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Technical Paper #4

Natural Refrigerant Applications in North American Supermarkets

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Abstract

Recent achievements with the application of CO_2 secondary and cascade systems in North American supermarkets are presented. Practical information gained through start-up and operation of these systems as well as customer experiences is included. The results of a detailed analysis are presented which focus on impacts of distribution piping, material costs, and heat gain compared to conventional direct expansion systems using HFC refrigerants. An energy and TEWI comparison is made between several system types including CO_2 secondary and cascade, as well as various primary refrigerants. Projects successfully containing the refrigerant charge in the mechanical room renew the possibility of using ammonia on these systems—a discussion of the potential of commercial ammonia systems in the U.S. is added, highlighting needed developments for successful implementation in the future.

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Introduction

Implementation of natural refrigerants in North American commercial refrigeration systems is in its infancy. As of 2008, though new systems use primarily HFCs, 64% of operating refrigeration systems in the US still utilize R-22 as the working fluid. Not subjected to the pressures of HFC bans or high taxation of synthetic refrigerants as in other parts of the world, system choices here are driven primarily by energy consumption and system first cost. A trend has been established towards equipment with lower refrigerant charge and lower leakage rates. This has for the most part been voluntary, driven by corporate initiatives that have been focused on the increasing awareness of the effects of greenhouse gas emissions on the environment and climate change. Government regulations have been slow to respond to these concerns and maximum leakage rates are still not yet mandated for systems operating with HFC refrigerants, though this is expected to change soon.

The use of natural refrigerants in commercial systems in the US has been very limited. One supermarket chain experimented with ammonia in a small system in the mid-1990s. Combined with a medium-temperature secondary coolant system, the ammonia system operated well for years and achieved the goals of lowering refrigerant charge and leakage rates. However, the system was replaced with an HFC system in 2003 as the chain had extreme difficulty in finding qualified technicians to service the ammonia equipment.

Although the use of ammonia in commercial systems has not caught on, efforts have continued to reduce both charge and leak rate of refrigerants. Distributed direct expansion systems, one popular approach, has been applied successfully for years and is available from a variety of manufacturers in various forms. Some moderate decreases in refrigerant charge can be achieved; however leakage rates remain a concern due to the increased number of components, connections, and still large amount of refrigerant-containing piping.

Secondary coolant systems offer an alternative approach which can provide more significant reductions in both charge and leakage rate. Introduced in the U.S. in 1996, medium-temperature secondary coolant systems using propylene glycol are now gaining wider acceptance and represent a significant portion of business with over 500 systems installed, primarily in the U.S. with a small percentage of installations in Canada and Mexico. Today, these systems are primarily applied not only for the benefits of HFC charge reduction, but enhanced product quality and decreased system maintenance. Concerns about increased energy consumption have largely been overcome through proper design practice (Kazachki and Hinde, 2006).

Low-Temperature secondary systems were introduced in the late 1990's using various potassium-based salts. Dozens of systems were installed by multiple manufacturers but some difficulties were experienced with leakage of the secondary fluid and resulting corrosion of surrounding materials. Although the potassium-based fluids exhibit superior heat-transfer performance compared to other single-phase fluids (Melinder, 2007), material compatibility continues to be a concern. These materials are particularly aggressive to zinc and other galvanized coatings which are widely applied in commercial refrigeration equipment. Additionally, the salts are hygroscopic in nature such that spilled fluid will not dry out and must be cleaned thoroughly to prevent continuous corrosion. Use of the potassium salts has slowed dramatically in the last several years with only a handful of new installations. However, recent introduction of cost-effective plastic piping materials and components may allow these systems to become a viable alternative for some applications in the future.

In searching for alternative fluids suitable for low-temperature application, it became clear that CO_2 as a two-phase secondary coolant showed several advantages compared to the single-phase salts. Primarily these were lower pumping power, smaller pipe sizes, excellent heat transfer properties, and good material compatibility with the additional benefit of the low cost of the fluid. The main disadvantage of CO_2 appeared to be the higher operating pressures and availability of components. As development of low-temperature secondary systems with CO_2 has increased, it becomes clear that extension of the application to medium-temperature systems would result in similar benefits, primarily the lower pumping power and smaller line sizes. Additionally, CO_2 can be applied successfully in cascade systems due to its excellent thermodynamic and transport properties. Resulting energy benefits are detailed as part of this study.

CO_2 Supermarket Installations In North America

As of late-2008, at least 11 low-temperature carbon dioxide systems have been installed in the U.S. and Canada. Ten of the systems utilized CO_2 as a low-temperature two-phase secondary fluid with stores ranging in size from small markets to large supercenters and warehouse-style stores, and with system loads ranging from 22 to 160 kW (75 to 550 kBtu/Hr). The ten systems also universally included a primary refrigeration system using an HFC (R-404A or R-507). All but one installation included a medium-temperature secondary coolant system using propylene glycol.

In early-2008, the first CO_2 cascade system was installed and started in a U.S. supermarket. This system utilized CO_2 as low-temperature direct expansion refrigerant with a system load of 70 kW (240kBtu/Hr). The CO_2 was condensed by an R-404a upper-cascade system which also chilled propylene glycol for medium-temperature loads. Questions still remain regarding system cost compared to today's HFC direct expansion systems and low-temperature CO_2 secondary systems, and laboratory investigations will continue to optimize energy consumption and determine how to best commercialize this technology.

Higher pressures and availability of components has not proven to be problematic. Concurrent to the introduction of these systems, significant introduction of components suitable for application with R-410A, increasingly becoming the domestic

A/C refrigerant of choice for replacement of R-22, made the majority of these components readily available.

Successful installation of the systems has relied heavily on comprehensive contractor training programs developed specifically for these CO_2 secondary coolant and cascade applications and will continue to be critical to the implementation of these and other types of CO_2 systems in the future.

The selection of CO_2 grade or purity-level has been carefully considered. Initial systems used CO_2 gas of 99.99% purity (Coleman- or Instrument-Grade), however; some systems have started using 99.5% industrial-grade materials when better grades are not readily available. Charging the CO_2 through liquid filter-driers and a purge of non-condensable gases during start-up seems to make specific requirements unnecessary. CO_2 costs and purity-level availability appear to vary widely throughout the country, however, most installations have been able to obtain the materials at around 1.10 $US/kg (0.50 $US/lb).

CO_2 systems are susceptible to the same mistakes that can plague any field-installed refrigeration system—problems have included contractors not evacuating 100% of the piping network, and charging of impure refrigerant.

Since a CO_2 secondary coolant system is essentially a recirculated liquid system with wet returns (circulation rate greater than 1), balancing the flow between loads had not been a problem. Proper application of the piping network combined with careful coil design has shown that balancing can be done during the design-phase of the project and removes any complicated field-balancing procedures from consideration.

Back-up or auxiliary refrigeration units have been installed on some stores to provide a source of cooling for the CO_2 during extended power outages or maintenance procedures. Opinion remains divided on future application of this cooling depending on customer experiences, reliability of the power supply, and availability of a back-up power-supply.

Energy consumption comparisons have been made between one CO_2 field installation and three similar DX HFC systems nearby. Results have so far indicated a 2–3% average reduction in energy required by the LT CO_2 secondary coolant store during several summer months in 2007 compared with the surrounding stores. This was better than anticipated as a dual-suction group HFC DX system was compared with a single-suction group CO_2 secondary system, and at best, energy consumption was not expected to be favorable during the warmer months, however, a more detailed analysis is required to get an idea of annual performance expectations.

Distribution Piping Effects

In the first implementations of low-temperature systems, as indicated above, the field data indicated performance of the LT CO_2 secondary system was somewhat better than expected. It was thought that the differences could be due to the changes in the distribution piping network, transitioning from a circuited HFC direct expansion system to a loop-type secondary coolant system.

The impact of the distribution piping network on system operation in commercial refrigeration systems has traditionally not been quantified. The majority of the piping networks consist of insulated copper tubing operating at temperatures below indoor ambient temperature and for the most part experiencing unwanted or parasitic heat gain resulting in temperature rise of the working fluid and an increase in the required refrigeration load of the system.

Although this heat load is not generally calculated, it is common practice is to add anywhere from 5 to 15% additional compressor capacity over the required refrigeration load to compensate for this, and in most situations this is sufficient to cover variations of this heat gain into distribution piping. When energy consumption is modeled for these systems, however, and decisions on technology strategy are made based on differences on the order of 5%, this heat gain can become quite significant in the efficiency of the overall system. Of particular interest is the effect of the significantly smaller line sizes associated with using CO_2, both as a secondary coolant and a direct expansion refrigerant. Laboratory testing has also indicated that distribution piping effects can vary significantly depending on system type, operating conditions, pipe sizing, and configuration of the piping network.

Previous work to quantify this heat gain into the piping network (Hinde, Zha, and Lan, 2008) established and verified the methodology and showed that the heat gain for low-temperature refrigeration systems varies significantly according to the system under consideration. Low-temperature system types considered for the analysis included an HFC direct expansion system with two suction groups (existing system), a CO_2 secondary coolant system with a single suction group, and a CO_2 direct expansion cascade system with two suction groups. This analysis was then extended to the medium-temperature systems for this study where systems considered included an HFC direct expansion system with two suction groups, and two different medium-temperature secondary systems using first propylene glycol and then CO_2.

Circuiting and line sizing for the HFC DX Baseline system was based on the existing circuited system configuration which is currently manufactured. Line sizes for the various glycol and CO_2 systems were selected specifically for this analysis based on current practice which dictates a loop-type of installation for both systems as control valves are located at the individual evaporator coils rather than having common valving for multiple coils (evaporator pressure regulators) as is the case with most HFC systems. It was assumed that all lines would be copper piping, of the thickness required for the specific applications—in several instances, the CO_2 piping required Type-K copper pipe. All lines were assumed to run inside the building envelope and in air-conditioned space with a constant ambient temperature of 24°C (75°F).

The analysis was based on the same representative fixture plan which was used in the previous analysis. Figure 1 shows the general layout of the 3600 m^2 (39,000 ft^2) store with the medium-temperature loads shown in green, and the low-temperature loads under investigation shown in blue.

Results of the heat gain analysis for the different low-temperature systems, as detailed in the previous study and shown in Table 1, show that heat gain, as a percentage of the refrigeration load, varies from 6–12% for the LT systems depending on technology applied. Heat gain was lowest for the CO_2-based systems due to a combination of smaller line sizes and a loop piping arrangement compared to the circuited arrangement for the DX Baseline system. Heat gain for the medium-temperature systems was significantly less, as expected, varying from 1–3% of the refrigeration load for the MT DX Baseline system (depending condensing temperature), around 5% for the MT glycol secondary system, and around 2.5% for the MT CO_2 secondary system.

In addition to heat gain, it is also of interest to look at how the different system types compare with each other in terms of installed feet and weight of copper. This gives some indication of potential differences in installation cost between the various system types.

Table 2 summarizes the installed copper length and weight of copper pipe for each of the medium- and low-temperature system types separately, and Table 3 shows installed length and weight for the system combinations under analysis in this study. As expected, significant decreases in installed pipe length are associated with the alternative systems as a consequence of utilizing loop piping versus circuited piping for the baseline system. Even more significant are the savings in copper weight associated with the CO_2 technologies compared with the baseline. Of note is the increased weight of copper pipe compared to the baseline system for the medium-temperature glycol secondary system. Although copper pipe has been used in the past, current installations are increasingly using ABS plastic piping as an alternative material which has shown to reduce both installation time and total installed cost.

System Performance Analysis

Following the heat gain and piping analysis, a TEWI (Total Equivalent Warming Impact) analysis was performed on the system combinations under investigation, each of which is shown schematically in Figure 2. The system combinations examined are:

R-404A Direct Expansion Baseline System on both MT (2-groups) and LT (2-groups) – Figure 2A

R-404A Primary, MT Glycol Secondary (1-group) and LT CO_2 Secondary (1-group) – Figure 2B

R-404A Primary, MT CO_2 Secondary (1group) and LT CO_2 Secondary (1-group) – Figure 2C

R-404A Primary, MT CO_2 Secondary (1-group) and LT CO_2 DX Cascade (2-groups) – Figure 2D

R-410A Primary, MT CO_2 Secondary (1-group) and LT CO_2 DX Cascade (2-groups) – Figure 2D

R-717 Primary, MT CO_2 Secondary (1-group) and LT CO_2 DX Cascade (2-groups) – Figure 2D

A TEWI analysis for refrigeration systems looks a both direct and indirect equivalent CO_2 emissions. The direct impact is due to the leakage of refrigerant into the atmosphere and is a function of the refrigerant charge, the leakage rate, and the Global Warming Potential (GWP) of the refrigerant under investigation. The primary refrigerant charge of the DX HFC system was 4,300 lbs. based on information from actual installations. Charge reduction for the alternative systems was assumed to be conservatively 60% of the DX baseline charge—typical reductions seen in the field range from 60–80% depending on type of condensers and heat reclamation systems being applied. Leakage rates for the DX Baseline system were assumed to be 25% which is the 2008 estimate by the US EPA. Leakage rates for the alternative systems are assumed to be 5% as the refrigerant-containing piping is located in the machine room, consists of nearly all factory-made joints, and leaks are much more easily detected and repaired in comparison to the baseline system.

The indirect impact of the TEWI analysis is a function of the amount of energy consumed by system and the amount of carbon emitted when the energy is produced. The energy consumption for this analysis was based on weather data for Atlanta, Georgia using temperature bin data in 2.8K (5R) increments. Figure 3 illustrates the temperature variations for this climactic region and hourly occurrence per year. Atlanta was chosen as the comparison climate based on results from the previous study (Hinde, Zha, and Lan, 2008) as this represents a hotter climate which does not favor the EEV-based systems and lower condensing temperature associated with cooler climate conditions. CO_2 emissions from the production of energy at the power source were based on the latest US average of 0.623 kg CO_2/kWh (1.37 lbs. CO_2/kWh) (Energy Information Administration, 2002)

Minimum condensing temperatures for the energy analysis were set at 21°C (70°F) for the systems utilizing thermostatic expansion valves (TXVs) and 10°C (50°F) for those using electronic expansion valves (EEVs); the primary HFC portion of the glycol and CO_2 systems would use electronic expansion valves which are well-suited to handle large variations in differential pressure from lower condensing pressures while the HFC DX system would typically use thermostatic expansion valves that would not operate efficiently or would require costly re-adjustment to function well under these conditions. For both the baseline system and the system combinations with LT CO_2 secondary, the low-temperature system was subcooled by the medium-temperature system following industry conventions.

Figure 4 shows the results of the TEWI analysis for all the systems under study and includes a breakdown of the TEWI calculation both the direct (emissions) and indirect (energy) portions. As shown, for the HFC DX Baseline system, the direct impact from refrigerant leakage dominates the TEWI calculation with 89% of the total impact.

Figure 5 shows a detail of the results of the TEWI analysis for the five lower-charge systems such that smaller-scale differences can be better understood. At this point, comparing the lower-charge, lower-leak-rate systems shows that the energy and emissions components of the TEWI calculation are now on the same order of magnitude. As expected, using CO_2 as the secondary fluid on the MT system has clear energy advantages compared to using propylene glycol, decreasing the energy consumption by 8.8%. This is due to the significantly lower pumping energy associated with CO_2 resulting from both lower fluid viscosity of the CO_2 as well as order-of-magnitude lower flow rate compared to the propylene glycol, as well as the superior heat transfer capabilities of CO_2.

Introduction of the CO_2 cascade on the low-temperature system combined with CO_2 secondary on the medium-temperature system allows for a single combined primary system to service the entire store—this provides further reductions in refrigerant charge, though this may meet with some resistance in the marketplace due to a lack of redundancy that many customers require.

Regarding choice of refrigerant, R-410A (GWP = 2100) can be applied on the primary system significantly lowering the direct emission of the TEWI calculation compared with the current standard of using R-404A (GWP = 3900) or R-507 (GWP = 4000). R-410A is well suited to close-coupled applications as the low suction superheat keeps discharge gas temperatures at an acceptable level. Efficiency improvements can also be realized with R-410A, here decreasing energy consumption by 3.2% compared with an otherwise identical R-404A system. Finally, use of ammonia (GWP $\approx$ 0) has the obvious benefit of removing the direct emissions component entirely, while maintaining very similar efficiency levels as the R-410A system.

Future Systems

Further implementation of low-charge, low leak-rate supermarket systems in North America is expected to increase significantly due to heightened concern about climate change and impending regulations dealing with reduction of greenhouse gas emissions. CO_2 will play an increasingly important part in this shift. Low-temperature CO_2 secondary systems went into full commercialization in 2008, and many new installations are under way. Extension of the technology to medium-temperature applications is viable and offers several performance improvements compared to the propylene glycol secondary systems used today, however systems need to be carefully designed and optimized in order to ensure they can be produced in an economically viable manner. Low-temperature CO_2 cascade systems are also expected to expand in application in North America and are well suited to be combined with medium-temperature secondary CO_2 systems with a common primary refrigeration system. Successful implementation of these systems is a critical step which will allow use of alternative primary refrigerants.

Use of R-410A as the primary refrigerant in any low-charge, low-leak rate, close-coupled system is now viable from both a technology and component availability aspect because of increasing use in domestic A/C applications. Several other refrigerants with lower GWP than R-404A and R-507 are expected to increase in popularity as restrictions on emissions are put in place.

CO_2 as a primary refrigerant in a transcritical system has seen much research in recent years and several successful projects have been installed primarily in Europe. Although some questions remain about system efficiency in warmer climates, this represents a viable option if elimination of HFC fluids becomes mandatory.

Ammonia has the same advantages for medium-temperature refrigeration as in water chilling applications (Pearson, A, 2007) due to its thermodynamic and transport properties. Ammonia may be used as primary refrigerant in supermarket refrigeration systems, though several challenges will have to be met to achieve significant acceptance. Charge must be lowered to a level where leakage will not be detectable by surrounding neighbors. To accomplish this, a more modular approach may need to be applied; rather than a single system, multiple smaller systems could be linked in parallel and combined on the secondary side. Systems must be very simple and easy to install, use, and service. Training will also be especially critical to success and must consist of programs designed specifically for technicians who normally service HCFC and HFC commercial equipment as it would be cost-prohibitive to transfer industrially-trained ammonia service technicians to the commercial sector.

Cost and component availability will also be a challenge (Palm, B., 2007). Selection of equipment for small system sizes can be limited and such equipment is generally much more expensive than that designed for use with HFCs. In the end, however, HFC refrigerant regulations will likely be the main incentive for the use of ammonia in supermarkets.

Discussion

A basic understanding of the operation of low-temperature CO_2 secondary and cascade systems has been established through experience with the first pilot installations in the field. A study of the effects of heat gain into the distribution piping network was presented and highlights the ability of CO_2 to lower installation costs compared with more traditional systems through reductions in copper piping length, size, and cost. Successful implementation of these systems forms a foundation on which the application of other natural refrigerants to the primary side of the system can be possible in the future.

A TEWI analysis was presented which indicates both low-temperature CO_2 secondary and direct expansion systems together with medium-temperature CO_2 secondary can be implemented with energy consumption equal to or better than traditional HFC direct expansion systems while dramatically reducing total emissions of greenhouse gases.

Alternative refrigerants such as R-410A and ammonia can result in further reductions of greenhouse gases. Finally, a discussion of the challenges of implementing ammonia in a supermarket system highlights the need for trained commercial technicians as particularly important.

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Table 1. Heat Gain Analysis for Different System Types as Percentage of Refrigeration Load

System Type	Piping Heat Gain As % of Refrigeration Load[NOTE 2]
LT HFC DX Baseline (circuited piping)	12.3%
LT CO_2 Secondary (loop piping)	9.1%
LT CO_2 DX Cascade (loop piping)	6.1
MT HFC DX Baseline (circuited piping)	0.7–3.0[NOTE 1]
MT Glycol Secondary (loop piping)	5.1
MT CO_2 Secondary (loop piping)	2.5

NOTE 1: Heat gain for MT DX Baseline System varies with condensing temperature at higher ambient conditions and stabilizes to 3.0% at ambient temperatures of 12.8°C (55°F) and lower.

NOTE 2: Methodology for calculating Piping Heat Gain is described in detail in reference Hinde, Zha, and Lan 2008.

Table 2. Installed Length and Weight of Copper Piping for Various LT and MT Systems

System Type	Installed Copper Length in meters (in Feet), % change with Baseline	Installed Copper Weight in kg (in lbs.), % change
LT HFC DX Baseline (dual-group)	1690 m (5544 ft)	1147 kg (2530 lbs.)
LT CO_2 Secondary (single-group)	753 m (2472 ft), –55%	309 kg (681 lbs.), –73%
LT CO_2 DX Cascade (dual-group)	809 m (2655 ft), –52%	249 kg (549 lbs.), –78%
MT HFC DX Baseline (dual-group)	2151m (7056 ft)	1518 kg (3346 lbs.)
MT Glycol Secondary (single-group)	942 m (3092 ft), –56%	1707 kg (3764 lbs.), +12%
MT CO_2 Secondary (single-group)	1044 m (3424 ft), –51%	898 kg (1979 lbs.), –41%

Table 3. Installed Length and Weight of Copper Piping for System Combinations

System Type	Installed Copper Length in meters (in Feet), % change with Baseline	Installed Copper Weight in kg (in lbs.), % change
HFC DX Baseline	3841 m (12600 ft)	2666 kg (5877 lbs.)
MT Secondary Glycol, LT Secondary CO_2	1696 m (5564 ft), –56%	2016 kg (4445 lbs.), –24%
MT Secondary CO_2, LT Secondary CO_2	1797 m (5896 ft), –53%	1207 kg (2660 lbs.), –55%
MT Secondary CO_2, LT DX Cascade CO_2	1853 m (6079 ft), –52%	1147 kg (2528 lbs.), –57%

Figure 1: Representative Store Layout

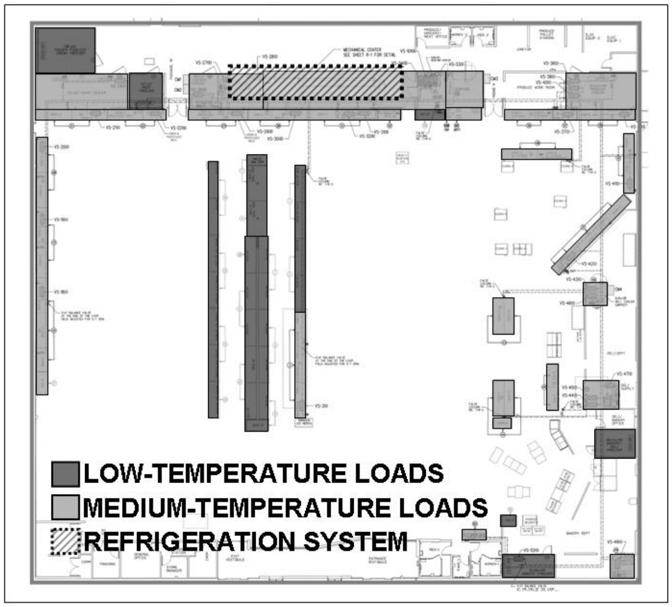


Figure 2: System Configurations Under Investigation

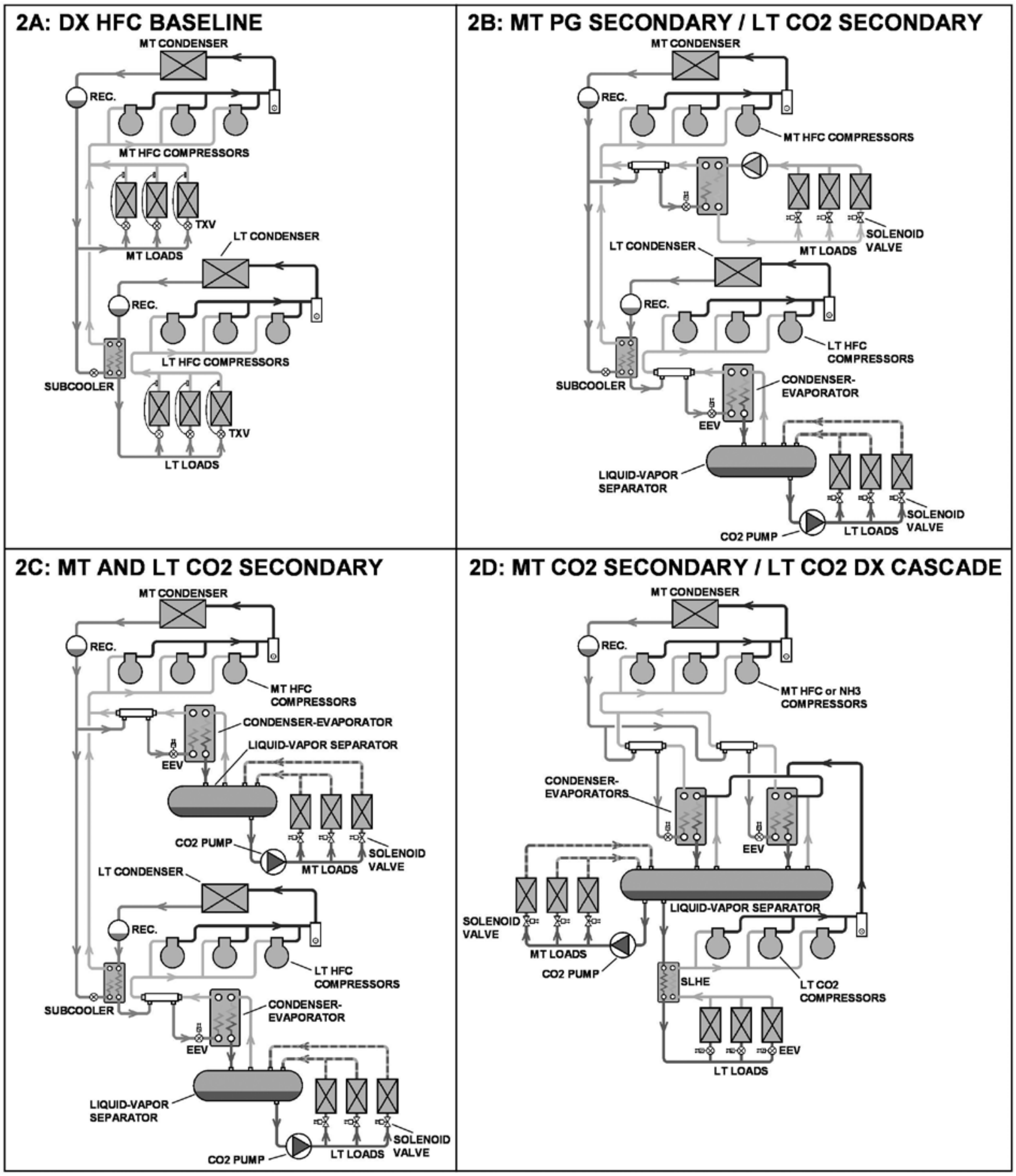


Figure 3: Bin Weather Data for Atlanta, GA

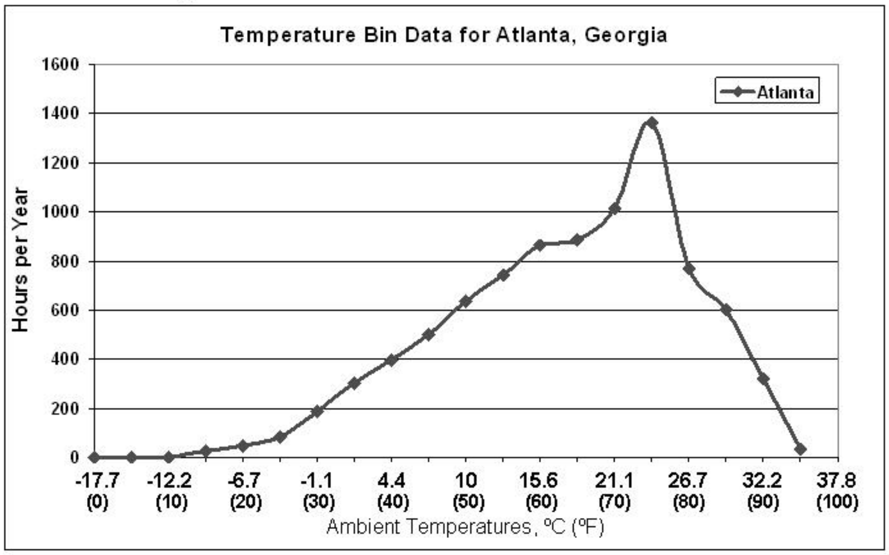


Figure 4: TEWI Comparison Analyzed Systems

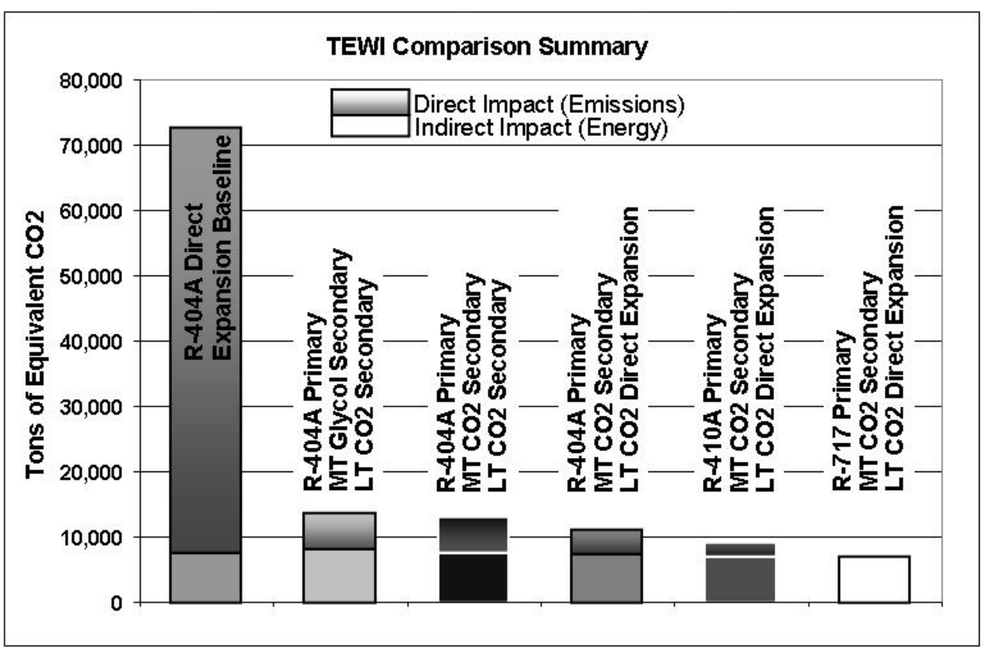
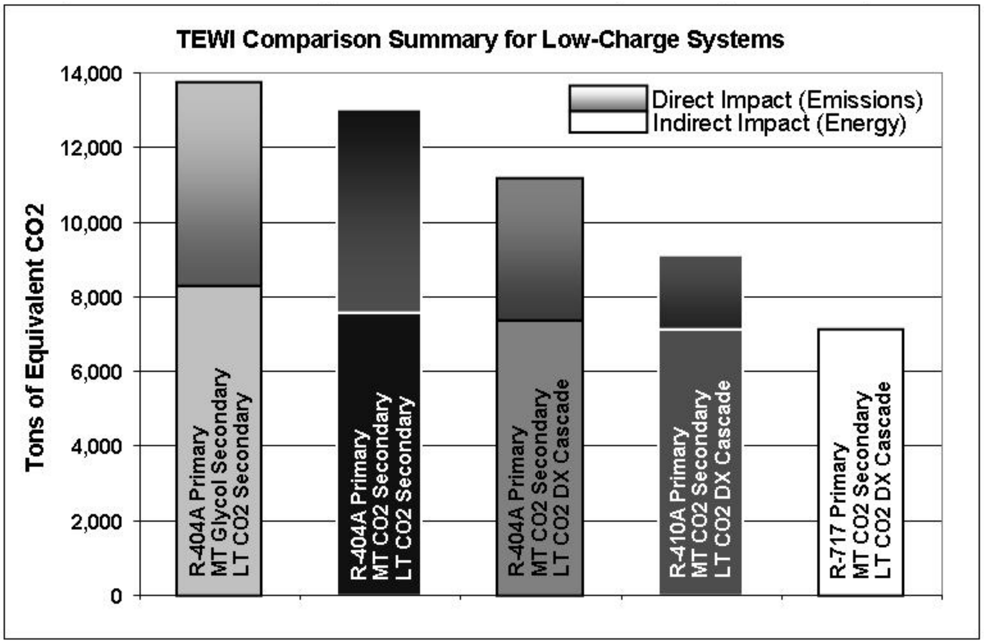


Figure 5: TEWI Comparison of Low-Charge Refrigeration Systems



Notes: