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Omar Abdelaziz
abdelazizoa@ornl.gov

Brian Fricke

Edward A. Vineyard

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Omar ABDELAZIZ[1]*, Brian FRICKE[2], Edward VINEYARD[3]

Oak Ridge National Laboratory[†], Energy and Transportation Science Division,
Oak Ridge, TN, USA
[1](P) 865-574-2089, (F) 865-574-9338, abdelazizoa@ornl.gov
[2](P) 865-576-0822, (F) 865-574-9338, frickeba@ornl.gov
[3](P) 865-574-0576, (F) 865-574-9338, vineyardea@ornl.gov

* Corresponding Author

ABSTRACT

Commercial refrigeration systems are known to be prone to high leak rates and to consume large amounts of electricity. As such, both the direct emissions related to refrigerant leakage and the indirect emissions resulting from primary energy consumption contribute greatly to their Life Cycle Climate Performance (LCCP). In this paper, an LCCP design tool is used to evaluate the performance of a typical commercial refrigeration system with alternative refrigerants and minor system modifications to provide lower Global Warming Potential (GWP) refrigerant solutions with improved LCCP compared to baseline systems. The LCCP design tool accounts for system performance, ambient temperature, and system load; system performance is evaluated using a validated vapor compression system simulation tool while ambient temperature and system load are devised from a widely used building energy modeling tool (EnergyPlus). The LCCP design tool also accounts for the change in hourly electricity CO_2 emission rates to yield an accurate prediction of indirect emissions. The analysis shows that conventional commercial refrigeration system life cycle emissions are largely due to direct emissions associated with refrigerant leaks and that system efficiency plays a smaller role in the LCCP. However, as a transition occurs to low GWP refrigerants, the indirect emissions become more relevant. Low GWP refrigerants may not be suitable for drop-in replacements in conventional commercial refrigeration systems; however some mixtures may be introduced as transitional drop-in replacements. These transitional refrigerants have a significantly lower GWP than baseline refrigerants and as such, improved LCCP. The paper concludes with a brief discussion on the tradeoffs between refrigerant GWP, efficiency and capacity.

1. INTRODUCTION

The refrigerant industry has undergone significant transformation over recent decades to systematically phase-out harmful ozone-depleting substances. Chlorofluorocarbons (CFC) were the first substances to be regulated, and now hydrochloroflurocarbons (HCFC), which are Class II ozone-depleting substances, are targeted for phase-out.

All developed countries are subject to consumption and production restrictions for HCFCs under the Montreal Protocol on Substances that Deplete the Ozone Layer, which governs the use of ozone-depleting substances. When it was first ratified in 1987, the Protocol established a timeline for the gradual phase-out of CFC refrigerants, wherein CFC production ceased in all developed countries by 1996. Subsequent rulings resulted in the phase-out of halons, carbon tetrachloride, trichloroethane, hydrobromofluorocarbons (HBFC), bromochloromethane and methyl bromide. The only phase-out still in process is for HCFCs, which became predominant in the market as replacements for

CFCs. As of January, 1, 2010, developed countries reduced their HCFC use by 75% of the baseline level. This draw down will reach 90% in 2015, 99.5% in 2020, and 100% in 2030. Implementation of the Montreal Protocol in the United States has been regulated through Title VI of the Clean Air Act, under which the U.S. Environmental Protection Agency has accelerated the phase-out of ozone-depleting substances.

With HCFCs being removed from the market, manufacturers have turned to hydrofluorocarbons (HFC) as substitutes. The zero ozone-depleting potential (ODP) of HFCs is appealing; however, many have relatively high GWPs. The need to be compliant with the Montreal Protocol and related mandates has driven the industry to revisit the use of long-term alternative refrigerants, such as hydrocarbons, that are favorable for the environment and exhibit strong thermodynamic properties. Hydrocarbons are considered as a replacement for HCFCs and HFCs due to their zero ODP and relatively low GWP. Some hydrocarbons have been approved for use in the United States in 2010 under the U.S. Environmental Protection Agency's (EPA) Significant New Alternatives Policy (SNAP) program, which was created in 1994 to facilitate the approval and adoption of more environmentally-friendly refrigerants in the United States. Unfortunately, hydrocarbons (and hydrocarbon blends) are flammable, which poses a safety concern when used in equipment originally designed for CFC or HCFC use. This is especially relevant for equipment subject to leaks and large charge amounts such as commercial refrigeration equipment.

Another refrigerant which has recently gained the attention of the commercial refrigeration industry is carbon dioxide. Since carbon dioxide has zero ODP and a GWP value of 1, it is an environmentally-friendly refrigerant option. Carbon dioxide is nonflammable and nontoxic, and furthermore, it has very good heat transfer characteristics. However, operating pressures for carbon dioxide are significantly higher compared to commonly used HFC refrigerants. In addition, the critical temperature of CO_2 is low, which indicates that at high ambient temperatures, CO_2 will be supercritical. Thus, in the commercial refrigeration sector, transcritical CO_2 booster refrigeration systems are currently being designed which maximize performance and safety. In cool and moderate climates, the on-going developments in transcritical CO_2 booster refrigeration technology have resulted in systems which are competitive with HFC systems (Ge and Tassou 2011).

While guidelines are in place to specify refrigerants which cannot be used in the United States, little or no policies or incentives currently exist nationwide to specifically promote low-GWP refrigerants, with the possible exception of the EPA's GreenChill program. This voluntary program is a partnership between the EPA and food retailers to reduce refrigerant emissions and to decrease the environmental impact of commercial refrigeration systems (EPA 2012). Regardless, manufacturers anticipate a continued momentum set by the government in search for cleaner, lower-GWP refrigerants, so they are conducting ongoing research and development for such compounds.

In this paper, a Life Cycle Climate Performance (LCCP) design tool is used to evaluate the performance of a typical commercial refrigeration system with alternative refrigerants and minor system modifications in order to provide guidance on lower Global Warming Potential (GWP) refrigerant solutions with improved LCCP compared to baseline systems.

2. METHODOLOGY

Life Cycle Climate Performance of different refrigerants for a specific application is the sum of direct and indirect emissions. According to Papasavva et al. (2010), direct and indirect emissions can be calculated as shown in Equations 1 and 2 below

$$Em_{direct} = Em_{leakage} + Em_{servicing} + Em_{accicents} + Em_{EOL\ lost} + Em_{leakage(prod.\&trans.)} \tag{1}$$

Where:
Em_{direct} = Direct Emissions
$Em_{leakage}$ = Emissions due to refrigerant leakage
$Em_{accicents}$ = Irregular emissions due to such things as accidents
$Em_{servicing}$ = Emissions due to leakage of refrigerant from servicing
$Em_{EOL\ lost}$ = Emissions due to end-of-life of system where refrigerant is lost
$Em_{leakage\ prod.\&transp.}$ = Emissions due to leakage that occurs from the refrigerant production and transportation

$$Em_{indirect} = Em_{manuf.,\ sys.} + Em_{manuf.,\ ref} + Em_{E.\ EOL,\ sys.} + Em_{E.\ EOL,\ ref.} + Em_{E.\ op.} + Em_{E.\ ,transp.} \quad (2)$$

Where:

$Em_{indirect}$ = Indirect Emissions

$Em_{manuf.,\ sys.}$= Emission due to energy used to manufacture components and display cases in system

$Em_{manuf.,\ ref}$ = Emission due to energy used to manufacture refrigerant

$Em_{E.\ EOL,\ sys.}$= Emission due to energy used for end-of-life of components

$Em_{E.\ EOL,\ ref.}$ = Emission due to energy used for end-of-life of refrigerants

$Em_{E.\ op.}$ = Emission due to energy consumption of system during its lifetime

$Em_{E.\ ,transp.}$ = Emission due to energy used to transport equipment

In this paper, we will only consider direct emissions due to leakage, servicing, and end-of-life loss and indirect emissions due to energy consumption of system during its lifetime. For the emissions related to energy consumption during the lifetime; we used EnergyPlus (2012) to model typical supermarkets in relevant US locations to incorporate the climate significance. The EnergyPlus simulations provide estimates of the hourly electric energy consumption for the commercial refrigeration system. The hourly electric consumption is then multiplied by the hourly electric emission rate for the relevant interconnect region based on data published in NREL (2011). The annual emissions were then multiplied by the system lifetime to estimate the total indirect emissions.

3. BUILDING ENERGY MODELING

The EnergyPlus supermarket model used in this study was based on the new construction reference supermarket model developed by the U.S. Department of Energy (Deru et al. 2010). This model single-story supermarket has a floor area of 4,181 m^2 with a floor-to-ceiling height of 6.1 m, and is divided into six zones (office, dry storage, deli, sales, produce and bakery). Exterior wall construction consists of stucco, concrete block, insulation and gypsum while roof construction consists of roofing membrane, insulation and metal decking. Internal loads include people and lighting, as well as miscellaneous gas and electric loads in the deli and bakery zones. HVAC is provided by packaged constant volume units with gas heat and electric cooling. Specifications for the refrigerated display cases and walk-in coolers/freezers used in the model supermarket are shown in Tables 1 and 2, respectively.

To determine the significance of climate on refrigeration system performance, the energy consumption of the model supermarket was estimated for 16 U.S. cities using EnergyPlus. These 16 cities, shown in Table 3, are representative of the eight climate zones in the U.S.

Table 1: Refrigerated Display Case Specifications used in Model Supermarket

Case Type	Length (m)	Cooling Capacity (W/m)	Evaporator Fan (W/m)	Lighting (W/m)	Defrost (W/m)	Anti-Sweat Heater (W/m)	Case Temperature (°C)
Medium Temperature							
Multi-deck meat	36.6	1,442	88	39	443	66	2.2
Multi-deck other	79.2	1,442	41	60	0	0	2.2
Low Temperature							
Reach-in	81.7	538	66	108	1,312	233	-15.0
Single-level open	39.0	529	33	0	1,378	79	-12.2

Table 2: Refrigerated Walk-In Cooler/Freezer Specifications used in Model Supermarket

Walk-In Type	Area (m^2)	Cooling Capacity (W/m^2)	Evaporator Fan (W/m^2)	Lighting (W/m^2)	Defrost (W/m^2)	Anti-Sweat Heater (W/m^2)	Walk-In Temperature (°C)
Medium Temperature							
Meat Cooler	37.2	190	40	11	210	0	2.2
Food Cooler	241.5	190	26	11	0	0	2.2
Low Temperature							
Food Freezer	92.9	250	43	11	310	0	-23.3

Table 3: Climate Zones and Cities used in the EnergyPlus Simulations

Climate Zone	City	Annual Average Temperature (°C)
1A	Miami, FL	24.9
2A	Houston, TX	20.7
2B	Phoenix, AZ	23.8
3A	Atlanta, GA	17.0
3B	Los Angeles, CA	17.3
3B	Las Vegas, NV	20.2
3C	San Francisco, CA	14.4
4A	Baltimore, MD	13.3
4B	Albuquerque, NM	14.2
4C	Seattle, WA	11.4
5A	Chicago, IL	10.0
5B	Boulder, CO	10.3
6A	Minneapolis, MN	8.0
6B	Helena, MT	7.2
7	Duluth, MN	4.3
8	Fairbanks, AK	-2.1

4. CANDIDATE REFRIGERATION SYSTEMS

In this paper we consider 2 different commercial refrigeration systems to serve the medium and low temperature loads described in section 3 above. The first is a typical multiplex direct expansion (DX) supermarket refrigeration system conceptually described in Figure 1 below. The medium temperature (MT) load is served with 2 MT compressor racks. The low temperature (LT) load is served with 2 LT compressor racks. The overall multiplex DX system configuration is summarized in Table 4 below. Each of the compressor racks is connected to a dedicated air-cooled condenser. This system was originally designed for R404A; which is used as a baseline for this paper. The compressors used for the 4 racks are commercially available compressors with published performance maps. The compressor manufacturer provided the compressor maps of these compressors when operating using different refrigerants such as R22, R407A, and R507A. Only R407A was considered for analysis in this paper since it has 0 ODP and almost half the GWP of the baseline refrigerant. R407A has superior thermo-physical properties compared to R404A resulting in improved condenser performance which was accounted for during the EnergyPlus simulations.

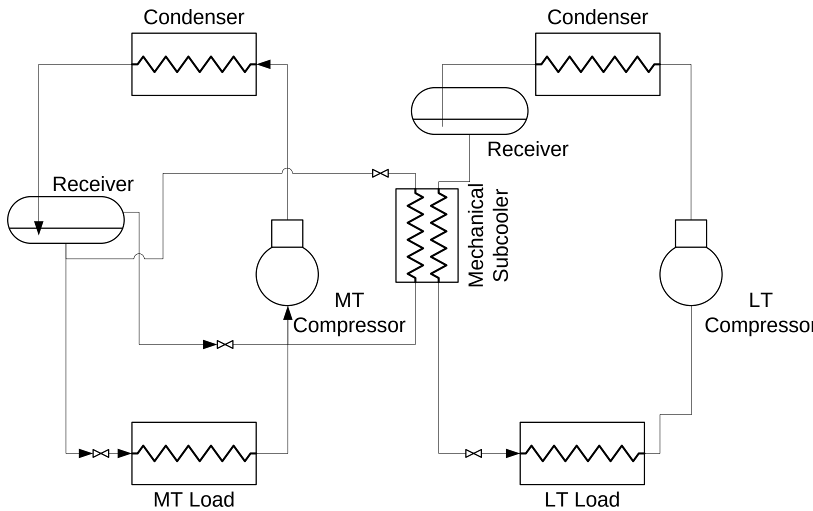

Figure 1: Typical multiplex DX system configuration

Table 4: HFC (R404A/R407A) Compressor Racks' Configuration

Compressor Rack	Capacity (kW)	Charge (kg)	Source of Mechanical Subcooling
MT1	167.1	1905	N.A.
MT2	52.8	589.7	N.A.
LT1	64.6	589.7	MT1
LT2	23.4	215.5	MT2

The second system is a natural refrigerant system. R744 is used in a transcritical booster supermarket configuration as shown in figure 2 below. The system comprises two compressor racks and 2 dedicated condensers. The overall system configuration is summarized in Table 5 below. When the compressor discharge conditions are such that the R744 is in the supercritical region, then the high-side operating pressure is independent of the gas cooler exit temperature (Sawalha, 2008). Thus, for a given gas cooler exit temperature, there is an optimum pressure to achieve the maximum coefficient of performance (COP). Several researchers have developed correlations to determine the optimum high-side pressure in transcritical R744 refrigeration systems. The refrigeration system energy simulation assumed system operation at optimal gas pressure control.

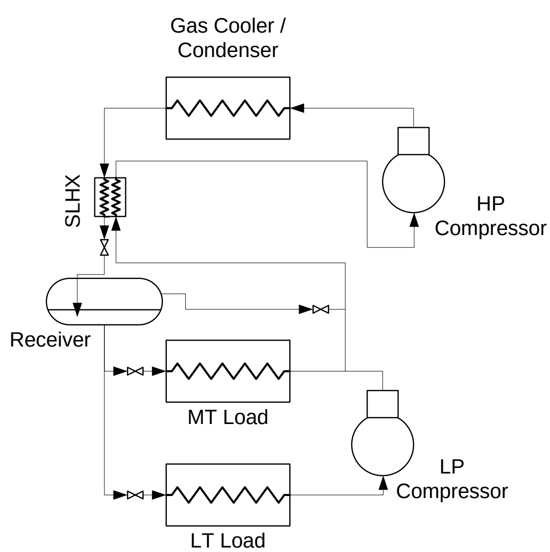

Figure 2: Typical transcritical booster supermarket refrigeration system

During transcritical operation, the gas cooler approach temperature, which is the difference between the gas cooler outlet temperature and the ambient air temperature, was set to 3°C. During subcritical operation, the temperature difference between the saturated condensing temperature and the ambient air temperature was set to 10°C. The pressure in the receiver is typically between 3.5 to 4.0 MPa. The saturated suction temperature for the medium temperature loads is typically around −5°C while for the low temperature loads, the saturated suction temperature is typically around −30°C. A superheat of 10 K or more is used at the exit of the evaporators. Compressor maps for commercially available R744 compressors were used for the energy simulations.

Table 4: R744 Compressor Racks' Configuration

Rack #	MT Capacity (kW)	LT Capacity (kW)	Charge (kg)
1	118.6	64.6	680.4
2	101.2	23.4	453.6

5. RESULTS AND DISCUSSION

The annual energy consumption of the refrigeration system was evaluated for the 16 cities shown in Table 3 using EnergyPlus. Sample hourly energy consumption data for the weather extremes (Miami and Fairbanks) are shown in Figure 3. It can be seen that there is a large variation in hourly energy consumption due to operating schedules and ambient conditions. This is clearly presented in Figure 4 for the month of January.

The annual energy consumption data are summarized in Figure 5 using the three refrigerants R404A, R407A and R744. It can be seen that the R404A and R407A systems consumed approximately the same amount of energy regardless of weather. On the other hand, R744 energy consumption depended greatly on climate zone. The supercritical operation degraded its performance significantly. However, for colder climates, the R744 system operated subcritically for extended periods of time thereby enhancing its efficiency. The R744 system consumed 7% and 19% more energy than the R404A system in Fairbanks, AK and Phoenix, AZ, respectively.

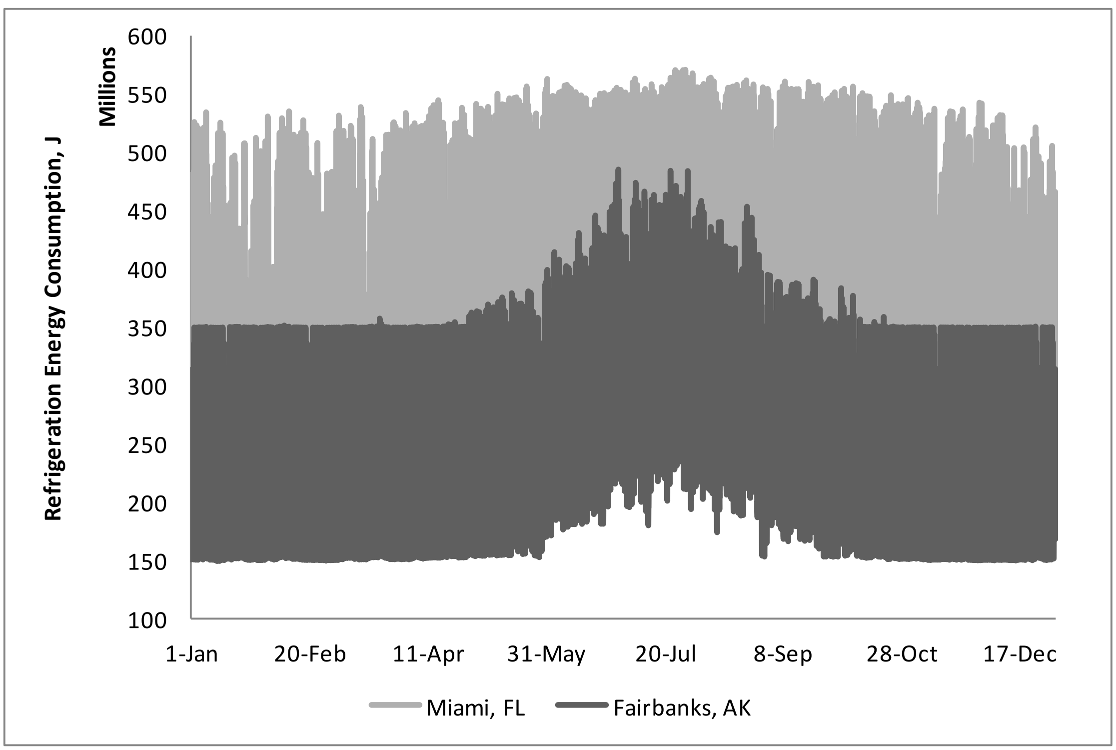

Figure 3: R404A Refrigeration system energy consumption over a year in weather extremes: Miami, FL and Fairbanks, AK with an average energy consumption of 396 MJ and 254 MJ respectively.

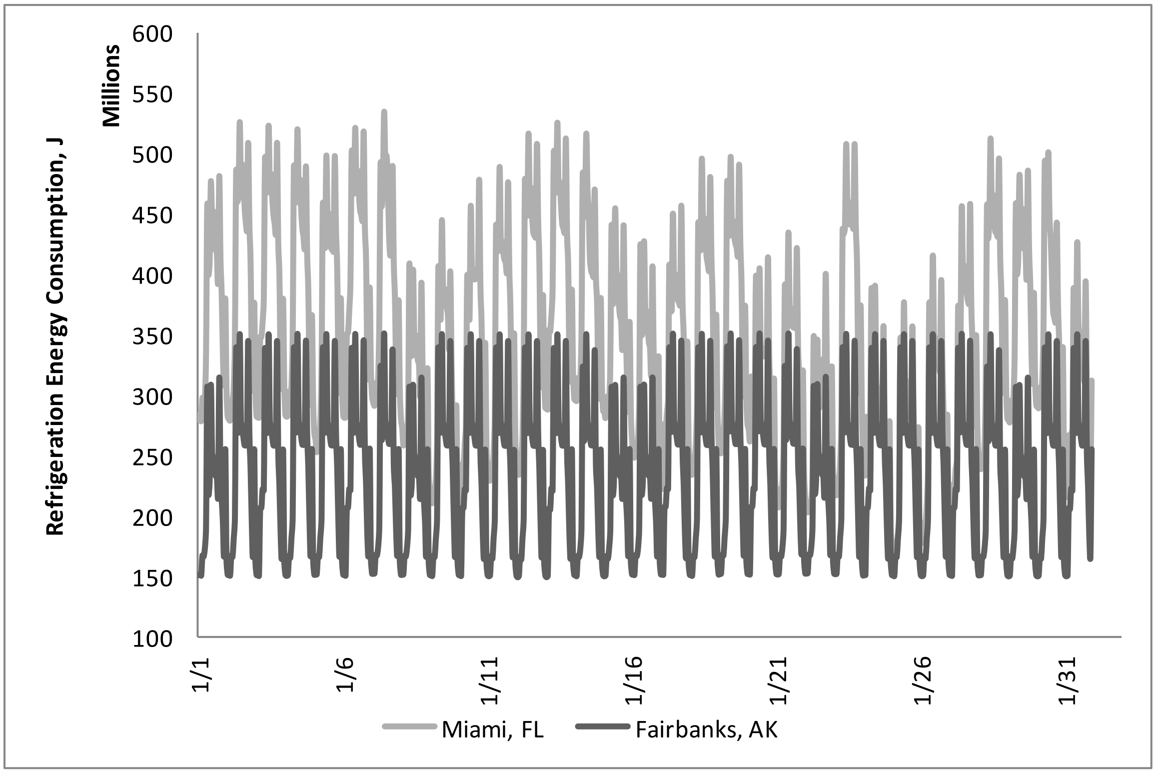

Figure 4: R404A Refrigeration system energy consumption for the month of January in weather extremes: Miami, FL and Fairbanks, AK with an average energy consumption of 348 MJ and 232 MJ respectively.

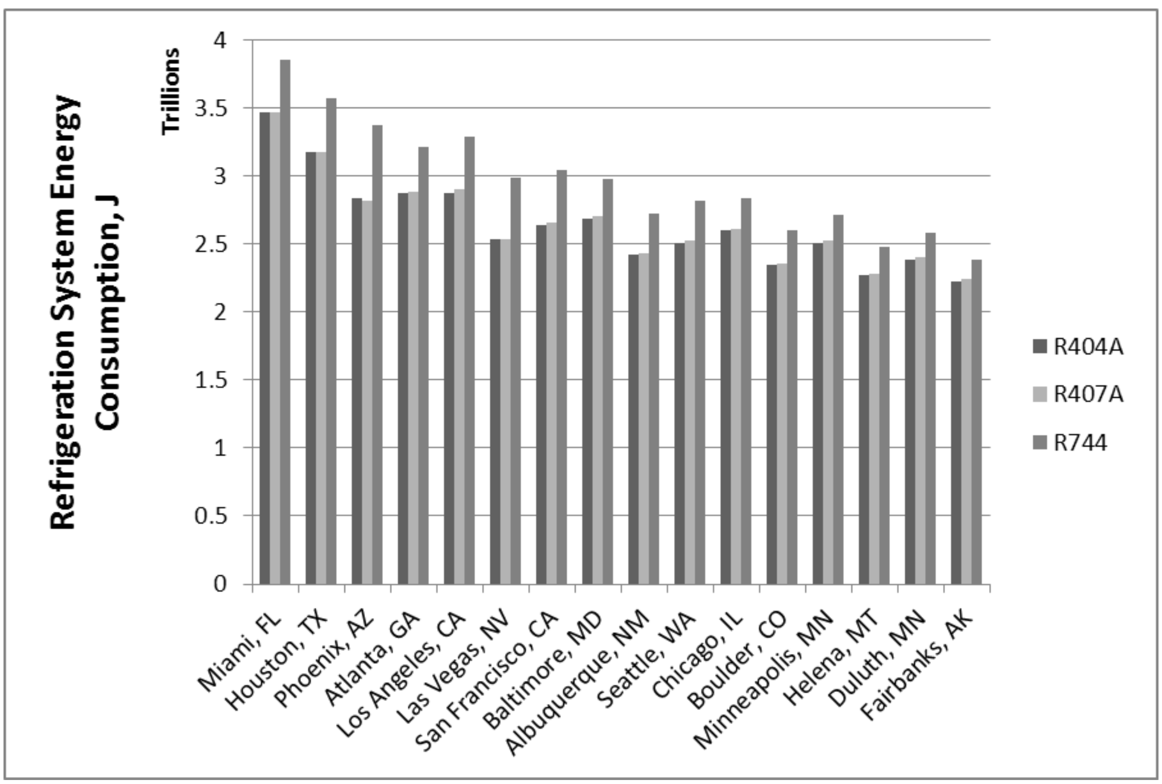

Figure 5: Refrigeration System Annual Energy Consumption.

The hourly energy consumption was then multiplied by the hourly electric emission rate of the corresponding grid interconnect region, resulting in the annual indirect emission. Figure 6 shows the lifetime indirect emissions for the refrigeration systems in the 16 cities. In the same climate, the indirect emissions can vary significantly due to the region emission rate. For example, Baltimore, MD, Albuquerque, NM and Seattle, WA all reside in the same climate zone (zone 4), but Seattle has significantly lower indirect emission rates due to the abundance of hydroelectric power.

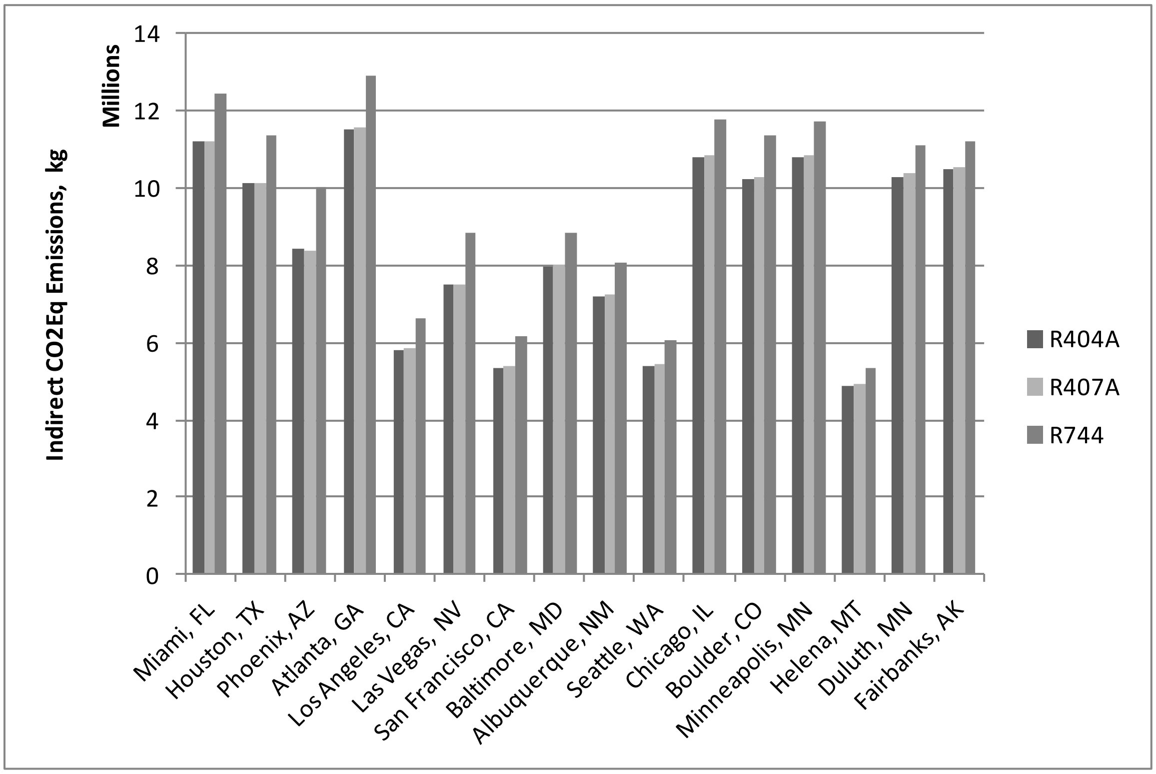

Figure 6: Lifetime Indirect Emissions due to Supermarket Refrigeration System Operation

Direct emissions depend greatly on the maintenance schedule and operating behavior. In this study, a constant leak rate was used for all locations and systems. The annual leak rates were assumed to be 10% with a 5% service leakage rate every other year. It was further assumed that 10% of the refrigerant charge would be lost at the end of the 20 year service life due to recovery inefficiencies. The latest GWP data from the EPA was used: GWP_{R404A} = 3922, GWP_{R407A} = 2107, and GWP_{R744} = 1.

Based on these assumptions, the direct CO_{2eq} emissions for the three refrigerants were estimated to be 33.65, 18.07, and 0.003 Mkg for R404A, R407A, and R744, respectively. The direct emissions of R404A were approximately three times that of the lifetime indirect emissions. This is largely driven by the high GWP of R404A. Although R744 consumed on average 11% more energy than R404A, it had practically no direct emissions. Over the lifetime, R744 has a significant emissions payback. Adding direct and indirect emissions result in the LCCP shown in Figure 7. On average for the 16 cities, R407A and R744 resulted in 37% and 77% less LCCP compared to R404A, respectively. R407A is a current drop-in replacement for R404A with a significant potential for emissions reduction with minimum energy penalty. For the long-term, R744 present the greatest emissions reduction. Even if the technology advances such that a 1% annual leakage rate can be achieved, R744 still presents significant emissions reduction, as shown in Figure 8. On average; R744 system resulted in 14.2% and 4.2% LCCP reduction compared to R404A and R407A respectively. R407A with only 1% annual leakage rate showed to result in lower LCCP than R744 systems in hot climates such as Phoenix, AZ. It is noted that several market barriers exist for R744 which hinder its market penetration. Among these are first cost due to a total system changeout and energy costs as the result of higher energy consumption and peak demand charges.

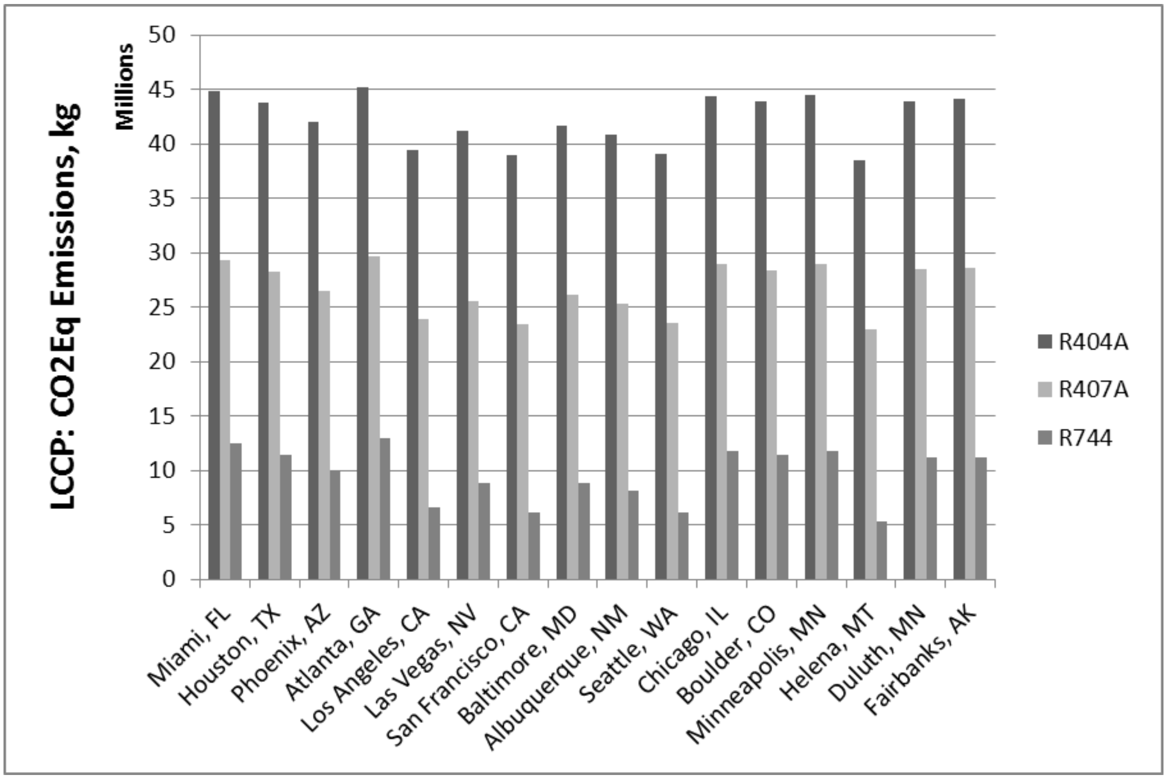

Figure 7: Lifetime CO_{2eq} Emissions for the Supermarket Refrigeration System in the 16 Cities.

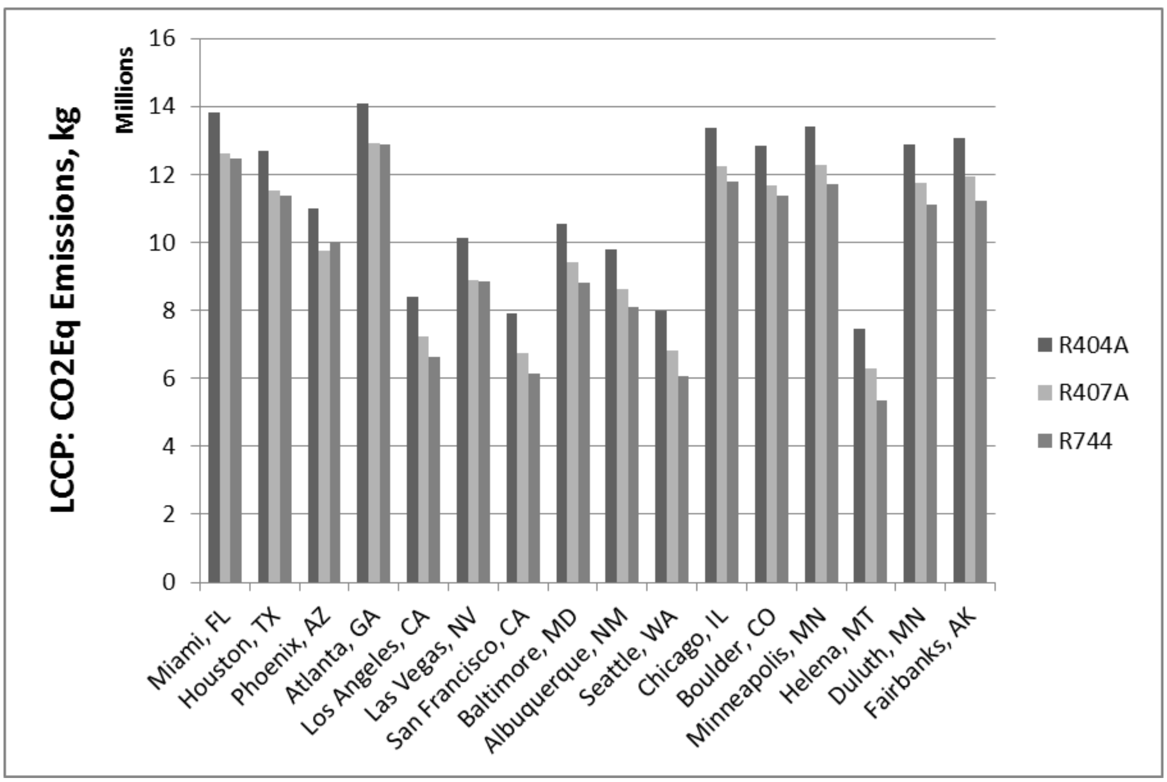

Figure 8: Lifetime CO_{2eq} Emissions for the Supermarket Refrigeration System in the 16 Cities Assuming Annual Leak Rate of Only 1%.

6. CONCLUSIONS

In this study, an LCCP analysis was performed to determine the impact of drop-in and new refrigerant choices. R407A was examined as a drop-in replacement for R404A in the traditional multiplex DX system while the R744 transcritical booster system was investigated as replacement for the HFC multiplex DX system. R407A showed a 37% emission reduction compared to R404A. R407A is an attractive drop-in replacement for R404A and requires minimum system modifications. The R744 transcritical booster system exhibited a 77% emissions reduction compared to R404A. This level of emissions savings cannot be achieved by simply replacing the refrigerant. A total system change is required. Furthermore, the emissions savings comes at the cost of higher energy consumption, which is critical for the operation of supermarket refrigeration systems. While not considered in this study, secondary loop systems with lower GWP refrigerants may also warrant further investigation as a means to achieve significant emissions reductions.

NOMENCLATURE

COP	Coefficient of Performance
GWP	Global Warming Potential
LCCP	Life Cycle Climate Performance
LT	Low Temperature
MT	Medium Temperature

Subscripts

eq	Equivalent

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