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# Carbon Dioxide

## In North American Supermarkets

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Commercial refrigeration system design in North America has not been subjected to the pressures of HFC bans or high taxation of refrigerants as in other parts of the world and is still driven primarily by energy consumption and system cost. The recent move towards systems with lower refrigerant charge and lower leakage rates has been, for the most part, 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 mandated for systems operating with HFC refrigerants. However, this is expected to change soon.

When the reduction of refrigerant charge is desired, two systems are generally considered: distributed direct expansion systems, and secondary coolant systems. Distributed direct expansion

systems have been applied successfully and are 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 another alternative that can provide more significant reductions in charge and leakage rate. Introduced in the U.S. in 1996, medium-temperature secondary coolant systems using propylene glycol are gaining wider acceptance and represent a significant portion of business with more than 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

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and decreased system maintenance. Concerns about increased energy consumption have largely been overcome through proper design practice.<sup>1</sup>

Low-temperature secondary systems were introduced in the late 1990s 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 performance to other single-phase fluids,<sup>2</sup> material compatibility continues to be a concern. 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 future applications.

In searching for alternative fluids suitable for low-temperature application, it became clear that CO<sub>2</sub> 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<sub>2</sub> appeared to be the higher operating pressures and availability of components.

### CO<sub>2</sub> Supermarket Installations in North America

In 2001, laboratory testing of low-temperature CO<sub>2</sub> secondary systems was initiated. After extensive investigation of the system operation, display-case and unit-cooler performance, and various piping configurations and control methods, the first U.S. system was installed in the field in mid-2006.

By the end of 2008, nine low-temperature carbon dioxide systems were operational in the U.S. and Canada. All systems used CO<sub>2</sub> 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/h). The systems also universally included a primary refrigeration system using an HFC (R-404A or R-507A). Most installations included a medium-temperature secondary coolant system using propylene glycol, although one site was installed with a centralized direct-expansion HFC system that served as a comparison site for energy and performance monitoring.

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 secondary coolant applications and will continue to be critical to the implementation of these and other types of CO<sub>2</sub> systems in the future.

The selection of CO<sub>2</sub> grade or purity-level has been carefully considered. Initial systems used CO<sub>2</sub> 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<sub>2</sub> through liquid filter-driers and a purge of non-condensable gases during start-up seems to make specification of a higher purity unnecessary. CO<sub>2</sub> costs and purity-level availability appear to vary widely throughout the country, however most installations have been able to obtain the materials around 1.10 \$US/kg (0.50 \$US/lb).

CO<sub>2</sub> systems have shown to be susceptible to the same types of 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. Similar to a direct expansion system, non-condensable gases not evacuated from the system or charged into the system from impure refrigerant tend to accumulate in the condenser-evaporator heat exchangers, resulting in what appears to be reduced condensing capacity. In extreme cases, flow of the CO<sub>2</sub> thermosiphon effect to these heat exchangers can be blocked completely, requiring system reevacuation and recharging. Good-quality filter-driers installed in the liquid lines and changed shortly after start-up have eliminated any problems associated with moisture in the systems.

Since a secondary-loop CO<sub>2</sub> system is essentially a recirculated liquid system with wet returns (circulation rate greater than one), 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.

Since CO<sub>2</sub> systems are typically designed for maximum working pressures well below the saturation pressure at ambient temperatures (60 bar at 23°C, or 900 psig at 75°F), back-up or auxiliary refrigeration units have been installed on some stores to provide a source of cooling for the CO<sub>2</sub> during extended power outages or maintenance procedures. Experience has shown that it takes several hours for the pressure in the system to reach levels where relief to ambient would be required, and 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<sub>2</sub> field installation and three similar HFC DX systems nearby. Results have so far indicated a 2% to 3% average reduction in energy required by the low-temperature CO<sub>2</sub> 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<sub>2</sub> 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 previously, the field data indicated performance of the low-temperature CO<sub>2</sub> secondary system that 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 and, for the most part, experience 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 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 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<sub>2</sub>, both as a secondary coolant and in the future, as 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.

To estimate heat gain into distribution piping, a simple heat transfer model of an insulated pipe can be used as shown in *Figure 1*. Insulation properties are available from the insulation manufacturers and internal and external heat transfer coefficients can be easily calculated for different system types using well-known correlations. Equation 1 can be used to calculate total heat transfer once the pipe's UA value is calculated as shown in Equation 2. If the temperature change of the fluid in the piping is relatively small, an average fluid temperature can be used to calculate the heat transfer from the fluid to ambient and is shown in Equation 3, with the final outlet temperature  $T_2$  calculated using an iterative approach. The external heat transfer coefficient from the insulation surface to the surrounding low-velocity air was calculated from Equation 4, which is a simplified approximation shown separately in SI and I-P units, and the convective heat exchange inside the copper tube was calculated from Equations 5 and 6.

$$Q = UA \cdot (T_{AMB} - T_{FLUID}) \quad (1)$$

$$UA = \frac{L}{\frac{1}{\pi h_{inside} d_i} + \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi k_{tube}} + \frac{\ln\left(\frac{d_a}{d_o}\right)}{2\pi k_{insulation}} + \frac{1}{\pi h_{outside} d_a}} \quad (2)$$

$$Q_{heatgain} = \dot{m}_{FLUID} (h_2 - h_1) = UA \left( T_{AMB} - \frac{T_1 + T_2}{2} \right) \quad (3)$$

## Nomenclature

$Q$	= Overall heat transfer in W (Btu/h)
$UA$	= Overall heat transfer coefficient in W/K (Btu/h · °F)
$T_{AMB}$	= Ambient temperature surrounding pipe in °C (°F)
$T_{FLUID}$	= Average fluid temperature in pipe section in °C (°F)
$L$	= Length of pipe section in m (ft)
$d_o$	= Outside diameter of pipe in m (ft)
$d_i$	= Inside diameter of pipe in m (ft)
$d_a$	= Outside diameter if insulation in m (ft)
$h_{inside}$	= Convective heat transfer coefficient inside pipe in between fluid and internal pipe wall in W/m <sup>2</sup> · K (Btu/h · ft <sup>2</sup> · °F)
$h_{outside}$	= Convective heat transfer coefficient between ambient and surface of outside insulation in W/m <sup>2</sup> · K (Btu/h · ft <sup>2</sup> · °F)
$k_{tube}$	= Thermal conductivity of tube wall material in W/m · K (Btu/h · ft · °F)
$k_{insulation}$	= Thermal conductivity of insulation material in W/m · K (Btu/h · ft · °F)
$\dot{m}_{FLUID}$	= Mass flow rate of refrigerant inside pipe in kg/s (lb/h)
$h_1$	= Fluid enthalpy at pipe inlet in kJ/kg (Btu/lb)
$h_2$	= Fluid enthalpy at pipe outlet in kJ/kg (Btu/lb)
$T_1$	= Fluid temperature at pipe inlet in °C (°F)
$T_2$	= Fluid temperature at pipe outlet in °C (°F)
$V$	= Ambient air velocity in m/s (ft/s)
$D$	= Outside diameter of insulation in m (ft)
$Nu_{inside}$	= Nusselt number of internal pipe flow (nondimensional)
$Re$	= Reynolds number of internal pipe flow (nondimensional)
$Pr$	= Prandtl number of internal pipe flow (nondimensional)
$k_{FLUID}$	= Thermal conductivity of fluid in W/m · K (Btu/h · ft · °F)
$Re$	= Reynolds number of internal pipe flow (nondimensional)

$$h_{outside} = 2.755 V^{0.471} / D^{0.529} \quad (\text{SI Units Only}) \quad (4)^3$$

$$h_{outside} = 0.5198 V^{0.471} / D^{0.529} \quad (\text{I-P Units Only})$$

$$Nu_{inside} = 0.23 Re^{0.8} Pr^{0.3} \quad (5)$$

$$\text{where } h_{inside} = Nu_{inside} k_{FLUID} / d_i \quad (6)$$

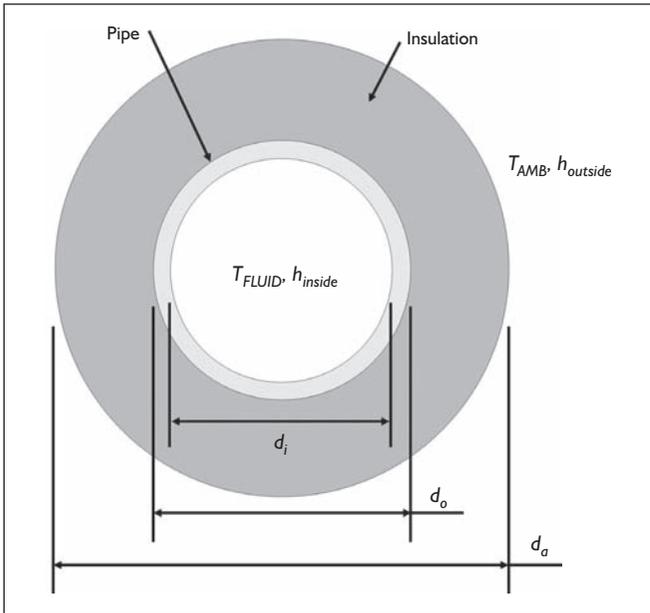


Figure 1. Distribution piping heat transfer.

To verify that the heat gain calculation method was appropriately applied, measurements were made on an existing 22 kW (75 kBtu/h) experimental low-temperature CO<sub>2</sub> cascade system installed in the laboratory. System piping included several supply and return lines installed in a loop-style network in the same manner as installations in the field, and consisted of copper piping from 3/8 in. to 1 5/8 in. outside diameter (9 to 41 mm), insulated with 1 in. thick (25 mm) closed cell elastomeric foam and in lengths from 12 to 30 m (40 to 100 ft.). Several conditions were tested at various temperatures and measurements were made on the supply and return lines of the system. Pressure and temperature were measured at the entrance and exit of each supply and return line using calibrated

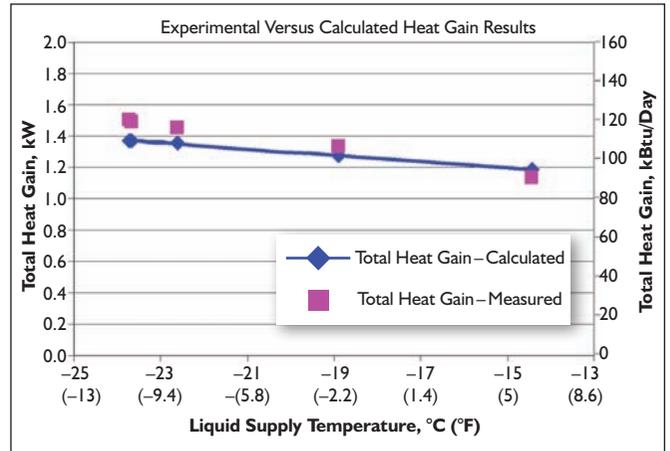


Figure 2: Heat gain calculation verification.

thermocouples and pressure transducers, and fluid mass flow rate measurements were made using a coreolis meter. Figure 2 shows the calculated and measured total heat gain into the supply and return distribution piping based on the methodology outlined previously. The calculations show good agreement with the measured values with a maximum difference of 7%.

With the methodology established, an analysis of a typical supermarket was carried out to quantify the differences in the heat gain into the distribution piping network for a variety of system types. A representative fixture plan was selected on which to base the analysis. Figure 4 shows the general layout of the 3600 m<sup>2</sup> (39,000 ft<sup>2</sup>) store with the medium-temperature loads shown in green, and the low-temperature loads under investigation shown in blue.

Low-temperature system types under consideration for the analysis included:

- Low-temperature HFC direct expansion system with two suction groups (existing system);

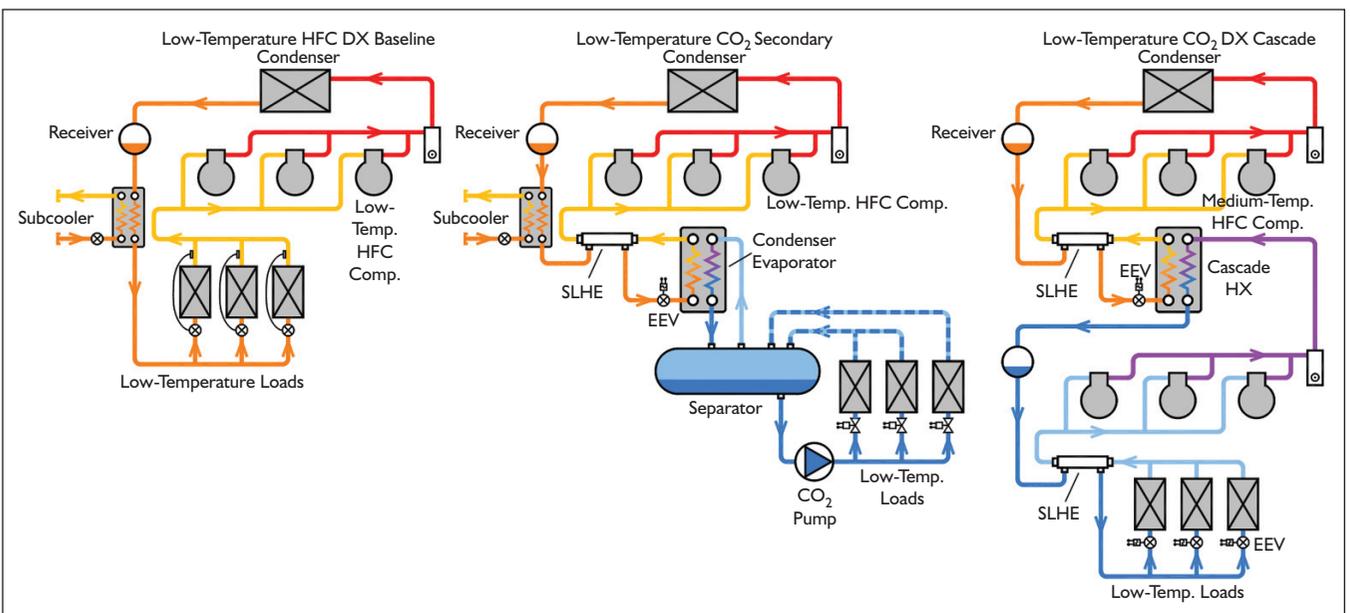


Figure 3: Basic piping configuration of three low-temperature systems under analysis.

- Low-temperature carbon dioxide secondary coolant system with single suction group; and
- Low-temperature carbon dioxide direct expansion cascade system with two suction groups.

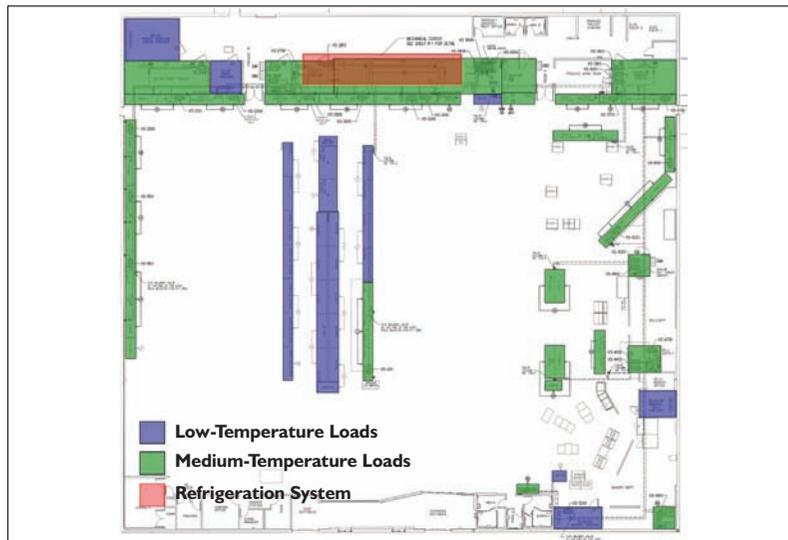
Basic piping diagrams for the three systems are shown in *Figure 3*. In the DX baseline system and the CO<sub>2</sub> secondary system, the HFC component is a low-temperature system, which can be subcooled by another medium-temperature system installed nearby. Subcooling of low-temperature systems using a system operating at higher saturated suction temperature is an important energy saving feature that is considered in the following analysis.

Circuiting and line sizing for the low-temperature HFC DX baseline system was based on the existing circuited system configuration that is currently manufactured. Line sizes for the various CO<sub>2</sub> 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<sub>2</sub> 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).

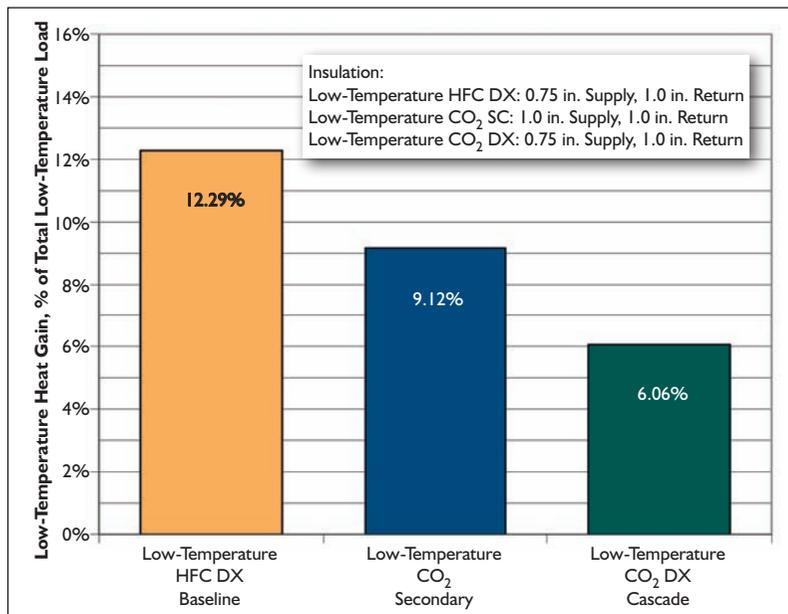
With system configurations established, the analysis for each system type was carried out. The bar graph in *Figure 5* shows the results of the analysis of distribution piping heat gain for the three different low-temperature systems. The analysis indicates that although the line temperatures are lower for the CO<sub>2</sub> secondary coolant system, the HFC direct expansion system has significantly higher heat gain due to the larger pipe diameters and installed lengths of copper piping. The smaller lines and somewhat warmer pipe temperatures in the CO<sub>2</sub> cascade system lead to further reductions in heat gain.

In addition to heat gain, it is interesting 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.

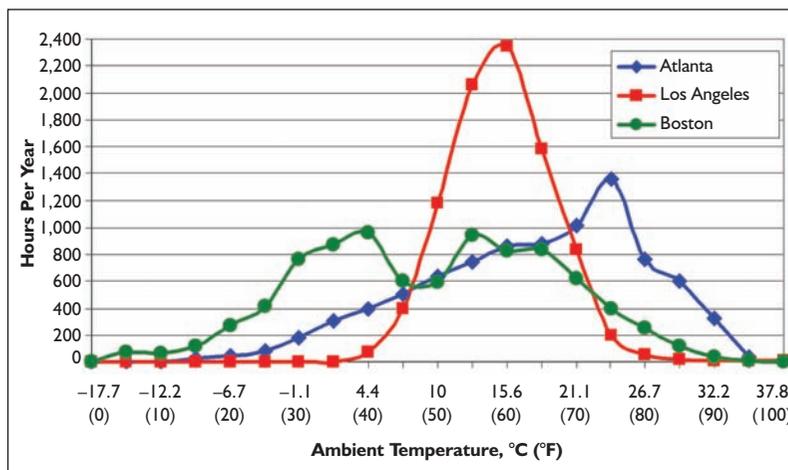
*Table 1* summarizes the installed copper length and weight of copper pipe for each of the system



*Figure 4: Representative store layout.*



*Figure 5: Low-temperature system heat gain into distribution piping.*



*Figure 6: Bin weather data for selected cities.*

System Type	Installed Copper Length m (ft), Percent Change	Installed Copper Weight kg (lb), Percent Change
Low-Temperature HFC DX Baseline (Dual-Group)	1690 m (5,544 ft)	1147 kg (2,530 lb)
Low-Temperature CO <sub>2</sub> Secondary (Single-Group)	753 m (2,472 ft), -55%	309 kg (681 lb), -73%
Low-Temperature CO <sub>2</sub> DX Cascade (Single-Group)	753 m (2,472 ft), -55%	239 kg (527 lb), -79%
Low-Temperature CO <sub>2</sub> DX Cascade (Dual-Group)	809 m (2,655 ft), -52%	249 kg (549 lb), -78%

Table 1: Installed length and weight of copper piping for various system types.

types. An additional configuration was added for the low-temperature CO<sub>2</sub> cascade system with only one suction group to see if any significant installation savings could be obtained through combining the systems. Significant decreases in installed length of copper, from 52% to 55%, and dramatic decreases in installed weight, from 73% to 79%, are characteristic of the CO<sub>2</sub> system types compared to the HFC baseline system. This indicates that the installation costs of the CO<sub>2</sub> systems, once fully commercialized, can be expected to be lower than existing HFC systems due to both reduced labor and material costs.

Regarding investigation of the CO<sub>2</sub> cascade system with both one and two suction groups, it is clear that savings associated with a single suction group are quite minimal (7% less installed length and 4% less installed copper weight) and would not be significant compared to the higher energy consumption associated with the single suction group.

### System Performance Analysis

Following the heat gain and piping analysis, an annual energy analysis was performed on the three systems under investigation. A bin analysis was performed based on weather data for three different climate regions: Atlanta, Boston, and Los Angeles, using weather bin data in 2.8 K (5°R) increments. Figure 6 illustrates the temperature variations for the three climactic regions and hourly occurrence per year.

Minimum condensing temperatures were set at 21°C (70°F) for the low-temperature HFC DX system and 10°C (50°F) for the low-temperature CO<sub>2</sub> systems; the primary HFC portion of the CO<sub>2</sub> systems would use electronic expansion valves that 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 readjustment to function well under these conditions. Analysis was also performed with all systems operating with minimum condensing temperatures of 10°C (50°F) for comparison.

Figure 7 shows the results of the annual energy analysis for the three climactic regions for the low-temperature HFC DX

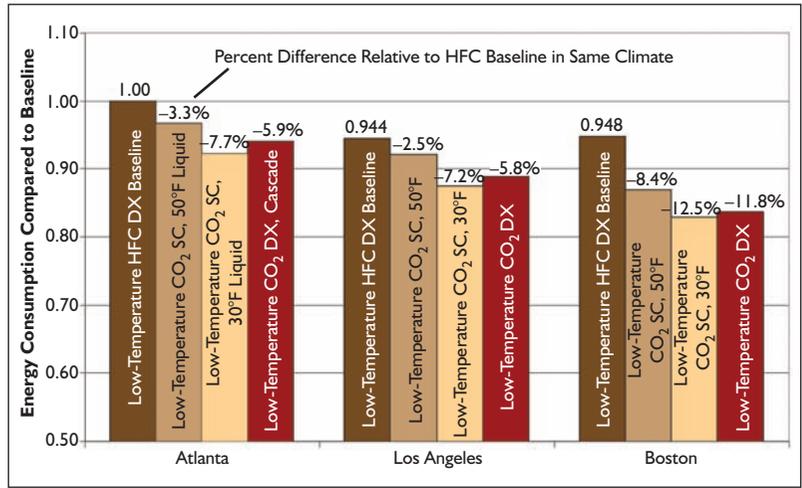


Figure 7: Relative system energy consumption comparison.

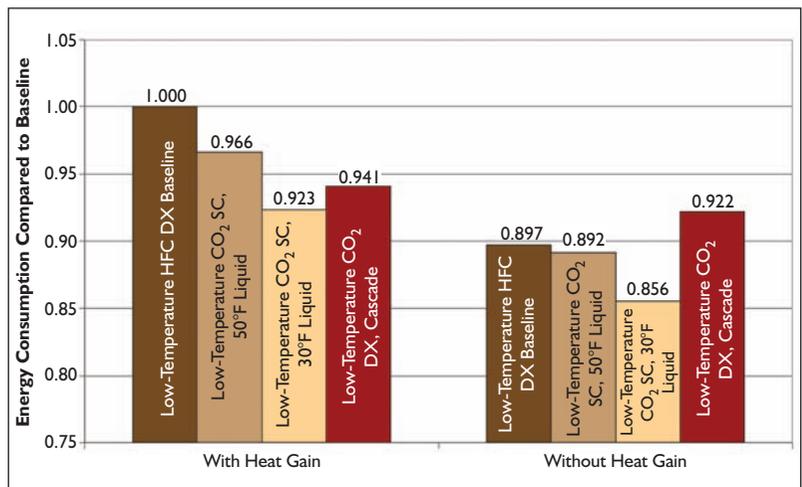


Figure 8: Impact of neglecting heat gain.

system, the low-temperature CO<sub>2</sub> secondary system, and the low-temperature CO<sub>2</sub> DX cascade system. Two variations of the low-temperature CO<sub>2</sub> secondary system were analyzed, one with liquid subcooling from the medium-temperature system providing +10°C (+50°F) subcooled liquid (the same as the low-temperature HFC DX system) and one with deeper liquid subcooling providing -1°C (+30°F) subcooled liquid. The deeper subcooling provided by the medium-temperature system is a unique feature that the low-temperature CO<sub>2</sub> secondary systems can take full advantage of due to the close proximity

System Type	HFC Charge kg (lb), Percent Change	Direct Effect (Emissions) tons CO <sub>2</sub> , Percent Change	Indirect Effect (Energy) tons CO <sub>2</sub> , Percent Change	TEWI tons CO <sub>2</sub> , Percent Change
Low-Temperature HFC DX Baseline (Dual-Group)	680 kg (1,500 lb)	10,283 ton	3,470 ton	13,753 ton
Low-Temperature CO <sub>2</sub> Secondary (Single-Group), 10°C (50°F) Subcooled Liquid	272 kg (600 lb), -60%	866 ton, -92%	3,354 ton, -3.4%	4,220 ton, -69%
Low-Temperature CO <sub>2</sub> Secondary (Dual-Group), -1°C (30°F) Subcooled Liquid	272 kg (600 lb), -60%	866 ton, -92%	3,203 ton, -7.7%	4,069 ton, -70%
Low-Temperature CO <sub>2</sub> DX Cascade (Dual-Group)	272 kg (600 lb), -60%	866 ton, -92%	3,265 ton, 5.9%	4,131 ton, -70%

Table 2: Refrigerant charge and TEWI versus system type for Atlanta climate conditions.

of the low-temperature subcooler and evaporators.

The analysis shows that low-temperature secondary coolant systems can save 3% to 12% compared to the low-temperature HFC DX baseline system depending on climate and level of subcooling. Additionally, the low-temperature DX cascade systems can save between 5% and 11% depending on the same subcooling and climate factors. If the minimum condensing pressures for the DX systems are allowed to float to the same minimum level of 10°C (50°F) level as the DX HFC baseline system, the differences between the system types becomes smaller though still favorable for the CO<sub>2</sub> systems. Energy consumption for the secondary coolant systems saves 0% to 5% compared to the low-temperature HFC DX baseline system, and the cascade system saves 3% to 4% depending on the climate under investigation.

Regarding the impact of including the heat gain analysis in the modeling, the analysis was repeated for the Atlanta climate conditions but without including the variation in load caused by the piping heat gain. Figure 8 compares the low-temperature system energy both with and without the heat gain impacts. A different comparison of the systems can emerge when the

heat gain is ignored: loads on all the system types are under-predicted and the CO<sub>2</sub> cascade system becomes an unfavorable alternative compared to the other system choices. Analysis of the systems without the heat gain would provide a misleading view for comparison of the technologies.

Along with energy consumption, a comparison of refrigerant charge and total equivalent warming impact (TEWI) was performed. Table 2 shows the results of this analysis and the benefit to equivalent carbon emissions of all the CO<sub>2</sub> system types. The HFC charge of the DX baseline system was based on actual field experience, and a conservative estimate of 60% reduction in HFC charge was used for the CO<sub>2</sub> systems. Field experiences also indicate that better reductions can be achieved but depend on condenser type and type of heat reclaim that is used. Direct and indirect impacts of TEWI are shown for each system. As the energy consumption of the four systems analyzed is very close, the impact on total TEWI is primarily a result of the reduction in HFC charge. A value of 0.619 kg CO<sub>2</sub> per kWh electricity generated was used for calculating the indirect impacts. This is an average value for the state of Georgia as Atlanta climate conditions were used for the analysis. However this is close to the U.S.

average rate of 0.606 kg CO<sub>2</sub> per kWh.<sup>4</sup> Refrigerant leakage rate for the HFC direct expansion system was assumed to be the current U.S. average of 25%. Leakage rates for the HFC systems confined in the mechanical room were assumed to be 5%. An end-of-life recovery rate of 95% was used for all system types.

#### Future Systems

Low-temperature CO<sub>2</sub> secondary systems went into full commercialization in 2008 with many new installations under way. Extension of the technology to medium-temperature applications is viable and offers several performance improvements

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compared to the propylene glycol secondary systems used today, however systems need to be carefully designed and optimized to ensure they can be produced in an economically viable manner.

Additionally, in early 2008 the first low-temperature CO<sub>2</sub> cascade system was installed and started in a U.S. supermarket, and this site will be used as a pilot installation to evaluate the technology. Questions still remain regarding system cost compared to today's HFC direct expansion systems and low-temperature CO<sub>2</sub> secondary systems, and laboratory investigations will continue to optimize energy consumption and determine how to best commercialize this technology.

### Discussion

A basic understanding of the operation of low-temperature CO<sub>2</sub> secondary coolant systems has been established through experience with the first pilot installations in the field. A detailed study of the effects of heat gain into the distribution piping network was presented and indicates that careful consideration of these effects is necessary to avoid misleading conclusions when comparing technologies of fundamentally different types. CO<sub>2</sub> systems have also demonstrated to have significant installation savings in terms of the amount of required copper piping.

An energy analysis was presented, which indicates both low-temperature CO<sub>2</sub> secondary and direct expansion systems

can be implemented with energy consumption better than traditional HFC direct expansion systems by 3% to 12% depending on system type and configuration. Finally, a TEWI analysis demonstrates that these systems are of great importance when reductions in carbon emissions are required.

### Author's Note

A version of this paper was presented at the IIR Gustav Lorentzen Natural Working Fluids Conference held in 2008 in Copenhagen, Denmark. Calculations have been updated to reflect the latest available GWP values for the HFC refrigerants, U.S. average leak rates for supermarket systems, and carbon production per energy output used in TEWI calculations.

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