	93.5	113%
	Total Compressors Capacity	**235.2**		266.5	113%
	Rack Capacity to Load Ratio		**157%**		

Table 3: DX Baseline MT Unit #3, +24°F, Refrigerant HFC-404A

DX Header Loads						Load Line Sizes			Ctrl.Valves
# Loads	**ID**	**Load Description**	**Model**	**MBTU**	**Evap,°F**	**Run**	**Supply**	**Return**	**Suction**
		Unit #3 Circuit Manifold	Remote	468.0	26	50	2-1/8"	3-1/8"	
1	SC1	Rack #1 Subcooling		31.0	35	59	1/2"	1-1/8"	ORIT-PI-311
2	SC2	Rack #2 Subcooling		36.0	35	46	1/2"	1-1/8"	ORIT-PI-413
3	SP	SPARE					None	None	Ball Valve
4	62	AH-4, R=1-1/8"		9.0	44	360	1/2"	7/8"	ORIT-PI-29
5	61	AH-1, AH-2A, AH-2B, R=1-3/8"		30.0	44	250	5/8"	1-1/8"	ORIT-PI-311
6	47	32' Produce Cases, R=1-1/8		46.4	26	163	7/8"	1-5/8"	CDST-9-9
7	48	32' Produce Cases, R=1-1/8		46.4	26	210	7/8"	1-5/8"	CDST-9-9
8	42	Seafood Room Coil, R=1-1/8		36.7	27	91	5/8"	1-3/8"	CDST-9-9
9	44	8' Salad Case, R=5/8"		11.7	26	120	1/2"	7/8"	CDST-9-7
10	22	36' Beverage Cases, R=1-3/8		52.4	27	280	7/8"	1-5/8"	CDST-9-11
11	23	36' Dairy Cases, R=1-3/8		54.0	26	235	7/8"	2-1/8"	CDST-9-11
12	24	24' Dairy Cases, R=1-3/8		36.0	26	230	7/8"	1-5/8"	CDST-9-11
13	25	24' Dairy Cases, R=1-3/8		36.0	26	210	5/8"	1-5/8"	CDST-9-11
14	35	Market Room Coil, R=1-1/8		36.7	27	55	1/2"	1-1/8"	CDST-9-9
15	34	Market Room Coil, R=1-1/8		36.7	27	74	5/8"	1-3/8"	CDST-9-9
Total Load #3 +24°F			**MBTU**	**468.0**					

DX Compr. Rack #3, Design conditions +24/110°F, Subcooled liquid temp. =ambient temp.+10°F

Pos.	Compr. Model	Capacity, MBTU	% Cap.	Rej.MBTU	Rej. %
1		126.6	22%	172.4	136%
2		126.6	22%	172.4	136%
3		140.2	24%	190.9	136%
4		189.9	33%	257.7	136%
	Total Compressors Capacity	**583.3**		793.4	136%
	Rack Capacity to Load Ratio		**125%**		

Table 4: DX Baseline MT Unit #4, +20°F/15°F, Refrigerant HFC-404A

DX Header Loads +20°F

# Loads	ID	Load Description	Model	MBTU	Evap,°F	Load Line Sizes: Run	Supply	Return	Ctrl.Valves: Suction
		Unit #4 Circuit Manifold A	Remote	433.1	22	50	1-5/8"	3-1/8"	
1	SP	• SPARE					None	None	Ball Valve
2	39	• 12'x20'x10' Meat Cooler, R=7/8"		18.9	22	55	5/8"	1-1/8"	CDST-9-7
3	41	• 5'x8'x10' Seafood Cooler, R=5/8"		7.5	22	53	3/8"	7/8"	CDST-9-7
4	46	• 10'x24'x10' Produce Cir, R=7/8		18.1	22	140	1/2"	1-1/8"	CDST-9-7
5		• Loop 49, DR=7/8" & 5/8"		23.0	22	200	1/2"	1-1/8"	ORIT-PI-311
6	49A	•• 8'x12'x10' Deli Cooler		7.5	22	25	3/8"	7/8"	
7	49B	•• 12'x12'x10' Deli Cooler		9.9	22	25	3/8"	7/8"	
8	49C	•• 8'x8'x10' Bakery Cooler		5.6	22	25	3/8"	7/8"	
9	45	• 20' RL Produce Cases, R=1-1/8"		30.0	22	140	5/8"	1-3/8"	CDST-9-9
10	43	• 16' Produce Cases, R=7/8"		16.5	22	110	1/2"	1-1/8"	CDST-9-7
11	55	• 13' Floral Cases, R=7/8"		24.1	22	360	5/8"	1-3/8"	CDST-9-7
12	60	• 8' Deli Case, R=7/8"		14.2	22	240	1/2"	1-1/8"	CDST-9-7
13	56	• 32' Deli Island Cases, R=1-1/8		33.0	22	270	7/8"	1-5/8"	CDST-9-9
14	51	• 20' Deli Cases, R=1/2"		5.8	22	245	3/8"	7/8"	CDST-9-7
15	57	• 24' Deli Island Cases, R=1-1/8		24.7	22	230	5/8"	1-3/8"	CDST-9-9
16	58	• 32' Deli Island Cases, R=1-1/8		33.0	22	160	5/8"	1-5/8"	CDST-9-9
17	28	12'x38'x10' Dairy Cir, R=7/8"		24.2	22	120	5/8"	1-3/8"	CDST-9-7
18	27	36' RL Dairy Cases, R=1-3/8		54.1	22	117	7/8"	2-1/8"	CDST-9-11
19	31	20' Special Meat Case, R=7/8"		31.4	22	93	5/8"	1-3/8"	CDST-9-7
20	32	12'x128'x10' Chicken Cir, R=7/8"		13.9	22	57	5/8"	1-1/8"	CDST-9-7
21	33	24' Special Meat Cases, R=1-1/8"		37.7	22	65	5/8"	1-3/8"	CDST-9-9

Total Load #4 +20°F **MBTU** **433.1**

DX Header Loads +15°F

# Loads	ID	Load Description	Model	MBTU	Evap,°F	Load Line Sizes: Run	Supply	Return	Ctrl.Valves: Suction
		Unit #4 Circuit Manifold B	Remote	77.5	17	50	7/8"	1-5/8"	
1	SP	• SPARE					None	None	Ball Valve
2	38	• 12' Meat Cases, R=5/8"		6.7	17	90	5/8"	7/8"	CDS-9-9
3	37	• 24' Meat Cases, R=1-1/8"		35.4	17	60	5/8"	1-3/8"	CDS-9-9
4	36	• 24' Meat Cases, R=1-1/8"		35.4	17	37	5/8"	1-3/8"	CDS-9-9

Total Load #4 +15°F **MBTU** **77.5**

Table 4: DX Baseline MT Unit #4, +20°F/15°F, Refrigerant HFC-404A (cont'd)

DX Compr. Rack #4, Design conditions +20/110°F, Subcooled liquid temp. =ambient temp.+10°F

Pos.	Compr. Model	Capacity, MBTU	% Cap.	Rej.MBTU	Rej. %
1		94.0	20%	129.4	138%
2		116.3	24%	160.7	138%
3		128.8	27%	178	138%
4		135.9	29%	186.7	137%
	Total Compressors Capacity	**475.0**		654.8	138%
	Rack Capacity to Load Ratio		**110%**		

DX Compr. Rack #4, Design conditions +15/110°F, Subcooled liquid temp. =ambient temp.+10°F

Pos.	Compr. Model	Capacity, MBTU	% Cap.	Rej.MBTU	Rej. %
5		34.6	36%	48.5	140%
6		62.0	64%	86.1	139%
	Total Compressors Capacity	**96.6**		134.6	139%
	Rack Capacity to Load Ratio		**125%**		

Table 5: DS-1, -25°F, Refrigerant HFC-404A

DS-1 Header Loads -25°F						Load Line Sizes		
# Loads	ID	Load Description	Model	MBTU	Evap,°F	Run	Supply	Return
4	6	12'+(1)E Fz Island Case		10.5	-12			
5	5	12' Frozen Island Case		7.6	-12			
6	4	12'+(1)E Fz Island Case		10.5	-12			
7	21	10 Drs Ice Cream Cases		14.1	-20			
8	20	10 Drs Ice Cream Cases		14.1	-20			
9	19	10 Drs Ice Cream Cases		14.1	-20			
10	18	5 Drs Ice Cream Cases		7.0	-20			
11	29	16'x24'x10' IC Freezer		20.8	-22			
12	30	12'x18'x10' Meet Freezer		13.6	-18			
Total Load DS-1, -25°F			**MBTU**	**112.3**				

Table 6: DS-2, -14°F, Refrigerant HFC-404A

DS-2 Header Loads -14°F						Load Line Sizes		
# Loads	ID	Load Description	Model	MBTU	Evap,°F	Run	Supply	Return
4	10	10 Drs Frozen Fd Cases		13.5	-11	294	5/8"	1-3/8"
5	11	10 Drs Frozen Fd Cases		13.5	-11	264	5/8"	1-3/8"
6	9	10 Drs Frozen Fd Cases		13.5	-11	268	5/8"	1-3/8"
7	8	10 Drs Frozen Fd Cases		13.5	-11	237	5/8"	1-3/8"
8	12	10 Drs Frozen Fd Cases		13.5	-11	233	5/8"	1-3/8"
9	13	10 Drs Frozen Fd Cases		13.5	-11	202	5/8"	1-3/8"
10	7	10 Drs Frozen Fd Cases		13.5	-11	205	5/8"	1-3/8"
11	17	5 Drs Frozen Fd Cases		6.7	-11	302	5/8"	7/8"
12	16	10 Drs Frozen Fd Cases		13.5	-11	289	5/8"	1-3/8"
13	15	10 Drs Frozen Fd Cases		13.5	-11	257	5/8"	1-3/8"
14	14	10 Drs Frozen Fd Cases		13.5	-11	226	5/8"	1-3/8"
Total Load DS-2 -14°F			**MBTU**	**141.7**				

Table 7: DS-3, +24°F, Refrigerant HFC-404A

DS-3 Header Loads +24°F						Load Line Sizes			Ctrl.Valves
# Loads	ID	Load Description	Model	MBTU	Evap,°F	Run	Supply	Return	Suction
1	SC1	Rack #1 Subcooling		27.0	35				
2	SC2	Rack #2 Subcooling		34.0	35				
5	61	AH-1, AH-2A, AH-2B		30.0	44				
10	22	36' Beverage Cases		52.4	27				
11	23	36' Dairy Cases		54.0	26				
12	24	24' Dairy Cases		36.0	26				
13	25	24' Dairy Cases		36.0	26				
Total Load DS-3, +24°F			**MBTU**	**269.4**					

Table 8: Distributed system, Loads DS-4a +20°F and DS-4b +15°F, Refrigerant HFC-404A

DS-4a, Header Loads +20°F

# Loads	ID	Load Description	Model	MBTU	Evap,°F	Load Line Sizes Run	Supply	Return	Ctrl.Valves Suction
		Unit #4 Circuit Manifold A	Remote	213.2	22				
1	SP	• SPARE							
2	39	• 12'x20'x10' Meat Cooler, R=7/8"		18.9	22				
16	58	• 32' Deli Island Cases, R=1-1/8		33.0	22				
17	28	12'x38'x10' Dairy Cir, R=7/8"		24.2	22				
18	27	36' RL Dairy Cases, R=1-3/8		54.1	22				
19	31	20' Special Meat Case, R=7/8"		31.4	22				
20	32	12'x128'x10' Chicken Cir, R=7/8"		13.9	22				
21	33	24' Special Meat Cases, R=1-1/8"		37.7	22				
Total Load DS-4a, +20°F			**MBTU**	**213.2**					

DS-4b, Header Loads +15°F

# Loads	ID	Load Description	Model	MBTU	Evap,°F	Load Line Sizes Run	Supply	Return	Ctrl.Valves Suction
		Unit #4 Circuit Manifold B	Remote	77.5	17				
1	SP	• SPARE							
2	38	• 12' Meat Cases, R=5/8"		6.7	17				
3	37	• 24' Meat Cases, R=1-1/8"		35.4	17				
4	36	• 24' Meat Cases, R=1-1/8"		35.4	17				
Total Load DS-4b, +15°F			**MBTU**	**77.5**					

Table 9: DS-5, +24°F, Refrigerant HFC-404A

DS-5 Header Loads +24°F

# Loads	ID	Load Description	Model	MBTU	Evap,°F	Load Line Sizes Run	Supply	Return	Ctrl.Valves Suction
6	47	32' Produce Cases		46.4	26				
7	48	32' Produce Cases		46.4	26				
8	42	Seafood Room Coil		36.7	27				
9	44	8' Salad Case		11.7	26				
14	35	Market Room Coil		36.7	27				
15	34	Market Room Coil		36.7	27				

Total Load DS-5, +24°F	**MBTU**	**214.6**

Table 10: DS-6a +20°F AND DS-6b -20°F, Refrigerant HFC-404A

DS-6a, Header Loads +20°F						**Load Line Sizes**			**Ctrl.Valves**
# Loads	**ID**	**Load Description**	**Model**	**MBTU**	**Evap,°F**	**Run**	**Supply**	**Return**	**Suction**
	SC3	System DS-6b Subcooling		6.1	22				
3	41	• 5'x8'x10' Seafood Cooler		7.5	22				
4	46	• 10'x24'x10' Produce Cir		18.1	22				
6	49A	•• 8'x12'x10' Deli Cooler		7.5	22				
7	49B	•• 12'x12'x10' Deli Cooler		9.9	22				
8	49C	•• 8'x8'x10' Bakery Cooler		5.6	22				
9	45	• 20' RL Produce Cases		30.0	22				
10	43	• 16' Produce Cases		16.5	22				
11	55	• 13' Floral Cases		24.1	22				
12	60	• 8' Deli Case		14.2	22				
13	56	• 32' Deli Island Cases		33.0	22				
14	51	• 20' Deli Cases		5.8	22				
15	57	• 24' Deli Island Cases		24.7	22				
3*	58	• 32' Deli Island Cases		33.0	22				
4*	62	AH-4, R=1-1/8"		9.0	44				

Total Load DS-6a, +20°F	**MBTU**	**245.0**

DS-6b, Header Loads -20°F						**Load Line Sizes**			**Ctrl.Valves**
# Loads	**ID**	**Load Description**	**Model**	**MBTU**	**Evap,°F**	**Run**	**Supply**	**Return**	**Suction**
		Unit #4 Circuit Manifold B	Remote	25.3	17				
1	SP	Spare							
2*	54	8'x12'x10' Bakery Frzr		8.0	-18				
3*	52	10'x12'x10' Bakery/Deli Frzr		9.3	-18				
2*	40	3 Drs Frozen Fd Cases		4.0	-11				
3*	59	3 Drs Frozen Fd Cases		4.0	-11				

Total Load DS-6b, -20°F	**MBTU**	**25.3**

Table 11: Ambient Dry-Bulb Temperature in Atlanta, GA

Mid-pts	DB (F)	Total Hrs	January Hrs	February Hrs	March Hrs	April Hrs	May Hrs	June Hrs	July Hrs	August Hrs	September Hrs	October Hrs	November Hrs	December Hrs
97.5	95 to 100	9							9					
92.5	90 to 95	56						18	27	10		1		
87.5	85 to 90	196				2	10	40	83	56	5			
82.5	80 to 85	758				24	93	154	150	182	141	14		
77.5	75 to 80	768			8	59	117	139	154	142	93	56		
72.5	70 to 75	1314		7	31	82	146	222	251	247	232	84	11	1
67.5	65 to 70	885	4	23	45	93	143	110	51	84	172	108	41	11
62.5	60 to 65	1027	30	73	105	157	156	35	15	23	68	190	104	71
57.5	55 to 60	790	33	78	150	106	69	2	4		9	120	150	69
52.5	50 to 55	673	89	75	131	81	10					78	109	100
47.5	45 to 50	641	118	95	121	53						64	113	77
42.5	40 to 45	436	99	50	72	30						25	55	105
37.5	35 to 40	560	151	84	58	31						4	84	148
32.5	30 to 35	323	102	70	23	2							41	85
27.5	25 to 30	181	68	45									12	56
22.5	20 to 25	72	22	36										14
17.5	15 to 20	64	28	29										7
12.5	10 to 15	7		7										

Table 12: Ambient Dry-Bulb Temperature in Boulder, CO

Mid-pts	DB (F)	Total Hrs	January Hrs	February Hrs	March Hrs	April Hrs	May Hrs	June Hrs	July Hrs	August Hrs	September Hrs	October Hrs	November Hrs	December Hrs
97.5	95 to 100	22						1	18	3				
92.5	90 to 95	96						8	45	33	10			
87.5	85 to 90	115					2	16	37	40	20			
82.5	80 to 85	382			4		26	81	108	83	66	14		
77.5	75 to 80	440			1	17	52	66	114	98	55	37		
72.5	70 to 75	489			14	30	55	81	112	83	59	49	6	
67.5	65 to 70	503	6		10	34	51	71	119	92	73	34	13	
62.5	60 to 65	907	17	11	28	72	87	142	178	194	99	66	13	
57.5	55 to 60	698	16	21	41	77	110	94	11	107	103	72	36	10
52.5	50 to 55	754	44	38	55	98	120	93	2	11	111	92	60	30
47.5	45 to 50	762	63	61	69	77	114	56			79	116	63	64
42.5	40 to 45	633	67	45	80	102	59	11			22	92	92	63
37.5	35 to 40	834	62	118	114	121	52				22	102	143	100
32.5	30 to 35	717	102	115	135	58	16				1	55	114	121
27.5	25 to 30	611	113	124	95	25						15	94	145
22.5	20 to 25	251	65	53	37	9							42	45
17.5	15 to 20	201	58	28	36								44	35
12.5	10 to 15	130	60	18	18									34
7.5	5 to 10	89	27	23	7									32
2.5	0 to 5	83	20	14										49
-2.5	-5 to 0	28	11	3										14
-7.5	-10 to -5	15	13											2

Table 13: Ambient Dry-Bulb Temperature in Philadelphia

Mid-pts	DB (F)	Total Hrs	January Hrs	February Hrs	March Hrs	April Hrs	May Hrs	June Hrs	July Hrs	August Hrs	September Hrs	October Hrs	November Hrs	December Hrs
97.5	95 to 100	3							3					
92.5	90 to 95	52						11	30	11				
87.5	85 to 90	104						34	53	15	2			
82.5	80 to 85	477				6	13	86	184	132	52	4		
77.5	75 to 80	656				2	68	97	198	168	96	17	10	
72.5	70 to 75	907				24	100	161	161	198	200	52	11	
67.5	65 to 70	619				23	96	117	62	137	96	78	10	
62.5	60 to 65	983			21	72	203	179	52	79	165	145	66	1
57.5	55 to 60	625			53	118	123	29	1	4	91	102	96	8
52.5	50 to 55	540		19	66	115	89	5			13	127	93	13
47.5	45 to 50	576	21	22	122	187	36	1			5	80	66	36
42.5	40 to 45	552	86	38	113	97	15					51	75	77
37.5	35 to 40	1067	142	197	196	61	1					65	148	257
32.5	30 to 35	685	119	155	105	15						17	106	168
27.5	25 to 30	442	153	73	56							6	35	119
22.5	20 to 25	248	101	92	12								4	39
17.5	15 to 20	184	98	60										26
12.5	10 to 15	40	24	16										

Figure 1a: Piping schematics of the baseline and alternative systems: Baseline and Alternative C

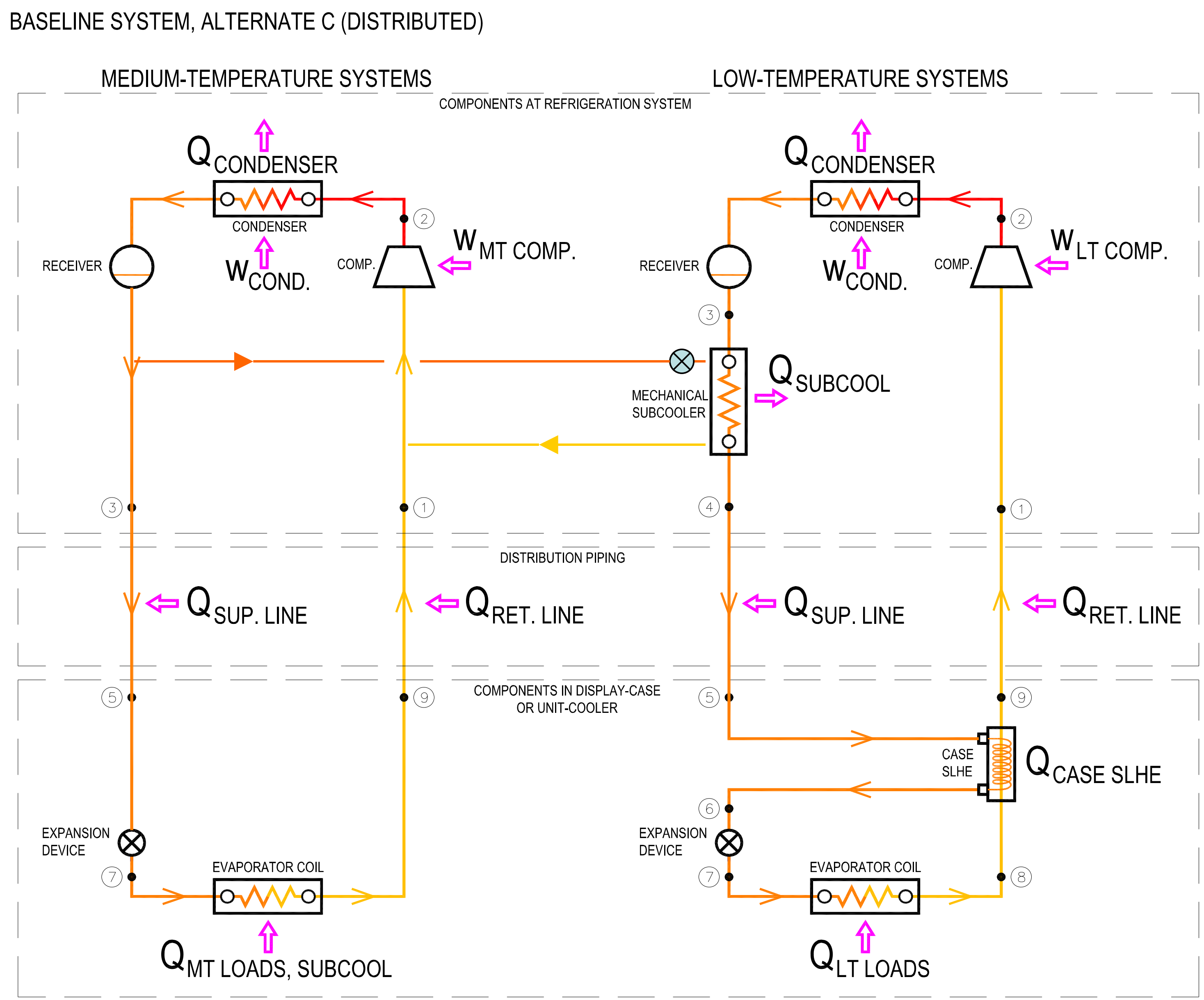


Figure 1b: Piping schematics of the baseline and alternative systems: Alternative A

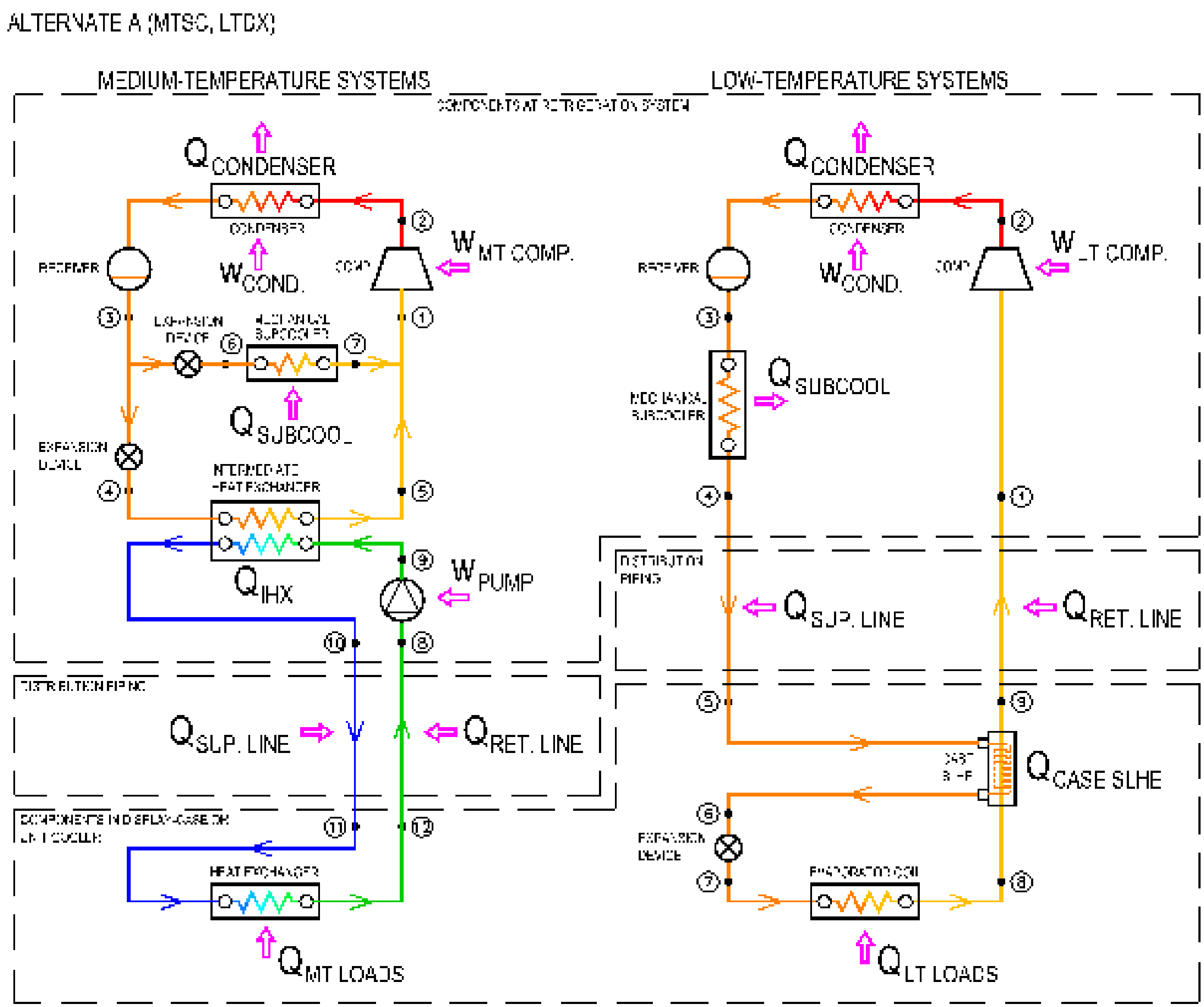


Figure 1c: Piping schematics of the baseline and alternative systems: Alternative B

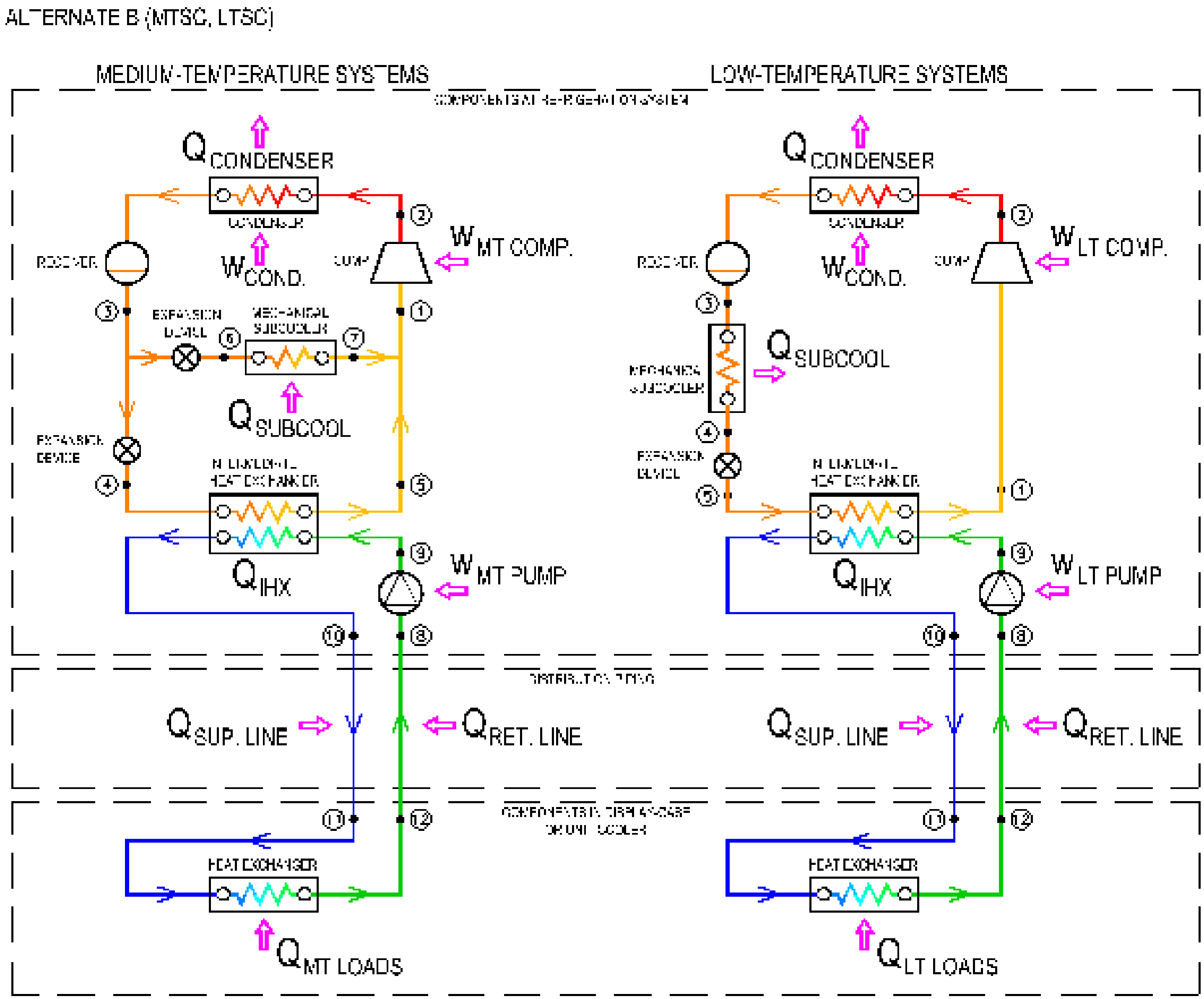
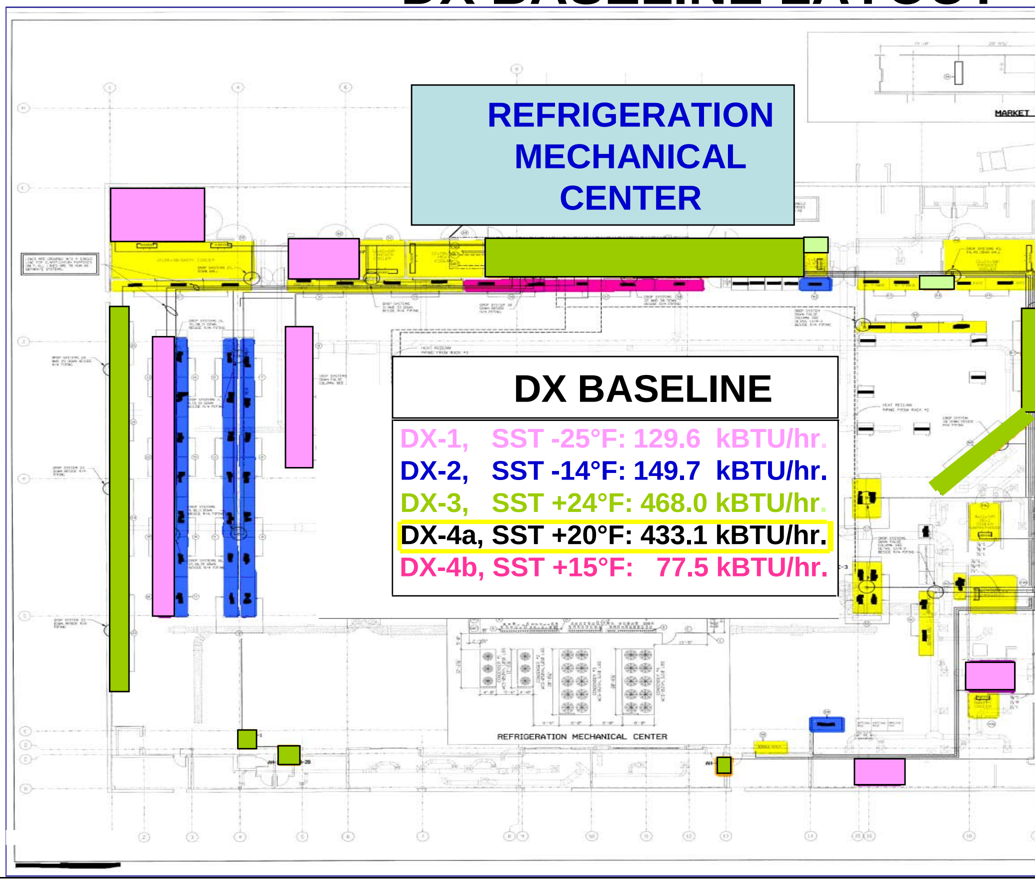


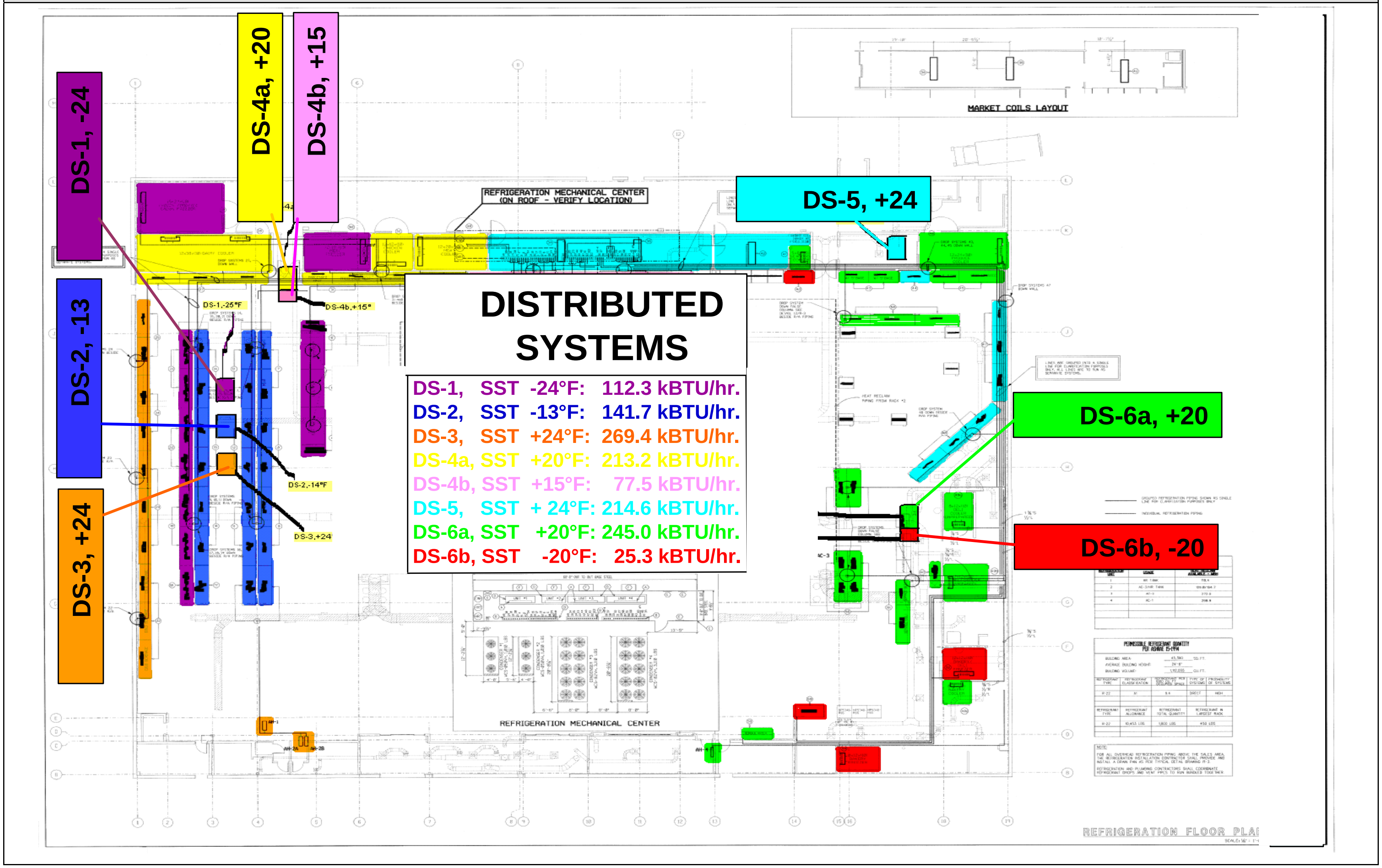
Figure 2: Fixture plan DX baseline

DX BASELINE LAYOUT



Rack #1 Subcooling	31.0	35
Rack #2 Subcooling	36.0	35
AH-4	9.0	44
AH-1, AH-2A, AH-2B	30.0	44
32' Produce Cases	46.4	26
32' Produce Cases	46.4	26
Seafood Room Coil	36.7	27
8' Salad Case	11.7	26
36' Beverage Cases	52.4	27
36' Dairy Cases	54.0	26
24' Dairy Cases	36.0	26
24' Dairy Cases	36.0	26
Market Room Coil	36.7	27
Market Room Coil	36.7	27
8'x12'x10' Bakery Frzr	8.0	-18
10'x12'x10' Bakery/Deli Fr	9.3	-18
12'+(1)E Fz Island Case	10.5	-12
12' Frozen Island Case	7.6	-12
12'+(1)E Fz Island Case	10.5	-12
10 Drs Ice Cream Cases	14.1	-20
10 Drs Ice Cream Cases	14.1	-20
10 Drs Ice Cream Cases	14.1	-20
5 Drs Ice Cream Cases	7.0	-20
16'x24'x10' IC Freezer	20.8	-22
12'x18'x10' Meet Freezer	13.6	-18

Figure 3: Fixture plan Distributed System



Results Tables:
Annual Energy Consumption, Power Input, and Weather Data, by Bin and Geographic Location

Appendix C provides a detailed set of results tables. Presented for each baseline/alternative within each geographical location, these tables present annual energy consumption, power input, and weather data by bin. As illustrated in the tables, annual energy consumption per bin (kWh) is calculated by multiplying power input per bin (kW) times the number of hours at the average ambient temperature for that bin.

Table C.1: Baseline bin and annual energy for Atlanta, GA (DX)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Syst. Power Input kW**	**MT System Power Input kW**	**Total System Power kW**	**Weather Bin Data Atlanta, GA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.04	124.3	171.3	9	423	1118	1,542
90-95	105	45.75	112.5	158.3	56	2562	6301	8,864
85-90	100	44.51	102.24	146.7	196	8724	20039	28,763
80-85	95	43.07	92.31	135.4	758	32644	69972	102,617
75-80	90	42.00	84.09	126.1	768	32255	64578	96,834
70-75	85	40.39	76.64	117.0	1314	53077	100703	153,780
65-70	80	39.14	68.75	107.9	885	34638	60847	95,485
60-65	75	37.85	62.27	100.1	1027	38867	63955	102,822
55-60	70	36.41	56.35	92.8	790	28766	44517	73,282
50-55	65	36.41	55.79	92.2	673	24505	37547	62,052
45-50	60	36.41	55.24	91.6	641	23340	35407	58,747
40-45	55	36.41	54.69	91.1	436	15876	23845	39,720
35-40	50	36.41	54.15	90.6	560	20391	30323	50,714
30-35	50	36.41	54.15	90.6	323	11761	17490	29,251
25-30	50	36.41	54.15	90.6	181	6591	9801	16,391
20-25	50	36.41	54.15	90.6	72	2622	3899	6,520
15-20	50	36.41	54.15	90.6	64	2330	3465	5,796
10-15	50	36.41	54.15	90.6	7	255	379	634
5-10	50	36.41	54.15	90.6	0	0	0	0
0-5	50	36.41	54.15	90.6	0	0	0	0
Annual (hours, kWh)					**8760**	**339,627**	**594,186**	**933,813**

Table C.2 : Baseline Total Bin and Annual Energy for Boulder, CO (DX)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Syst. Power Input kW**	**MT System Power Input kW**	**Total System Power kW**	**Weather Bin Data Boulder, CO h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.04	124.3	171.3	22	1035	2734	3,769
90-95	105	45.75	112.5	158.3	96	4392	10802	15,195
85-90	100	44.51	102.24	146.7	115	5119	11757	16,876
80-85	95	43.07	92.31	135.4	382	16451	35263	51,714
75-80	90	42.00	84.09	126.1	440	18480	36998	55,478
70-75	85	40.39	76.64	117.0	489	19752	37476	57,229
65-70	80	39.14	68.75	107.9	503	19687	34583	54,270
60-65	75	37.85	62.27	100.1	907	34326	56482	90,808
55-60	70	36.41	56.35	92.8	698	25416	39332	64,748
50-55	65	36.41	55.79	92.2	754	27455	42066	69,521
45-50	60	36.41	55.24	91.6	762	27746	42091	69,837
40-45	55	36.41	54.69	91.1	633	23049	34619	57,667
35-40	50	36.41	54.15	90.6	834	30368	45159	75,527
30-35	50	36.41	54.15	90.6	717	26108	38824	64,932
25-30	50	36.41	54.15	90.6	611	22248	33084	55,332
20-25	50	36.41	54.15	90.6	251	9139	13591	22,731
15-20	50	36.41	54.15	90.6	201	7319	10884	18,203
10-15	50	36.41	54.15	90.6	130	4734	7039	11,773
5-10	50	36.41	54.15	90.6	89	3241	4819	8,060
0-5	50	36.41	54.15	90.6	126	4588	6823	11,411
Annual (hours, kWh)					**8760**	**330,651**	**544,427**	**875,078**

Table C.3: Baseline bin and annual energy for Philadelphia, PA (DX)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Syst. Power Input kW**	**MT System Power Input kW**	**Total System Power kW**	**Weather Bin Data Philadelphia, PA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.04	124.3	171.3	3	141	373	514
90-95	105	45.75	112.5	158.3	52	2379	5851	8,230
85-90	100	44.51	102.24	146.7	104	4629	10633	15,262
80-85	95	43.07	92.31	135.4	477	20543	44033	64,575
75-80	90	42.00	84.09	126.1	656	27551	55161	82,712
70-75	85	40.39	76.64	117.0	907	36637	69511	106,148
65-70	80	39.14	68.75	107.9	619	24227	42559	66,786
60-65	75	37.85	62.27	100.1	983	37202	61215	98,417
55-60	70	36.41	56.35	92.8	625	22758	35219	57,976
50-55	65	36.41	55.79	92.2	540	19663	30127	49,789
45-50	60	36.41	55.24	91.6	576	20973	31816	52,790
40-45	55	36.41	54.69	91.1	552	20100	30189	50,288
35-40	50	36.41	54.15	90.6	1067	38852	57776	96,628
30-35	50	36.41	54.15	90.6	685	24942	37091	62,034
25-30	50	36.41	54.15	90.6	442	16094	23933	40,028
20-25	50	36.41	54.15	90.6	248	9030	13429	22,459
15-20	50	36.41	54.15	90.6	184	6700	9963	16,663
10-15	50	36.41	54.15	90.6	40	1456	2166	3,622
5-10	50	36.41	54.15	90.6		0	0	0
0-5	50	36.41	54.15	90.6		0	0	0
Annual (hours, kWh)					**8760**	**333,877**	**561,044**	**894,921**

Table C.4: Alternative A bin and annual energy for Atlanta, GA (MTS)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Syst. Power Input kW**	**MT Syst. Power Input kW**	**Total System Power kW**	**Weather Bin Data Atlanta, GA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.04	132.7	179.7	9	423	1,194	1,618
90-95	105	45.75	121.5	167.2	56	2,562	6,803	9,365
85-90	100	44.51	109.7	154.2	196	8,724	21,508	30,232
80-85	95	43.07	100.8	143.9	758	32,644	76,439	109,083
75-80	90	42.00	91.5	133.5	768	32,255	70,241	102,497
70-75	85	40.39	84.4	124.7	1314	53,077	110,841	163,918
65-70	80	39.14	76.8	115.9	885	34,638	67,943	102,581
60-65	75	37.85	69.9	107.8	1027	38,867	71,796	110,663
55-60	70	36.41	64.1	100.5	790	28,766	50,622	79,388
50-55	65	36.41	58.1	94.5	673	24,505	39,117	63,623
45-50	60	36.41	52.7	89.1	641	23,340	33,798	57,138
40-45	55	36.41	47.8	84.2	436	15,876	20,842	36,718
35-40	50	36.41	43.3	79.7	560	20,391	24,252	44,643
30-35	50	36.41	43.3	79.7	323	11,761	13,988	25,749
25-30	50	36.41	43.3	79.7	181	6,591	7,839	14,429
20-25	50	36.41	43.3	79.7	72	2,622	3,118	5,740
15-20	50	36.41	43.3	79.7	64	2,330	2,772	5,102
10-15	50	36.41	43.3	79.7	7	255	303	558
5-10	50	36.41	43.3	79.7	0	0	0	0
0-5	50	36.41	43.3	79.7	0	0	0	0
Annual (hours, kWh)					**8760**	**339,627**	**623,416**	**963,043**

Table C.5: Alternative A bin and annual energy for Boulder, CO (MTS)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Syst. Power Input kW**	**MT Syst. Power Input kW**	**Total System Power kW**	**Weather Bin Data Boulder, CO h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.04	132.7	179.7	22	1,035	2,919	3,954
90-95	105	45.75	121.5	167.2	96	4,392	11,662	16,054
85-90	100	44.51	109.7	154.2	115	5,119	12,620	17,738
80-85	95	43.07	100.8	143.9	382	16,451	38,522	54,973
75-80	90	42.00	91.5	133.5	440	18,480	40,242	58,722
70-75	85	40.39	84.4	124.7	489	19,752	41,249	61,001
65-70	80	39.14	76.8	115.9	503	19,687	38,616	58,303
60-65	75	37.85	69.9	107.8	907	34,326	63,407	97,733
55-60	70	36.41	64.1	100.5	698	25,416	44,727	70,142
50-55	65	36.41	58.1	94.5	754	27,455	43,825	71,280
45-50	60	36.41	52.7	89.1	762	27,746	40,178	67,924
40-45	55	36.41	47.8	84.2	633	23,049	30,260	53,308
35-40	50	36.41	43.3	79.7	834	30,368	36,118	66,486
30-35	50	36.41	43.3	79.7	717	26,108	31,051	57,158
25-30	50	36.41	43.3	79.7	611	22,248	26,460	48,708
20-25	50	36.41	43.3	79.7	251	9,139	10,870	20,009
15-20	50	36.41	43.3	79.7	201	7,319	8,705	16,024
10-15	50	36.41	43.3	79.7	130	4,734	5,630	10,363
5-10	50	36.41	43.3	79.7	89	3,241	3,854	7,095
0-5	50	36.41	43.3	79.7	126	4,588	5,457	10,045
Annual (hours, kWh)					**8760**	**330,651**	**536,371**	**867,022**

Table C.6: Alternative A bin and annual energy for Philadelphia, PA (MTS)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Syst. Power Input kW**	**MT Syst. Power Input kW**	**Total System Power kW**	**Weather Bin Data Philadelphia, PA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.04	132.7	179.7	3	141	398	539
90-95	105	45.75	121.5	167.2	52	2,379	6,317	8,696
85-90	100	44.51	109.7	154.2	104	4,629	11,413	16,042
80-85	95	43.07	100.8	143.9	477	20,543	48,102	68,645
75-80	90	42.00	91.5	133.5	656	27,551	59,998	87,549
70-75	85	40.39	84.4	124.7	907	36,637	76,509	113,146
65-70	80	39.14	76.8	115.9	619	24,227	47,521	71,748
60-65	75	37.85	69.9	107.8	983	37,202	68,720	105,922
55-60	70	36.41	64.1	100.5	625	22,758	40,049	62,807
50-55	65	36.41	58.1	94.5	540	19,663	31,387	51,049
45-50	60	36.41	52.7	89.1	576	20,973	30,371	51,344
40-45	55	36.41	47.8	84.2	552	20,100	26,387	46,487
35-40	50	36.41	43.3	79.7	1067	38,852	46,208	85,060
30-35	50	36.41	43.3	79.7	685	24,942	29,665	54,607
25-30	50	36.41	43.3	79.7	442	16,094	19,142	35,236
20-25	50	36.41	43.3	79.7	248	9,030	10,740	19,770
15-20	50	36.41	43.3	79.7	184	6,700	7,968	14,668
10-15	50	36.41	43.3	79.7	40	1,456	1,732	3,189
5-10	50	36.41	43.3	79.7	0	0	0	0
0-5	50	36.41	43.3	79.7	0	0	0	0
Annual (hours, kWh)					**8760**	**333,877**	**562,628**	**896,505**

Table C.7: Alternative B bin and annual energy for Atlanta, GA (SC 50°F)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Atlanta, GA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.85	133.1	180.9	9	431	1,198	1,628
90-95	105	46.85	121.8	168.7	56	2,624	6,821	9,445
85-90	100	45.64	110.0	155.6	196	8,945	21,560	30,505
80-85	95	44.22	101.1	145.3	758	33,517	76,605	110,121
75-80	90	42.84	91.6	134.5	768	32,902	70,377	103,279
70-75	85	41.51	84.5	126.0	1314	54,542	111,028	165,569
65-70	80	40.50	76.9	117.4	885	35,846	68,040	103,886
60-65	75	39.24	70.0	109.2	1027	40,295	71,882	112,177
55-60	70	38.04	64.1	102.2	790	30,052	50,670	80,722
50-55	65	36.86	58.2	95.0	673	24,809	39,145	63,954
45-50	60	35.70	52.8	88.5	641	22,886	33,814	56,700
40-45	55	34.29	47.8	82.1	436	14,950	20,847	35,797
35-40	50	33.21	43.3	76.5	560	18,598	24,252	42,850
30-35	50	31.06	43.3	74.4	323	10,031	13,988	24,019
25-30	50	29.05	43.3	72.4	181	5,257	7,839	13,096
20-25	50	29.05	43.3	72.4	72	2,091	3,118	5,209
15-20	50	29.05	43.3	72.4	64	1,859	2,772	4,631
10-15	50	29.05	43.3	72.4	7	203	303	506
5-10	50	29.05	43.3	72.4	0	0	0	0
0-5	50	29.05	43.3	72.4	0	0	0	0
Annual (hours, kWh)					**8760**	**339,838**	**624,258**	**964,096**

Table C.8: Alternative B total bin and annual energy for Boulder, CO (SC 50°F)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Boulder, CO h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.85	133.1	180.9	22	1,053	2,928	3,980
90-95	105	46.85	121.8	168.7	96	4,498	11,693	16,191
85-90	100	45.64	110.0	155.6	115	5,248	12,650	17,899
80-85	95	44.22	101.1	145.3	382	16,891	38,606	55,497
75-80	90	42.84	91.6	134.5	440	18,850	40,320	59,170
70-75	85	41.51	84.5	126.0	489	20,297	41,319	61,616
65-70	80	40.50	76.9	117.4	503	20,373	38,671	59,045
60-65	75	39.24	70.0	109.2	907	35,587	63,483	99,069
55-60	70	38.04	64.1	102.2	698	26,552	44,769	71,322
50-55	65	36.86	58.2	95.0	754	27,795	43,856	71,652
45-50	60	35.70	52.8	88.5	762	27,206	40,197	67,403
40-45	55	34.29	47.8	82.1	633	21,705	30,267	51,972
35-40	50	33.21	43.3	76.5	834	27,698	36,118	63,816
30-35	50	31.06	43.3	74.4	717	22,267	31,051	53,318
25-30	50	29.05	43.3	72.4	611	17,748	26,460	44,208
20-25	50	29.05	43.3	72.4	251	7,291	10,870	18,161
15-20	50	29.05	43.3	72.4	201	5,838	8,705	14,543
10-15	50	29.05	43.3	72.4	130	3,776	5,630	9,406
5-10	50	29.05	43.3	72.4	89	2,585	3,854	6,439
0-5	50	29.05	43.3	72.4	126	3,660	5,457	9,117
Annual (hours, kWh)					**8760**	**316,919**	**536,903**	**853,822**

Table C.9: Alternative B 50°F Total Bin and Annual Energy for Philadelphia, PA (SC 50°F)								
Amb. Temp. °F	Cond. Temp. °F	LT Sys. Power Input kW	MT Sys. Power Input kW	Total System Power kW	Weather Bin Data Philadelphia, PA h	LT System Bin Energy kWh	MT System Bin Energy kWh	Total System Bin Energy kWh
95-100	110	47.85	133.1	180.9	3	144	399	543
90-95	105	46.85	121.8	168.7	52	2,436	6,334	8,770
85-90	100	45.64	110.0	155.6	104	4,746	11,440	16,186
80-85	95	44.22	101.1	145.3	477	21,092	48,206	69,298
75-80	90	42.84	91.6	134.5	656	28,104	60,113	88,217
70-75	85	41.51	84.5	126.0	907	37,648	76,638	114,286
65-70	80	40.50	76.9	117.4	619	25,072	47,590	72,661
60-65	75	39.24	70.0	109.2	983	38,568	68,802	107,371
55-60	70	38.04	64.1	102.2	625	23,775	40,087	63,863
50-55	65	36.86	58.2	95.0	540	19,906	31,409	51,315
45-50	60	35.70	52.8	88.5	576	20,565	30,385	50,950
40-45	55	34.29	47.8	82.1	552	18,928	26,394	45,321
35-40	50	33.21	43.3	76.5	1067	35,436	46,208	81,645
30-35	50	31.06	43.3	74.4	685	21,273	29,665	50,939
25-30	50	29.05	43.3	72.4	442	12,839	19,142	31,980
20-25	50	29.05	43.3	72.4	248	7,204	10,740	17,944
15-20	50	29.05	43.3	72.4	184	5,345	7,968	13,313
10-15	50	29.05	43.3	72.4	40	1,162	1,732	2,894
5-10	50	29.05	43.3	72.4	0	0	0	0
0-5	50	29.05	43.3	72.4	0	0	0	0
Annual (hours, kWh)					**8760**	**324,243**	**563,253**	**887,496**

Table C.10: Alternative B (40°F) Total Bin and Annual Energy for Atlanta, GA (SC 40°F)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Atlanta, GA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	45.34	134.4	179.76	9	408	1,210	1,618
90-95	105	44.44	123.1	167.54	56	2,489	6,894	9,382
85-90	100	43.33	111.2	154.57	196	8,492	21,803	30,295
80-85	95	42.02	102.3	144.27	758	31,848	77,511	109,359
75-80	90	40.74	92.8	133.52	768	31,292	71,249	102,540
70-75	85	39.51	85.6	125.10	1314	51,915	112,467	164,382
65-70	80	38.31	77.9	116.23	885	33,905	68,959	102,864
60-65	75	37.39	71.0	108.37	1027	38,403	72,891	111,294
55-60	70	36.03	65.1	101.11	790	28,466	51,408	79,873
50-55	65	34.94	59.0	93.98	673	23,511	39,734	63,246
45-50	60	33.85	53.6	87.42	641	21,700	34,338	56,038
40-45	55	32.57	48.6	81.15	436	14,202	21,180	35,382
35-40	50	31.52	44.0	75.54	560	17,653	24,649	42,302
30-35	50	30.28	43.7	73.94	323	9,779	14,102	23,881
25-30	50	29.05	43.3	72.35	181	5,257	7,839	13,096
20-25	50	29.05	43.3	72.35	72	2,091	3,118	5,209
15-20	50	29.05	43.3	72.35	64	1,859	2,772	4,631
10-15	50	29.05	43.3	72.35	7	203	303	506
5-10	50	29.05	43.3	72.35	0	0	0	0
0-5	50	29.05	43.3	72.35	0	0	0	0
Annual (hours, kWh)					**8760**	**323,473**	**632,425**	**955,899**

Table C.11: Alternative B (40°F) Total Bin and Annual Energy for Boulder, CO (SC 40°F)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Boulder, CO h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	45.34	134.4	179.76	22	998	2,957	3,955
90-95	105	44.44	123.1	167.54	96	4,266	11,818	16,084
85-90	100	43.33	111.2	154.57	115	4,983	12,793	17,775
80-85	95	42.02	102.3	144.27	382	16,050	39,062	55,112
75-80	90	40.74	92.8	133.52	440	17,927	40,820	58,747
70-75	85	39.51	85.6	125.10	489	19,320	41,854	61,174
65-70	80	38.31	77.9	116.23	503	19,270	39,194	58,464
60-65	75	37.39	71.0	108.37	907	33,916	64,374	98,290
55-60	70	36.03	65.1	101.11	698	25,151	45,421	70,572
50-55	65	34.94	59.0	93.98	754	26,341	44,516	70,858
45-50	60	33.85	53.6	87.42	762	25,796	40,820	66,616
40-45	55	32.57	48.6	81.15	633	20,619	30,750	51,369
35-40	50	31.52	44.0	75.54	834	26,290	36,709	62,999
30-35	50	30.28	43.7	73.94	717	21,708	31,304	53,012
25-30	50	29.05	43.3	72.35	611	17,748	26,460	44,208
20-25	50	29.05	43.3	72.35	251	7,291	10,870	18,161
15-20	50	29.05	43.3	72.35	201	5,838	8,705	14,543
10-15	50	29.05	43.3	72.35	130	3,776	5,630	9,406
5-10	50	29.05	43.3	72.35	89	2,585	3,854	6,439
0-5	50	29.05	43.3	72.35	126	3,660	5,457	9,117
Annual (hours, kWh)					**8760**	**303,532**	**543,368**	**846,900**

Table C.12: Alternative B 40°F Total Bin and Annual Energy for Philadelphia, PA (SC 40°F)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Philadelphia, PA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	45.34	134.4	179.76	3	136	403	539
90-95	105	44.44	123.1	167.54	52	2,311	6,401	8,712
85-90	100	43.33	111.2	154.57	104	4,506	11,569	16,075
80-85	95	42.02	102.3	144.27	477	20,042	48,777	68,818
75-80	90	40.74	92.8	133.52	656	26,728	60,858	87,587
70-75	85	39.51	85.6	125.10	907	35,835	77,631	113,466
65-70	80	38.31	77.9	116.23	619	23,714	48,232	71,946
60-65	75	37.39	71.0	108.37	983	36,758	69,768	106,526
55-60	70	36.03	65.1	101.11	625	22,520	40,671	63,191
50-55	65	34.94	59.0	93.98	540	18,865	31,882	50,747
45-50	60	33.85	53.6	87.42	576	19,499	30,856	50,356
40-45	55	32.57	48.6	81.15	552	17,981	26,815	44,796
35-40	50	31.52	44.0	75.54	1067	33,635	46,965	80,600
30-35	50	30.28	43.7	73.94	685	20,739	29,907	50,646
25-30	50	29.05	43.3	72.35	442	12,839	19,142	31,980
20-25	50	29.05	43.3	72.35	248	7,204	10,740	17,944
15-20	50	29.05	43.3	72.35	184	5,345	7,968	13,313
10-15	50	29.05	43.3	72.35	40	1,162	1,732	2,894
5-10	50	29.05	43.3	72.35		0	0	0
0-5	50	29.05	43.3	72.35		0	0	0
Annual (hours, kWh)					**8760**	**309,817**	**570,318**	**880,135**

Table C.13: Alternative B 30°F Total Bin and Annual Energy for Atlanta, GA (SC 30°F)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Atlanta, GA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	43.41	135.60	179.00	9	391	1,220	1,611
90-95	105	42.27	124.25	166.53	56	2,367	6,958	9,325
85-90	100	41.25	112.33	153.58	196	8,085	22,017	30,102
80-85	95	39.77	103.31	143.08	758	30,144	78,312	108,456
75-80	90	38.60	93.78	132.37	768	29,644	72,020	101,664
70-75	85	37.71	86.56	124.26	1314	49,545	113,739	163,283
65-70	80	36.59	78.84	115.43	885	32,381	69,771	102,152
60-65	75	35.50	71.84	107.34	1027	36,460	73,782	110,242
55-60	70	34.45	65.90	100.35	790	27,217	52,060	79,277
50-55	65	33.42	59.81	93.23	673	22,489	40,255	62,744
45-50	60	32.20	54.29	86.49	641	20,639	34,802	55,441
40-45	55	31.00	49.25	80.25	436	13,514	21,474	34,988
35-40	50	30.01	44.64	74.65	560	16,805	25,000	41,805
30-35	50	28.84	44.30	73.14	323	9,314	14,310	23,625
25-30	50	27.68	43.97	71.65	181	5,010	7,958	12,968
20-25	50	27.68	43.97	71.65	72	1,993	3,166	5,159
15-20	50	27.68	43.97	71.65	64	1,772	2,814	4,586
10-15	50	27.68	43.97	71.65	7	194	308	502
5-10	50	27.68	43.97	71.65	0	0	0	0
0-5	50	27.68	43.97	71.65	0	0	0	0
Annual (hours, kWh)					**8760**	**307,964**	**639,967**	**947,931**

Table C.14: Alternative B 30°F Total Bin and Annual Energy for Boulder, CO (SC 30°F)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Boulder, CO h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	43.41	135.60	179.00	22	955	2,983	3,938
90-95	105	42.27	124.25	166.53	96	4,058	11,928	15,986
85-90	100	41.25	112.33	153.58	115	4,744	12,918	17,662
80-85	95	39.77	103.31	143.08	382	15,191	39,466	54,657
75-80	90	38.60	93.78	132.37	440	16,983	41,261	58,245
70-75	85	37.71	86.56	124.26	489	18,438	42,327	60,765
65-70	80	36.59	78.84	115.43	503	18,404	39,655	58,059
60-65	75	35.50	71.84	107.34	907	32,200	65,161	97,361
55-60	70	34.45	65.90	100.35	698	24,048	45,997	70,045
50-55	65	33.42	59.81	93.23	754	25,196	45,100	70,296
45-50	60	32.20	54.29	86.49	762	24,534	41,372	65,906
40-45	55	31.00	49.25	80.25	633	19,620	31,177	50,797
35-40	50	30.01	44.64	74.65	834	25,028	37,232	62,260
30-35	50	28.84	44.30	73.14	717	20,676	31,766	52,442
25-30	50	27.68	43.97	71.65	611	16,912	26,865	43,777
20-25	50	27.68	43.97	71.65	251	6,948	11,036	17,984
15-20	50	27.68	43.97	71.65	201	5,564	8,838	14,401
10-15	50	27.68	43.97	71.65	130	3,598	5,716	9,314
5-10	50	27.68	43.97	71.65	89	2,464	3,913	6,377
0-5	50	27.68	43.97	71.65	126	3,488	5,540	9,028
Annual (hours, kWh)					**8760**	**289,049**	**550,253**	**839,302**

Table C.15: Alternative B bin and annual energy for Philadelphia, PA (SC 30°F)

Amb. Temp. °F	Cond. Temp. °F	LT Sys. Power Input kW	MT Sys. Power Input kW	Total System Power kW	Weather Bin Data Philadelphia, PA h	LT System Bin Energy kWh	MT System Bin Energy kWh	Total System Bin Energy kWh
95-100	110	43.41	135.60	179.00	3	130	407	537
90-95	105	42.27	124.25	166.53	52	2,198	6,461	8,659
85-90	100	41.25	112.33	153.58	104	4,290	11,683	15,973
80-85	95	39.77	103.31	143.08	477	18,969	49,281	68,250
75-80	90	38.60	93.78	132.37	656	25,321	61,517	86,838
70-75	85	37.71	86.56	124.26	907	34,199	78,509	112,708
65-70	80	36.59	78.84	115.43	619	22,648	48,800	71,449
60-65	75	35.50	71.84	107.34	983	34,898	70,621	105,519
55-60	70	34.45	65.90	100.35	625	21,533	41,187	62,719
50-55	65	33.42	59.81	93.23	540	18,045	32,300	50,345
45-50	60	32.20	54.29	86.49	576	18,546	31,273	49,819
40-45	55	31.00	49.25	80.25	552	17,110	27,187	44,297
35-40	50	30.01	44.64	74.65	1067	32,020	47,634	79,654
30-35	50	28.84	44.30	73.14	685	19,753	30,348	50,102
25-30	50	27.68	43.97	71.65	442	12,235	19,434	31,669
20-25	50	27.68	43.97	71.65	248	6,865	10,904	17,769
15-20	50	27.68	43.97	71.65	184	5,093	8,090	13,183
10-15	50	27.68	43.97	71.65	40	1,107	1,759	2,866
5-10	50	27.68	43.97	71.65		0	0	0
0-5	50	27.68	43.97	71.65		0	0	0
Annual (hours, kWh)					**8760**	**294,959**	**577,396**	**872,355**

Table C.16: Alternative C Total Bin and Annual Energy for Atlanta, GA (DS)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Atlanta, GA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.04	116.22	163.26	9	423	1046	1,469
90-95	105	45.75	107.34	153.09	56	2562	6011	8,573
85-90	100	44.51	96.86	141.37	196	8724	18985	27,709
80-85	95	43.07	87.85	130.92	758	32644	66592	99,237
75-80	90	42.00	79.78	121.77	768	32255	61267	93,523
70-75	85	40.39	72.47	112.86	1314	53077	95228	148,305
65-70	80	39.14	65.34	104.48	885	34638	57830	92,468
60-65	75	37.85	58.98	96.82	1027	38867	60572	99,439
55-60	70	36.41	53.19	89.60	790	28766	42021	70,786
50-55	65	36.41	52.69	89.11	673	24505	35463	59,969
45-50	60	36.41	52.20	88.62	641	23340	33463	56,803
40-45	55	36.41	51.72	88.13	436	15876	22549	38,425
35-40	50	36.41	51.24	87.65	560	20391	28693	49,084
30-35	50	36.41	51.24	87.65	323	11761	16550	28,311
25-30	50	36.41	51.24	87.65	181	6591	9274	15,865
20-25	50	36.41	51.24	87.65	72	2622	3689	6,311
15-20	50	36.41	51.24	87.65	64	2330	3279	5,610
10-15	50	36.41	51.24	87.65	7	255	359	614
5-10	50	36.41	51.24	87.65	0	0	0	0
0-5	50	36.41	51.24	87.65	0	0	0	0
Annual (hours, kWh)					**8760**	**339,627**	**562,871**	**902,499**

Table C.17: Alternative C Total Bin and Annual Energy for Boulder, CO (DS)								
Amb. Temp. °F	Cond. Temp. °F	LT Sys. Power Input kW	MT Sys. Power Input kW	Total System Power kW	Weather Bin Data Boulder, CO h	LT System Bin Energy kWh	MT System Bin Energy kWh	Total System Bin Energy kWh
95-100	110	47.04	116.22	163.26	22	1035	2557	3592
90-95	105	45.75	107.34	153.09	96	4392	10305	14697
85-90	100	44.51	96.86	141.37	115	5119	11139	16258
80-85	95	43.07	87.85	130.92	382	16451	33560	50011
75-80	90	42.00	79.78	121.77	440	18480	35101	53581
70-75	85	40.39	72.47	112.86	489	19752	35439	55191
65-70	80	39.14	65.34	104.48	503	19687	32868	52555
60-65	75	37.85	58.98	96.82	907	34326	53494	87820
55-60	70	36.41	53.19	89.60	698	25416	37127	62543
50-55	65	36.41	52.69	89.11	754	27455	39731	67186
45-50	60	36.41	52.20	88.62	762	27746	39779	67525
40-45	55	36.41	51.72	88.13	633	23049	32738	55787
35-40	50	36.41	51.24	87.65	834	30368	42733	73100
30-35	50	36.41	51.24	87.65	717	26108	36738	62845
25-30	50	36.41	51.24	87.65	611	22248	31306	53554
20-25	50	36.41	51.24	87.65	251	9139	12861	22000
15-20	50	36.41	51.24	87.65	201	7319	10299	17618
10-15	50	36.41	51.24	87.65	130	4734	6661	11395
5-10	50	36.41	51.24	87.65	89	3241	4560	7801
0-5	50	36.41	51.24	87.65	126	4588	6456	11044
Annual (hours, kWh)					**8,760**	**330,651**	**515,452**	**846,102**

Table C.18: Alternative C Total Bin and Annual Energy for Philadelphia, PA (DS)								
Amb. Temp. °F	**Cond. Temp. °F**	**LT Sys. Power Input kW**	**MT Sys. Power Input kW**	**Total System Power kW**	**Weather Bin Data Philadelphia, PA h**	**LT System Bin Energy kWh**	**MT System Bin Energy kWh**	**Total System Bin Energy kWh**
95-100	110	47.04	116.22	163.26	3	141	349	490
90-95	105	45.75	107.34	153.09	52	2379	5582	7,961
85-90	100	44.51	96.86	141.37	104	4629	10074	14,703
80-85	95	43.07	87.85	130.92	477	20543	41906	62,448
75-80	90	42.00	79.78	121.77	656	27551	52333	79,884
70-75	85	40.39	72.47	112.86	907	36637	65732	102,368
65-70	80	39.14	65.34	104.48	619	24227	40448	64,675
60-65	75	37.85	58.98	96.82	983	37202	57977	95,179
55-60	70	36.41	53.19	89.60	625	22758	33244	56,002
50-55	65	36.41	52.69	89.11	540	19663	28455	48,118
45-50	60	36.41	52.20	88.62	576	20973	30069	51,043
40-45	55	36.41	51.72	88.13	552	20100	28549	48,648
35-40	50	36.41	51.24	87.65	1067	38852	54671	93,523
30-35	50	36.41	51.24	87.65	685	24942	35098	60,040
25-30	50	36.41	51.24	87.65	442	16094	22647	38,741
20-25	50	36.41	51.24	87.65	248	9030	12707	21,737
15-20	50	36.41	51.24	87.65	184	6700	9428	16,128
10-15	50	36.41	51.24	87.65	40	1456	2050	3,506
5-10	50	36.41	51.24	87.65		0	0	0
0-5	50	36.41	51.24	87.65		0	0	0
Annual (hours, kWh)					**8760**	**333,877**	**531,317**	**865,194**

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The GreenChill Advanced Refrigeration Partnership is an EPA cooperative alliance with the supermarket industry and other stakeholders to promote advanced technologies, strategies, and practices that reduce refrigerant charges and emissions of **ozone-depleting substances** and **greenhouse gases**.

Working with EPA, GreenChill Partners:

- Transition to non-ozone-depleting refrigerants;
- Reduce refrigerant charges;
- Reduce both ozone-depleting and greenhouse gas refrigerant emissions; and
- Promote supermarkets' adoption of advanced refrigeration technologies.

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