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Low Global Warming Potential Refrigerants for Commercial Refrigeration Systems

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Building Technologies Research and Integration Center

LOW GLOBAL WARMING POTENTIAL REFRIGERANTS FOR COMMERCIAL REFRIGERATION SYSTEMS

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ACRONYMS

AHRI	Air-Conditioning, Heating and Refrigeration Institute
AHRTI	Air-Conditioning, Heating and Refrigeration Technology Institute
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BTO	Building Technologies Office (DOE)
CEC	California Energy Commission
CFC	chlorofluorocarbon
COP	coefficient of performance
CO_2	carbon dioxide
CO_{2e}	carbon dioxide equivalent
DOE	U.S. Department of Energy
DTIE	Division of Technology, Industry and Economics (UNEP)
DX	direct expansion
EERE	Energy Efficiency and Renewable Energy (DOE)
EPA	U.S. Environmental Protection Agency
EPR	evaporator pressure regulator
GHG	greenhouse gas
GWP	global warming potential
HCFC	hydrochlorofluorocarbon
HFC	hydrofluorocarbon
HFO	hydrofluoro-olefin
HVAC&R	heating, ventilating, air-conditioning and refrigeration
IIR	International Institute of Refrigeration
IPCC	Intergovernmental Panel on Climate Change
LCCP	life cycle climate performance
LCWI	life cycle warming impact
LT	low-temperature
MAC	mobile air-conditioning
MT	medium-temperature
MTG	multi-disciplinary task group
NREL	National Renewable Energy Laboratory
ODP	ozone depleting potential
ORNL	Oak Ridge National Laboratory
PNNL	Pacific Northwest National Laboratory
RH	relative humidity
ROI	return on investment
TEWI	total equivalent warming impact
TMY	typical meteorological year
UNEP	United Nations Environment Program

EQUATION NOMENCLATURE

COP	coefficient of performance
e_{elec}	carbon dioxide emission factor associated with the generation and distribution of the electrical energy used to power the refrigeration system (kg CO_{2e}/kWh)
$e_{ref,EOL}$	carbon dioxide emission factor associated with the disposal of the refrigerant at the end-of-life of the system (kg CO_{2e}/kg)
$e_{ref,man}$	carbon dioxide emission factor associated with the manufacture of the refrigerant (kg CO_{2e}/kg)
$e_{sys,EOL}$	carbon dioxide emission factor associated with the disposal of a particular material at the end-of-life of the system (kg CO_{2e}/kg)
$e_{sys,man}$	carbon dioxide emission factor associated with the manufacture of that particular material (kg CO_{2e}/kg)
E	annual electrical energy consumption of the refrigeration system (kWh)
E_{accid}	total carbon dioxide equivalent emissions due to refrigerant release caused by accidents (kg CO_{2e})
E_{direct}	direct emission
E_{elec}	emission associated with the generation of electricity used to power the refrigeration system over its operating lifetime (kg CO_{2e})
E_{EOL}	total carbon dioxide equivalent emissions due to refrigerant release at the end-of-life of the system (kg CO_{2e})
$E_{indirect}$	indirect emission
E_{leak}	total carbon dioxide equivalent emissions due to annual refrigerant leakage from the system over its operating lifetime (kg CO_{2e})
$E_{ref,EOL}$	emission associated with the energy to dispose of the refrigerant at the end-of-life of the system (kg CO_{2e})
$E_{ref,man}$	emission associated with the energy to manufacture the refrigerant (kg CO_{2e})
$E_{service}$	total carbon dioxide equivalent emissions due to refrigerant release during servicing events, over the operating lifetime of the system (kg CO_{2e})
$E_{sys,EOL}$	emission associated with the energy to dispose of the refrigeration system components at the end-of-life of the system (kg CO_{2e})
$E_{sys,man}$	emissions associated with the energy to manufacture the refrigeration system (kg CO_{2e})
GWP	global warming potential of the refrigerant (kg CO_{2e}/kg)
GWP_{adp}	global warming potential of the atmospheric degradation products of the refrigerant (kg CO_{2e}/kg)
$h_{c,in}$	average compressor inlet enthalpy determined from measured refrigerant temperature and pressure
$h_{c,out,s}$	average compressor outlet enthalpy assuming isentropic compression between the inlet and outlet pressures
$h_{i,in}$	average refrigerant enthalpy at the inlet of load *i*
$h_{i,out}$	average refrigerant enthalpy at the exit of load *i*
L	lifetime of the system (in years)
$LCCP$	total CO_{2e} emissions
m_c	total refrigerant charge in the system (kg)
$\dot{m}_i$	average refrigerant mass flow through load *i*
m_{sys}	mass of a particular material used in the construction of the refrigeration system (kg)
$\dot{Q}_i$	average refrigeration capacity for load *i*
$\dot{Q}_{total}$	average total refrigeration capacity

$\dot{W}_{i,LT}$	average input power to low-temperature compressor *i*
$\dot{W}_{j,MT}$	average input power to medium-temperature compressor *j*
$\dot{W}_{total}$	average total compressor power
x_{accid}	annual refrigerant loss during accidents (in fraction of total refrigerant charge per year)
x_{EOL}	refrigerant loss occurring at the end-of-life decommissioning of the system (in fraction of total refrigerant charge)
x_{leak}	annual refrigerant leak rate (in fraction of total refrigerant charge per year)
x_{reuse}	fraction of the refrigerant in the system which is reclaimed refrigerant
$x_{service}$	annual refrigerant loss during service events (in fraction of total refrigerant charge per year)
$\eta_{i,th}$	average isentropic efficiency of the i^{th} compressor

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ABSTRACT

Supermarket refrigeration systems account for approximately 50% of supermarket energy use, placing this class of equipment among the highest energy consumers in the commercial building domain. In addition, the commonly used refrigeration system in supermarket applications is the multiplex direct expansion (DX) system, which is prone to refrigerant leaks due to its long lengths of refrigerant piping. This leakage reduces the efficiency of the system and increases the impact of the system on the environment. The high Global Warming Potential (GWP) of the hydrofluorocarbon (HFC) refrigerants commonly used in these systems, coupled with the large refrigerant charge and the high refrigerant leakage rates leads to significant direct emissions of greenhouse gases into the atmosphere.

Environmental concerns are driving regulations for the heating, ventilating, air-conditioning and refrigeration (HVAC&R) industry towards lower GWP alternatives to HFC refrigerants. Existing lower GWP refrigerant alternatives include hydrocarbons, such as propane (R-290) and isobutane (R-600a), as well as carbon dioxide (R-744), ammonia (R-717), and R-32. In addition, new lower GWP refrigerant alternatives are currently being developed by refrigerant manufacturers, including hydrofluoro-olefin (HFO) and unsaturated hydrochlorofluorocarbon (HCFO) refrigerants.

The selection of an appropriate refrigerant for a given refrigeration application should be based on several factors, including the GWP of the refrigerant, the energy consumption of the refrigeration system over its operating lifetime, and leakage of refrigerant over the system lifetime. For example, focusing on energy efficiency alone may overlook the significant environmental impact of refrigerant leakage; while focusing on GWP alone might result in lower efficiency systems that result in higher indirect impact over the equipment lifetime.

Thus, the objective of this Collaborative Research and Development Agreement (CRADA) between Honeywell and the Oak Ridge National Laboratory (ORNL) is to develop a Life Cycle Climate Performance (LCCP) modeling tool for optimally designing HVAC&R equipment with lower life cycle greenhouse gas emissions, and the selection of alternative working fluids that reduce the greenhouse gas emissions of HVAC&R equipment. In addition, an experimental evaluation program is used to measure the coefficient of performance (COP) and refrigerating capacity of various refrigerant candidates, which have differing GWP values, in commercial refrigeration equipment. Through a cooperative effort between industry and government, alternative working fluids will be chosen based on maximum reduction in greenhouse gases at minimal cost impact to the consumer. This project will ultimately result in advancing the goals of reducing greenhouse gas emissions through the use of low GWP working fluids and technologies for HVAC&R and appliance equipment, resulting in cost-competitive products and systems.

An LCCP methodology is presented to determine the direct and indirect emissions associated with the lifetime operation of refrigeration systems, from system construction through system operation and system dismantling. The methodology is incorporated into an open-source LCCP Design Tool which can be used to estimate the lifetime direct and indirect carbon dioxide equivalent gas emissions of various commercial refrigeration system designs and refrigerant options, with the goal of providing guidance on lower GWP refrigerant solutions with improved LCCP compared to baseline systems. The LCCP Design Tool is available in a desktop computer version and a simplified web-based version. Both versions may be accessed from http://lccp.umd.edu/.

The open-source LCCP Design Tool is used to compare the lifetime emissions of four commercial refrigeration system configurations, using four refrigerants (R-404A, R-448A, R-744 [CO_2], and L-40), in six US cities representing different climate zones. The four refrigeration systems include the multiplex direct expansion (DX) system, the cascade/secondary loop system, the transcritical CO_2 booster system, and the medium-temperature secondary loop/low-temperature DX system. Finally, a sensitivity analysis is performed to identify the relative importance of various input parameters on the calculated total CO_2 equivalent emissions of supermarket refrigeration systems.

Comparing the total emissions for different cities suggests that the transcritical CO_2 booster system has the lowest CO_2 equivalent emissions, according to this analysis, in cold and temperate climates. Also,

the R-448A/L-40 secondary circuit refrigeration system was found to offer a good balance between emissions and electricity consumption for hot climates. The parametric analysis showed that shifting towards low GWP refrigerants decreases the effect of the annual leak rate on the total system emissions. Moreover, the sensitivity analysis showed that shifting towards low GWP refrigerants, or more charge conservative systems increases the effect of the hourly emission rate for electricity production on the total system emissions. Finally, an uncertainty analysis was performed showing that using low GWP refrigerants, or more charge conservative systems causes a noticeable drop in the impact of the uncertainty in the inputs related to the direct emissions.

The energy performance of an alternative lower global warming potential refrigerant, R-448A, was evaluated in a laboratory-scale commercial refrigeration system, and its performance was compared to that of the commonly used higher GWP refrigerant, R-404A, found in the commercial refrigeration industry. The laboratory-scale commercial refrigeration system installed in the environmental test chambers at ORNL consists of components typically found in most U.S. supermarket refrigeration systems, including a compressor rack, an air-cooled condenser and several medium-temperature (MT) and low-temperature (LT) refrigerated display cases. The refrigeration system has a low-temperature cooling capacity of approximately 5 tons at −20°F (18 kW at −29°C) and a medium-temperature cooling capacity of approximately 10 to 15 tons at 25°F (35 to 53 kW at −4°C). The compressor rack of the commercial refrigeration system is unique in that it contains two types of compressors that are commonly found in supermarket refrigeration systems: reciprocating compressors and scroll compressors. Having both reciprocating and scroll compressors allows for system performance tests to be performed using the two common compressor types employed in supermarket refrigeration systems.

Using each refrigerant (R-404A and R-448A), the performance of the laboratory-scale commercial refrigeration system was determined at four ambient temperature conditions: 60°F, 75°F, 95°F and 105°F (16°C, 24°C, 35°C, and 41°C). It was found that R-448A and R-404A performed similarly in the laboratory-scale commercial refrigeration system. Compared to R-404A, R-448A exhibited an energy benefit since system COP was increased and compressor power was decreased. The refrigeration capacity of R-448A was found to be similar to that of R-404A. For the same saturated evaporating and condensing temperatures, compressor suction pressures of R-448A were lower than that of R-404A while compressor discharge temperatures of R-448A were higher. Since system performance differences between R-448A and R-404A are small, it can be presumed that R-448A would be a suitable drop-in replacement refrigerant for R-404A. Only minor changes in the system would be required to retrofit R-404A with R-448A. Suction pressure setpoints would need to be reduced and expansion valve superheat settings would require adjustment.

Future efforts related to this project include completing a field evaluation of the lower GWP alternative refrigerant in third-party supermarkets. The main objective of the field evaluation is to compare the energy consumption of refrigeration systems using incumbent refrigerants and alternative refrigerants in actual, operating supermarkets, thereby providing motivation to supermarket owners and operators to retrofit existing systems with the lower GWP alternative. Honeywell and ORNL are currently negotiating the site selection and logistics for the field evaluation of R-448A with a major food retailer.

Motivated by the outstanding energy and environmental performance of the alternative lower GWP refrigerant R-448A, the CRADA partner, Honeywell has commercialized the refrigerant (Solstice® N40) as an alternative to R-404A in both new and existing commercial refrigeration systems.

1. INTRODUCTION

1.1 BACKGROUND

Mechanical vapor compression refrigeration cycles are used to move heat from one location to another via a working fluid (i.e., the refrigerant). The refrigerant typically changes phase as it flows around the refrigeration cycle. The refrigerant absorbs heat from one location, and in doing so, it evaporates. Subsequently, the refrigerant rejects heat to another location, and is condensed during the process. Vapor compression refrigeration cycles can be used for either of two purposes. A *refrigeration system* is used to maintain a space at a lower temperature relative to the ambient temperature while a *heat pump system* is used to maintain a space at a higher temperature relative to the ambient temperature.

1.1.1 Refrigerant Development

The first generation of vapor compression refrigeration systems manufactured in the latter half of the 19th century and the first quarter of the 20th century utilized refrigerants that were both readily available and found to produce the desired cooling effect. The refrigerants of that time included chemicals such as sulfur dioxide, ethyl chloride, methyl chloride and ammonia, among others. However, these refrigerants posed safety hazards due to their toxicity and/or flammability. Beginning in the 1930s, chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants were developed as safe, nontoxic and nonflammable alternatives for the early refrigerants (Midgley and Henne 1930). In the ensuing years, the application of CFCs and HCFCs expanded to include use as aerosol propellants, cleaning agents for the microelectronics industry, and blowing agents for foam insulation. As a result, the quantities of CFCs and HCFCs produced increased rapidly during the 1950s and 1960s.

Subsequently, in the 1970s, scientists discovered that CFC and HCFC refrigerants were contributing to the depletion of the ozone in the earth's stratosphere (Molina and Rowland 1974a, b). It was found that the stability of the CFC and HCFC molecules cause them to remain in the atmosphere for a significant time, and once these molecules migrate into the stratosphere, they dissociate and release chlorine atoms. The chlorine atoms then react with ozone (O_3) to produce diatomic oxygen (O_2), thereby depleting the ozone in the stratosphere.

In 1987, an international treaty, the Montreal Protocol on Substances that Deplete the Ozone Layer, was ratified to protect the ozone layer by requiring the phase-out of numerous substances believed to be responsible for ozone depletion. The Protocol sets a mandatory timetable for the phase out of ozone depleting substances, including CFCs and HCFCs. In the United States, production and importation of CFCs were banned completely in 1996. HCFCs are being phased down, with complete phase-out set for 2030 (ASHRAE 2013).

Following the proposed phase-out of ozone depleting refrigerants per the Montreal Protocol, hydrofluorocarbon (HFC) refrigerants were introduced, and these refrigerants are in common use today. HFC refrigerants contain no chlorine atoms, and thus, they have no ozone depleting potential (ODP). However, HFC refrigerants typically have a high global warming potential (GWP), and thus, release of HFCs into the atmosphere represents a notable source of greenhouse gases. Since HFCs have a high GWP, there is strong interest to minimize the introduction and emissions of HFCs. Alternative refrigerants with lower or near zero GWP are available including hydrocarbons such as propane (R-290) and isobutane (R-600a), ammonia (R-717), carbon dioxide (R-744) and new hydrofluoro-olefin (HFO) refrigerants such as R-1234yf and R-1234ze(E) (UNEP 2010a, b).

1.1.2 Refrigerant Characteristics

The preceding discussion regarding the evolution of refrigerants highlights just a few of the factors affecting refrigerant selection, including safety concerns and environmental concerns. There are several other factors which also influence the selection of refrigerants for specific refrigeration applications, and the following list provides the properties that an ideal refrigerant should possess (ASHRAE 2013; Kuehn, Ramsey, and Threlkeld 1998):

- Latent heat of vaporization: The latent heat of vaporization of a refrigerant is directly related to its refrigerating effect (i.e., its ability to absorb heat) per unit mass. Thus, it is desirable to use refrigerants with a high latent heat of vaporization.
- Heat transfer characteristics: In order to reduce the size of heat exchangers in refrigeration systems, refrigerants should exhibit high heat transfer coefficients. Transport properties such as thermal conductivity, viscosity and density impact the heat transfer characteristics of the refrigerant.
- Critical temperature: Refrigeration systems operate most efficiently when the temperature of the refrigerant remains well below the critical temperature. The cooling capacity and heat rejection of refrigerants with a low critical temperature are greatly reduced as the condensing temperature approaches the critical temperature. In addition, power consumption increases significantly as the system operates near the critical temperature. Thus, refrigerants with high critical temperatures are preferred.
- Evaporating pressure: To ensure that air and moisture do not enter the refrigeration system through leaks in the system, the minimum operating pressure of the system (i.e., the evaporating pressure) should always be greater than atmospheric pressure. If air and moisture are drawn into the system through a leak, system performance will deteriorate. Non-condensable gases such as air in the refrigerant can reduce cooling capacity and system efficiency. Moisture in the system can react with refrigerants to form acids which lead to corrosion and "sludging" of lubricant. Moisture in the refrigerant can also freeze at expansion devices, thereby blocking refrigerant flow.
- Condensing pressure: Since compressor energy consumption is a function of the difference between condensing and evaporating pressures, a low condensing pressure is desired to reduce compressor energy consumption. Also, since components on the high-pressure side of the refrigeration system must withstand the operating pressures without failure, lower condensing pressures require less massive components.
- Inertness and stability: The refrigerant must be chemically inert and stable so as not react with any of the materials within the system, including the metals used for piping and other components, lubricants, plastic components, and elastomers in valves and fittings.
- Oil solubility: The lubricating oil must have good miscibility and solubility with the refrigerant so that the oil returns to the compressor and does not collect in other parts of the system.
- Toxicity: Both the acute (short-term) and chronic (long-term) toxicity of a refrigerant should be considered as they affect human safety during handling and servicing systems with refrigerants, and for occupants in refrigerated or air conditioned spaces. Thus, refrigerants should be non-toxic.
- Flammability: Preferably, a refrigerant should be nonflammable and not burn or support combustion in any concentration with atmospheric air.
- Ozone depletion potential (ODP): The refrigerant should not contribute to the depletion of atmospheric ozone, and thus should have zero ozone depletion potential.
- Global warming potential (GWP): The refrigerant should not contribute to the greenhouse effect or the global warming effect, and thus should have a low global warming potential.
- Detection: The refrigerant should be easily detected by refrigerant gas leak detection equipment.
- Cost: The refrigerant should be readily available at a low cost.
- Low viscosity to ensure minimal pressure drop through the piping, heat exchangers, and other components.

Currently, no single fluid satisfies all of the desirable attributes of an ideal refrigerant, and consequently, a variety of refrigerants are used. Table 1 list the most widely used refrigerants by their application.

Table 1. Commonly used refrigerants by application

Application	Commonly used refrigerants
Domestic refrigeration	R-134a
Commercial refrigeration	R-404A, R-134a, R-744
Industrial refrigeration	R-134a, R-404A, R-22, R-717
Transport refrigeration	R-404A, R-134a
Mobile air-conditioning	R-134a
Residential air-conditioning	R-410A, R-407C
Commercial air-conditioning	R-410A, R-407C
Chillers	R-134a, R-410A, R-407C
Heat pumps for space heating and water heating	R-410A, R-744

1.1.3 Commercial Refrigeration

In this project, attention will be focused on evaluating the greenhouse gas emissions associated with the operation of commercial refrigeration systems, and selecting refrigerants for commercial refrigeration applications which provide energy efficiency with reduced environmental impact.

The traditional multiplex direct expansion (DX) refrigeration system used in commercial applications is prone to significant refrigerant leakage, especially for relatively older existing systems. The EPA (2011) estimates that the U.S. supermarket industry-wide average refrigerant emission rate is approximately 25%. The use of high GWP refrigerants in these systems, combined with high refrigerant leakage, can result in considerable direct carbon dioxide equivalent (CO_{2e}) emissions. In addition, commercial refrigeration systems consume a substantial amount of electrical energy, resulting in high indirect CO_{2e} emissions. Thus, there are ongoing efforts to reduce the direct and indirect environmental impacts of commercial refrigeration systems through the use of leak reduction measures, refrigerant charge minimization, low GWP refrigerants and energy efficiency measures. An example of one such effort to promote these measures in the U.S. is the EPA GreenChill program (EPA 2016). 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. Partner stores in the GreenChill program have annual refrigerant emission rates ranging from ~13-14%, or about half the U.S. average (EPA 2016).

With the phase-out of chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants, manufacturers have turned to hydrofluorocarbons (HFC) as substitutes. The zero ozone-depleting potential of HFCs is appealing; however, many have relatively high GWPs. Thus, environmental concerns are driving regulations and the heating, ventilating, air-conditioning and refrigeration (HVAC&R) industry towards lower GWP alternatives to HFC refrigerants. Existing lower GWP refrigerant alternatives include hydrocarbons, such as propane (R-290) and isobutane (R-600a), as well as carbon dioxide (R-744), ammonia (R-717), and R-32. Note that with the exception of carbon dioxide, all of these existing alternatives are either mildly flammable (American Society of Heating, Refrigerating and Air-Conditioning Engineers [ASHRAE] safety classification 2L for ammonia and R-32) or have higher flammability (ASHRAE safety classification 3 for propane and isobutane). In addition to existing alternatives, new lower GWP refrigerant alternatives are currently being developed by refrigerant manufacturers, including hydrofluoro-olefin (HFO) and unsaturated hydrochlorofluorocarbon (HCFO) refrigerants. These next-generation refrigerants and their blends are typically either non-flammable (ASHRAE safety classification 1) or have lower flammability (ASHRAE safety classification A2L).

The selection of an appropriate refrigerant for a given refrigeration application should be based on several factors, including the GWP of the refrigerant, the energy consumption of the refrigeration system over its operating lifetime, and leakage of refrigerant over the system lifetime. For example, focusing on energy efficiency alone may overlook the significant environmental impact of refrigerant leakage; while focusing on GWP alone might result in lower efficiency systems that result in higher indirect impact over the equipment lifetime.

1.2 PROJECT OBJECTIVES

The objective of this Collaborative Research and Development Agreement (CRADA) between Honeywell and the Oak Ridge National Laboratory (ORNL) is to develop a Life Cycle Climate Performance (LCCP) modeling tool for optimally designing HVAC&R equipment with lower life cycle greenhouse gas emissions, and the selection of alternative working fluids that reduce the greenhouse gas emissions from HVAC&R equipment. In addition, an experimental testing program will be utilized to measure the COP and capacity of various refrigerant candidates, which have differing GWP values, in commercial refrigeration equipment. Through a cooperative effort between industry and government, alternative working fluids will be chosen based on maximum reduction in greenhouse gases at minimal cost impact to the consumer. This project will ultimately result in advancing the goals of reducing greenhouse gas emissions through the use of low GWP working fluids and technologies for HVAC&R and appliance equipment, resulting in cost-competitive products and systems.

The CRADA partner, Honeywell, with established leadership in the development of refrigerants, will develop alternative lower GWP refrigerants for refrigeration, air-conditioning, heat pump, and appliance equipment. The LCCP tool development effort, performed in conjunction with the University of Maryland, will recognize the current national (ASHRAE Multidisciplinary Task Group [MTG] on low GWP refrigerants) and international (International Institute of Refrigeration [IIR] LCCP working party) efforts to standardize the LCCP evaluation procedures as well as recognize the needs of the U.S. HVAC&R stakeholders.

1.3 MOTIVATION

The Department of Energy's (DOE) Building Technologies Office (DOE-BTO) has as its long term goal to create marketable technologies and design approaches that address energy consumption in existing and new buildings. The current vision that DOE-BTO has for achieving this goal involves reducing the energy and carbon emissions used by the energy service equipment (i.e., equipment providing space heating and cooling, water heating, etc.) by 50% compared to today's best common practice. Alternative refrigerants with low global warming potential are needed to achieve DOE's goal of reducing carbon emissions.

Hydrofluorocarbon refrigerants and their blends are essential to the operation of vapor compression equipment which dominates the residential and commercial heating, ventilating, air-conditioning and refrigeration market. As concerns about climate change intensify, it is becoming increasingly clear that suitable alternative refrigerants which have a lower GWP compared to today's commonly used refrigerants will be needed. Previous research efforts by the DOE during the period when ozone depletion was an issue resulted in DOE being a guiding force behind the selection of alternative refrigerants. By taking an early lead in research regarding low GWP refrigerants, the DOE will again have a primary role in determining which alternatives are selected.

Present research efforts by refrigerant suppliers are more focused on the manufacturing processes. In addition, refrigerant suppliers have limited resources and equipment for testing. This project will provide guidance to the HVAC&R community on selecting alternative, energy-efficient, low GWP refrigerants.

1.4 OUTLINE OF REPORT

The structure of this report is as follows:

- Chapter 2 presents the development of the Life Cycle Climate Performance (LCCP) methodology which is used to estimate the lifetime emissions associated with the construction, operation and dismantling of refrigeration systems.
- Chapter 3 presents an LCCP analysis of various refrigeration system designs and refrigerant options.
- Chapter 4 presents the performance evaluation of a laboratory-scale supermarket refrigeration system using the traditional refrigerant, R-404A, and an alternative refrigerant option, R-448A.
- Chapter 5 outlines a planned field study in which the performance of an actual operating supermarket refrigeration system is evaluated with its incumbent refrigerant and the alternative refrigerant, R-448A.
- Finally, Chapter 6 presents concluding remarks.

2. LIFE CYCLE CLIMATE PERFORMANCE

When energy from the sun reaches the Earth, roughly two-thirds of that solar energy is absorbed by the Earth's surface and the atmosphere, while the remaining one-third of the solar energy is reflected back to space (IPCC 2007). In order to balance the incoming solar radiation which is absorbed by the Earth, the Earth must radiate an equivalent amount of energy back to space. As the Earth re-emits radiation in the infrared portion of the spectrum, a significant portion of that energy is absorbed by the atmosphere. This phenomenon is referred to as the greenhouse effect, and it is due to the presence of greenhouse gases in the atmosphere. Greenhouse gases are those gases that can absorb and emit radiation within the thermal infrared range. Greenhouse gases (GHGs) in the earth's atmosphere trap heat near the earth's surface, thereby maintaining the Earth's surface temperature about 60°F warmer than would otherwise be the case if these gases were not present (ASHRAE 2013). Increasing concentrations of GHGs can lead to an increased warming of the Earth.

The most abundant greenhouse gases in Earth's atmosphere include water vapor (H_2O), carbon dioxide (CO_2), methane (CH_4), nitrous oxide (N_2O) and ozone (O_3). The atmospheric concentrations of greenhouse gases are determined by the balance between sources (emissions of greenhouse gases from human activities and natural systems) and sinks (the removal of greenhouse gases from the atmosphere by conversion to different chemical compounds) (Becker 2013).

Different greenhouse gases absorb and trap varying amounts of infrared radiation. They also persist in the atmosphere for differing time periods and influence atmospheric chemistry (especially the ozone) in different ways. The global warming potential of a greenhouse gas depends on both the efficiency of the molecule as a greenhouse gas and its atmospheric lifetime (EPA 2010).

The global warming potential of a GHG is an index describing its relative ability to trap radiant energy compared to carbon dioxide (CO_2), which has a very long atmospheric lifetime (ASHRAE 2013). Quantification of the climate impact of refrigerant emissions is thus often reported in CO_2 equivalent emissions (CO_{2e}). GWP may be calculated for any particular integration time horizon (ITH). Typically, a 100-year ITH is used for regulatory purposes, and may be designated as GWP_{100} (ASHRAE 2013).

Many refrigerants, such as CFCs, HCFCs, HFCs, hydrocarbons and carbon dioxide, are greenhouse gases. Newly developed hydrofluoro-olefin (HFO) refrigerants and their blends are being promoted as lower-GWP alternatives to existing HCFC and HFC refrigerants. These HFOs are also GHGs, but their GWPs are significantly lower than those of HCFCs and HFCs (ASHRAE 2013). Depending upon the specific refrigeration application, direct release of refrigerant into the atmosphere can be a significant source of greenhouse gases. Table 2 lists the GWP values of several common refrigerants.

One source of greenhouse gas emissions associated with the operation of refrigeration systems is direct release of refrigerant from these systems to the atmosphere. Large field-erected refrigeration systems, such as those found in supermarkets or refrigerated warehouses, contain may joints, fittings and components which are prone to leakage over time. This refrigerant leakage can be significant, releasing as much as 25% of the total refrigerant charge per year. On the other hand, small refrigeration systems such as those found in domestic refrigerators, window air conditioners, and beverage vending machines for example, are factory-sealed with only a few fittings and components, resulting in little or no refrigerant leakage over their operating lifetime. The major source of refrigerant emissions for sealed systems occurs at the end-of-life when refrigerant is recovered from the systems. Invariably, some refrigerant is lost to the atmosphere during the recovery process.

Another source of greenhouse gas emissions associated with refrigeration system operation is related to the electrical energy consumed by the systems. The electrical energy that refrigeration systems consume is typically produced from fossil-fuel-fired power plants. The production of this electrical energy results in the emission of CO_2 due to the combustion of the fossil fuel. Depending upon the specific refrigeration application, the indirect greenhouse gas emission associated with energy generation can frequently be much larger than the direct emission due to refrigerant release. Such is the case with small, factory-sealed refrigeration units which exhibit little or no refrigerant leakage over their operating lifetimes. Nearly all of the greenhouse gas emissions for these systems is associated with the electrical energy generation used to power these systems. On the other hand, for larger field-erected systems such as supermarket refrigeration systems, direct emissions due to refrigerant leakage and indirect emissions due to energy generation can equally contribute to the overall lifetime greenhouse gas emissions of these systems.

Table 2. GWP values for selected refrigerants (IPCC 2013; UNEP 2010c)

Refrigerant	GWP (kg CO_{2e}/kg)
R-22	1760
R-32	677
R-123	79
R-125	3170
R-134a	1300
R-143a	4800
R-152a	138
R-290 (propane)	3
R-404A	3943
R-410A	1924
R-600a (isobutane)	4
R-717 (ammonia)	0
R-744 (carbon dioxide)	1
R-1234yf	<1
R-1234ze(E)	<1

Several concepts are often used to evaluate the overall lifetime greenhouse gas impact of refrigeration systems, including the Total Equivalent Warming Impact (TEWI), Life Cycle Climate Performance (LCCP), and Life Cycle Warming Impact (LCWI). These concepts are similar in that they sum the total equivalent direct and indirect greenhouse gas emissions from the refrigeration system, over the lifetime of the system. These concepts are often used to compare different technologies to identify which components should be optimized to effectively reduce global warming. These measures of global warming include the carbon dioxide production resulting from the generation of electrical energy to operate the refrigeration system plus the influence of the refrigerant itself if released into the atmosphere. Consequently, they include the energy efficiency of the refrigeration system.

The Total Equivalent Warming Impact (TEWI) is an index which includes the effects of CO_2 production due to the energy use of the refrigeration system and the effects of the release of refrigerant over the useful life of the refrigeration system. The warming impact associated with CO_2 emissions due to energy use are referred to as indirect effects while the warming impact due to the release of refrigerant is referred to as direct effects (Lommers 2002). The total effect on the environment must be given for a period of time, which is typically 100 years (Baxter, Fischer, and Sand 1998).

The TEWI for a refrigeration system can be estimated from the total amount of refrigerant released during the lifetime of operation of the system and the total energy used by the refrigeration system over its useful lifetime as follows:

$$TEWI = \left(\dot{m}_{refrig}\right)\left(GWP_{refrig}\right) + \alpha E_{annual} L \qquad (1)$$

where $\dot{m}_{refrig}$ is the mass of refrigerant released over the useful life of the system, GWP_{refrig} is the global warming potential of the refrigerant, α is a conversion of energy use into CO_2 emissions, E_{annual}

is the annual energy usage of the system and L is the useful life of the system, in years (Baxter, Fischer, and Sand 1998).

Life Cycle Climate Performance (LCCP) is a method to determine the environmental impact of refrigeration system design, refrigerant selection and system operation over the operating lifetime of the refrigeration system, from system construction through system operation to system dismantling. The environmental impact of the refrigeration system is measured by estimating the system's greenhouse gas emissions in terms of carbon dioxide equivalent emissions. The carbon dioxide equivalent emission is the quantity of carbon dioxide that would have the same GWP as the greenhouse gas emissions of the refrigeration system under consideration (Hafner, Nekså, and Pettersen 2004; Horie et al. 2010; Johnson 2004; Papasavva, Hill, and Andersen 2010; Spatz and Yana Motta 2004; Zhang et al. 2011).

The LCCP concept has been used to compare the overall environmental impact of selected HFCs to other fluids and technologies in applications such as automobile air conditioning, residential and commercial refrigeration, unitary air conditioning, and chillers (ADL 2002). In addition, some tools for system evaluation based on LCCP have been presented in the literature. Papasavva, Hill, and Andersen (2010) developed a comprehensive life cycle analysis tool of alternative Mobile Air Conditioners (MACs), GREEN-MAC LCCP, and presented sample results generated from the tool. However, this tool is limited to LCCP analysis of MACs. Also, an LCCP analysis tool for residential heat pumps was presented by Zhang et al. (2011). This tool was specifically developed for residential heat pump analyses.

LCCP is intended to provide a more comprehensive environmental impact analysis than TEWI, and the LCCP methodology is discussed in further detail in the following sections.

2.1 EMISSIONS CALCULATIONS

In this project, life cycle climate performance (LCCP) will be used to represent the total carbon dioxide equivalent (CO_{2e}) emissions of refrigeration systems, including both the direct and indirect emissions occurring during system operation, system construction and system dismantling. The total carbon dioxide equivalent emissions (CO_{2e}) of a system, $LCCP$, are determined as follows:

$$LCCP = E_{direct} + E_{indirect} \tag{2}$$

where E_{direct} is the direct emission and $E_{indirect}$ is the indirect emission.

The direct emissions from a refrigeration system include those emissions related to the direct release of refrigerant from the system, including annual leakage, refrigerant loss at the end-of-life of the system and refrigerant loss during service events, among others. The indirect emissions from a refrigeration system include those emissions due to the generation of the electricity used to operate the refrigeration system over its lifetime, the emissions due to the manufacture of the materials and refrigerant used in the system, and the emissions associated with the disposal of the system at the end-of-life. The direct and indirect emissions associated with a refrigeration system are determined over the operating life of the system.

2.1.1 Direct Emissions

The direct emissions from a refrigeration system include those emissions related to the direct release of refrigerant from the system, including annual leakage, loss at the end-of-life of the system and loss during service events, among others. The direct emissions, E_{direct}, can be determined as follows:

$$E_{direct} = E_{leak} + E_{service} + E_{accid} + E_{EOL} \tag{3}$$

where E_{leak} is the total carbon dioxide equivalent emissions due to annual refrigerant leakage from the system over its operating lifetime (kg CO_{2e}), $E_{service}$ is the total carbon dioxide equivalent emissions due to refrigerant release during servicing events, over the operating lifetime of the system (kg CO_{2e}), E_{accid}

is the total carbon dioxide equivalent emissions due to refrigerant release caused by accidents (kg CO_{2e}), and E_{EOL} is the total carbon dioxide equivalent emissions due to refrigerant release at the end-of-life of the system (kg CO_{2e}).

By expanding the terms in Equation (3), the direct emissions can be calculated as follows:

$$E_{direct} = m_c \cdot [L \cdot (x_{leak} + x_{service} + x_{accid}) + x_{EOL}] \cdot \left(GWP + GWP_{adp}\right) \tag{4}$$

where m_c is the total refrigerant charge in the system (kg), L is the lifetime of the system (in years), x_{leak} is the annual refrigerant leak rate (in fraction of total refrigerant charge per year), $x_{service}$ is the annual refrigerant loss during service events (in fraction of total refrigerant charge per year), x_{accid} is the annual refrigerant loss during accidents (in fraction of total refrigerant charge per year), x_{EOL} is the refrigerant loss occurring at the end-of-life decommissioning of the system (in fraction of total refrigerant charge), GWP is the global warming potential of the refrigerant (kg CO_2/kg) and GWP_{adp} is the global warming potential of the atmospheric degradation products of the refrigerant (kg CO_2/kg).

The global warming potential, GWP, of a refrigerant is defined as the ratio of the radiative forcing due to the emission of a mass of the refrigerant to that of the emission of the same mass of carbon dioxide over a fixed period of time. Thus, GWP is a measure of the potency of a greenhouse gas relative to carbon dioxide. The global warming potential of a greenhouse gas depends on both the efficiency of the molecule as a greenhouse gas and its atmospheric lifetime (EPA 2010). If a gas has a high radiative forcing but also a short lifetime, it will have a large GWP on a 20-year time scale but a small GWP on a 100-year time scale. Conversely, if a molecule has a longer atmospheric lifetime than CO_2, its GWP will increase with the timescale considered. Carbon dioxide is defined to have a GWP of 1 over all time periods. Table 2 gives the 100-year timeline GWP values for a selection of commonly used refrigerants. A more comprehensive list of GWP values for a wide variety of refrigerants may be obtained from the United Nations Intergovernmental Panel on Climate Change (IPCC 2013).

The adaptive GWP is a measure of the effects of the atmospheric reaction products resulting from the breakdown of the refrigerant in the atmosphere. If known, this value should be included in the calculation of direct emissions.

The annual refrigerant leak rates, including gradual leakage, x_{leak}, service leakage, $x_{service}$, and catastrophic accident leakage, x_{accid}, vary widely depending upon system type, workmanship when the system was manufactured and installed, the quality of maintenance, and proximity of the system to occupational hazards, among others. For supermarket refrigeration systems, the annual gradual leakage rate, x_{leak}, is substantially larger than the refrigerant leakage due to either servicing or accidents. On average, it is reported that the annual refrigerant leakage from supermarket refrigeration systems is approximately 25% (Fricke, Abdelaziz, and Vineyard 2013). However, as more supermarket operators become aware of the issue of refrigerant leakage and implement active refrigerant management programs, annual refrigerant leak rates have begun to decline.

2.1.2 Indirect Emissions

The indirect emissions from a refrigeration system include those emissions due to generation of the electricity used to operate the refrigeration system over its lifetime, the emissions due to the manufacture of the materials and refrigerant used in the system, and the emissions associated with the disposal of the system at the end-of-life. The indirect emissions, $E_{indirect}$, can be determined as follows:

$$E_{indirect} = E_{elec} + E_{sys,man} + E_{sys,EOL} + E_{ref,man} + E_{ref,EOL} \tag{5}$$

where E_{elec} is the emission associated with the generation of electricity used to power the refrigeration system over its operating lifetime (kg CO_{2e}), $E_{sys,man}$ is the emissions associated with the energy to manufacture the refrigeration system (kg CO_{2e}), $E_{sys,EOL}$ is the emission associated with the energy to dispose of the refrigeration system components at the end-of-life of the system (kg CO_{2e}), $E_{ref,man}$ is the emission associated with the energy to manufacture the refrigerant (kg CO_{2e}), and $E_{ref,EOL}$ is the emission associated with the energy to dispose of the refrigerant at the end-of-life of the system (kg CO_{2e}).

By expanding the terms in Equation (5), the indirect emissions can be calculated as follows:

$$
\begin{gathered}
E_{indirect} = L \cdot E \cdot e_{elec} + \sum m_{sys} e_{sys,man} + \sum m_{sys} e_{sys,EOL} \\
+ m_c \cdot (1 + L \cdot x_{leak}) \cdot (1 - x_{reuse}) \cdot e_{ref,man} \\
+ m_c \cdot (1 - x_{EOL}) \cdot e_{ref,EOL}
\end{gathered} \tag{6}
$$

where E is the annual electrical energy consumption of the refrigeration system (kWh), e_{elec} is the carbon dioxide emission factor associated with the generation and distribution of the electrical energy used to power the refrigeration system (kg CO_{2e}/kWh), m_{sys} is the mass of a particular material used in the construction of the refrigeration system (kg), $e_{sys,man}$ is the carbon dioxide emission factor associated with the manufacture of that particular material (kg CO_{2e}/kg), $e_{sys,EOL}$ is the carbon dioxide emission factor associated with the disposal of a particular material at the end-of-life of the system (kg CO_{2e}/kg), x_{reuse} is the fraction of the refrigerant in the system which is reclaimed refrigerant, $e_{ref,man}$ is the carbon dioxide emission factor associated with the manufacture of the refrigerant (kg CO_{2e}/kg), and $e_{ref,EOL}$ is the carbon dioxide emission factor associated with the disposal of the refrigerant at the end-of-life of the system (kg CO_{2e}/kg).

The CO_{2e} emission due to electricity generation is the major contributor to indirect emissions. Note that the emission due to electricity generation depends upon the method of generation as well as temporal factors. Electricity generated from the burning of fossil fuels such as coal and natural gas has substantially higher CO_{2e} emissions as that generated from nuclear power and renewable resources such as solar and wind. Also, electricity consumed at the site may be generated by several sources. Finally, emissions vary by season as well as by time-of-day.

Since the electricity used by a refrigeration system will vary by time-of-day as well as by season, and since emissions associated with electricity generation also vary by time-of-day and season, hourly energy consumption data for the refrigeration system coupled with hourly electricity emission data are required in order to obtain the most accurate estimate of indirect emission due to electricity generation.

The electricity generation emission factors for power generated in various regions of the United States can be obtained from Deru and Torcellini (2007) or the *Hourly Energy Emission Factors for Electricity Generation in the United States* (NREL 2015). These emission factors include the effects of modes of generation as well as season and time-of-day.

Table 3 shows the emission factors associated with the manufacture of 100% virgin and 100% recycled materials that are used in refrigeration systems. Since many materials used today are a mixture of virgin and recycled materials, the actual emission factor associated with manufacturing these materials would lie somewhere between the virgin and recycled values listed in Table 3. Table 4 lists the emission factors for the manufacture of several refrigerants. The values reported in Table 4 are an average of data obtained from various manufactures (IIR 2016).

Table 3. Material manufacturing emissions (IIR 2016)

Material	Virgin material manufacturing emissions ($kg\ CO_{2e}/kg$)	Recycled material manufacturing emissions ($kg\ CO_{2e}/kg$)
Steel	1.8	0.54
Aluminum	12.6	0.63
Copper	3.0	2.46
Plastics	2.8	0.12

Table 4. Refrigerant manufacturing emissions (IIR 2016)

Refrigerant	Manufacturing emissions ($kg\ CO_{2e}/kg$)
R-32	7.2
R-134a	5.0
R-290	0.05
R-404A	16.7
R-410A	10.7
R-1234yf	13.7

2.2 LCCP FRAMEWORK

The LCCP calculation methodology described in Section 2.1 was incorporated into an open-source computer algorithm to create an LCCP design tool. The LCCP calculation algorithm interacts with various system performance models, load models, weather databases and emissions databases to estimate the life-time carbon dioxide equivalent emissions of refrigeration systems. Thus, the LCCP design tool allows refrigeration system designers to choose refrigerants and develop system designs that minimize both greenhouse gas emissions and energy consumption.

The framework for the LCCP computer algorithm is shown in Fig. 1 (Beshr and Aute 2013). The core module in this framework is the open-source *LCCP Calculation Methodology* which is connected to three input modules: the *System Model*, the *Load Model*, and the *Emission & Weather Database*. These various modules interact with each other via standardized communication interfaces that describe the data input-output processes. Due to the modular nature of the framework, any individual module can be replaced with a user-defined module. This creates a highly extensible framework that is suitable for analyzing a variety of systems.

The *System Model* calculates the hourly electrical energy consumption of the refrigeration system at full capacity based on user-supplied details of the refrigeration system in conjunction with weather data. The LCCP design tool can accept system performance data generated by several different systems models or tabular system performance data. In turn, the *Load Model* provides the hourly load values required for the calculation of the actual hourly electrical energy consumption of the system. Again, the LCCP design tool can accept load data generated by several different load models or tabular load data. The actual hourly electrical energy consumption is then multiplied by the hourly emission rate for electricity production, obtained from the *Emission & Weather Database* for the specified location, to obtain the hourly CO_2 emission associated with the energy consumed by the refrigeration system. The default values for the hourly emission rate for electrical energy production at various locations within the U.S. are obtained from Deru and Torcellini (2007). Different building energy modeling tools such as EnergyPlus (DOE 2012) can be used in the *Load Model* to determine the hourly load profile only, or both the hourly load and the system electric energy consumption. In the latter case, a separate system performance model is not required.

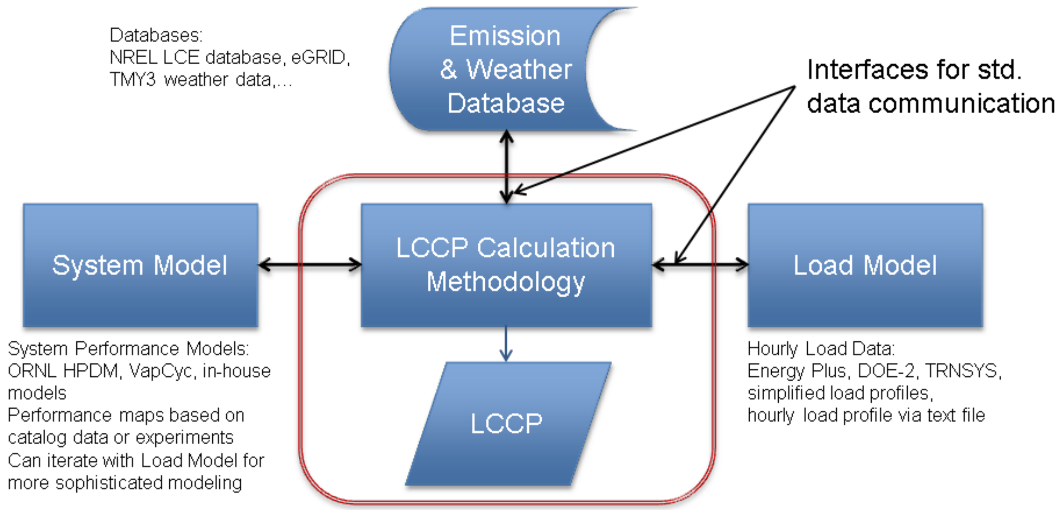

Fig. 1. Framework for open-source LCCP computer algorithm.

The default weather data contained in the *Emission & Weather Database* is obtained from the Typical Meteorological Year (TMY) data found in the National Solar Radiation Data Base (NREL 2012). The weather datasets in the *Emission & Weather Database* include hourly values for dry-bulb temperature, dew-point temperature, and relative humidity. The LCCP design tool contains weather data for 47 cities, with the capability of adding weather data for user-defined cities.

The default GWP values for various refrigerants used in the LCCP design tool are based on a 100-year time horizon (GWP_{100}) and are obtained from the IPCC Fifth Assessment Report (IPCC 2013). The GWP values of other refrigerants which are not listed in the IPCC Fifth Assessment Report were obtained from Zhang et al. (2011), based on values provided by manufacturers, or compiled from publicly available information.

2.3 LCCP DESIGN TOOL

A computerized, open-source LCCP Design Tool was developed, based on the framework outlined in Section 2.2 and the calculation methodology described in Section 2.1. This LCCP design tool can estimate the lifetime direct and indirect carbon dioxide equivalent gas emissions of various commercial refrigeration system designs and refrigerant options, with the goal of providing guidance on lower GWP refrigerant solutions with improved LCCP compared to baseline systems. The carbon dioxide equivalent emissions can be evaluated throughout the refrigeration system's entire lifetime, from system construction, system operation and system decommissioning and dismantling.

The LCCP Design Tool is available in a desktop computer version and a simplified web-based version. Both versions may be accessed from the following website:

http://lccp.umd.edu/

A video demonstrating the main features and capabilities of the LCCP design tool and a general discussion board about LCCP are also available at this website.

2.3.1 Desktop LCCP Design Tool

The main window of the Desktop LCCP Design Tool is shown in Fig. 2. From this window, users can input detailed information regarding the particular application and specify how the energy simulation and load modeling will be performed. Once all the required input data is entered, the LCCP analysis can be initiated and the results can be viewed from the main window.

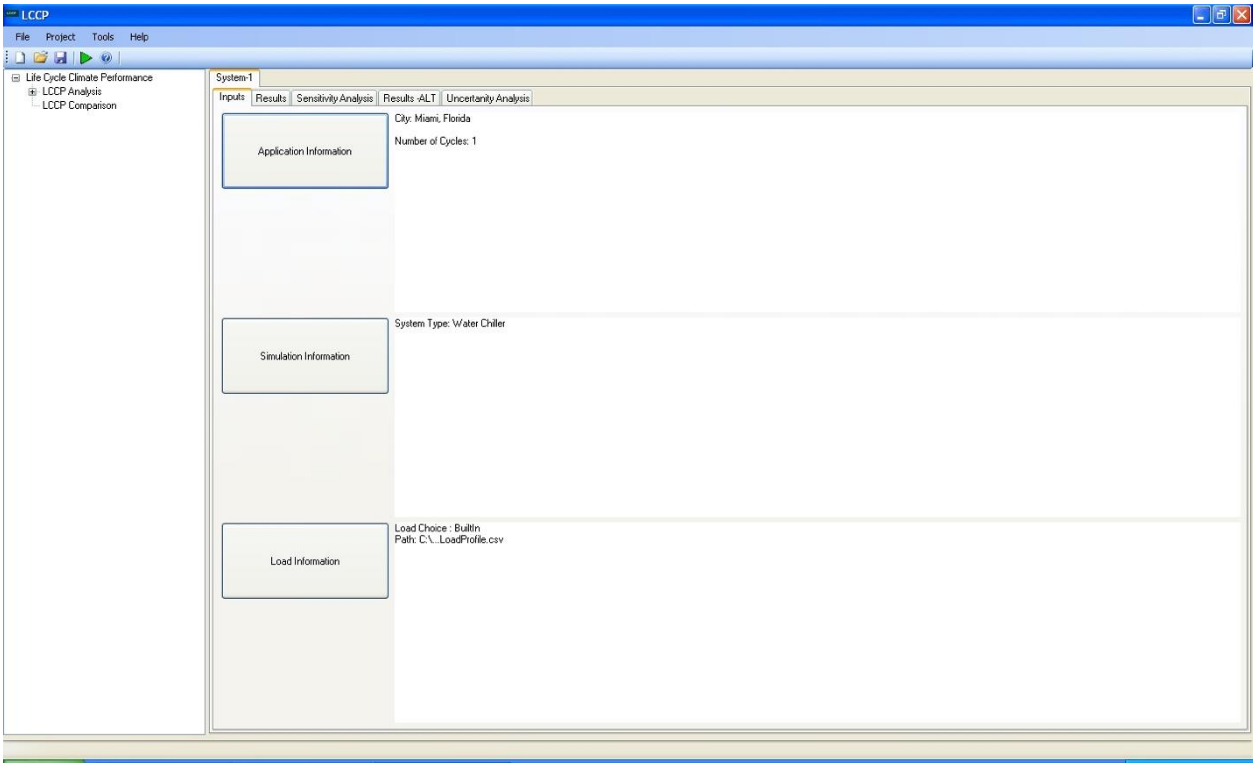

Fig. 2. Desktop LCCP Design Tool main input window.

In the *Application Information* window, shown in Fig. 3, details of the refrigeration or heat pump system to be modeled are specified. System types which can be modeled include centralized direct expansion (DX) supermarket refrigeration systems, secondary loop supermarket refrigeration systems, air-source heat pumps, and water chillers. The total refrigerant charge, annual refrigerant leakage rate, and mass of the system are also specified in the *Application Information* window. In addition, the location of the system is specified and the operating lifetime of the system is specified. Based on the location, the corresponding weather data and emissions values will be obtained from the TMY data files. Finally, default values for refrigerant GWP and CO_{2e} emissions associated with the production of various materials are provided, and the user may adjust these values as necessary.

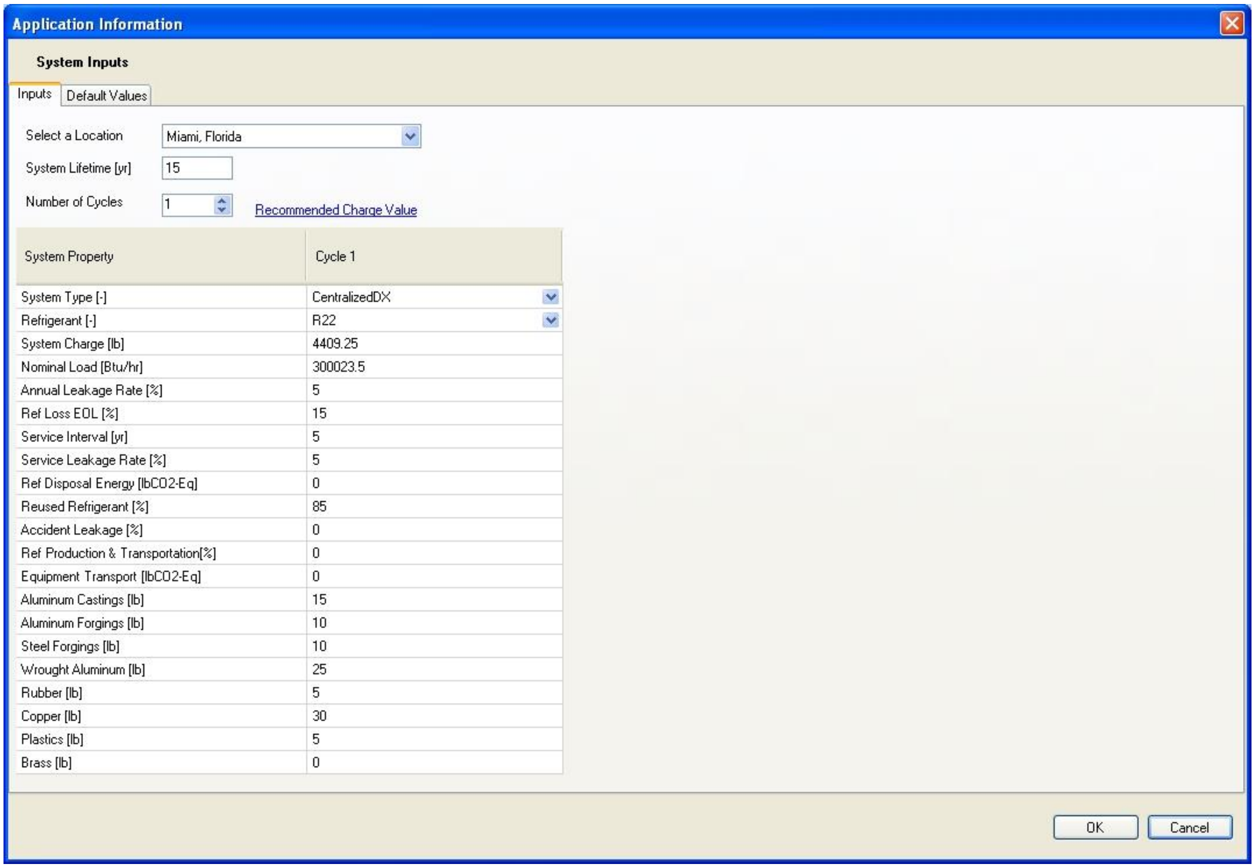

Fig. 3. Application Information input window.

In the *Simulation Information* window, shown in Fig. 4, the user specifies the path to the energy simulation tool executable file which is to be used to calculate the annual energy consumption of the refrigeration or heat pump system. The user also provides the command line prompt which is used to initiate the simulation software executable along with any required input arguments. Energy models which can be used include publically available models such as EnergyPlus or proprietary models developed by the user.

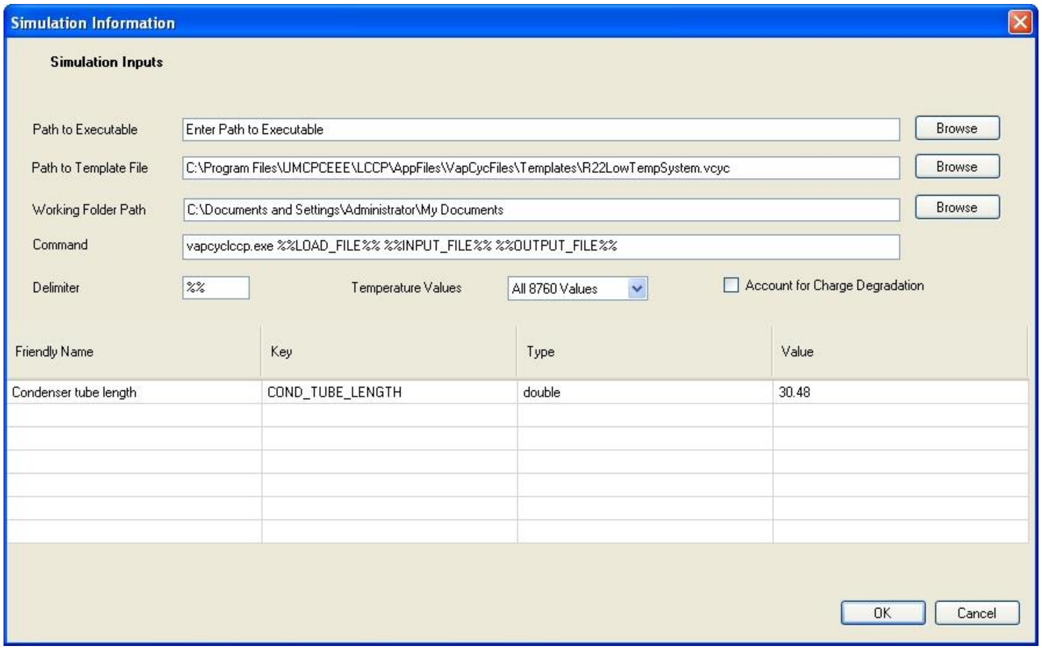

Fig. 4. Simulation Information input window.

In the *Load Information* input window, shown in Fig. 5, the user specifies the hourly annual load profile that the refrigeration or heat pump system is subjected to. The load profile is then used in conjunction with the energy simulation tool to determine the hourly energy consumption of the refrigeration or heat pump system.

The Desktop LCCP Design Tool contains a built-in load profile which is suitable for commercial refrigeration applications. Alternatively, the user may specify the load profile via a user-supplied text file. This text file should be in a comma separated value format (.csv) with the first column containing the hours of the day and the second column containing the corresponding load values. Finally, the load profile may be determined using EnergyPlus. For this method, the user specifies the paths to the files required to execute EnergyPlus. When selecting this option, EnergyPlus can be used to obtain only the hourly load values (and then a separate system simulation software is used to calculate the system's hourly energy consumption) or EnergyPlus can be used to obtain both the hourly load and the system's hourly power consumption (without using a separate energy system simulation software).

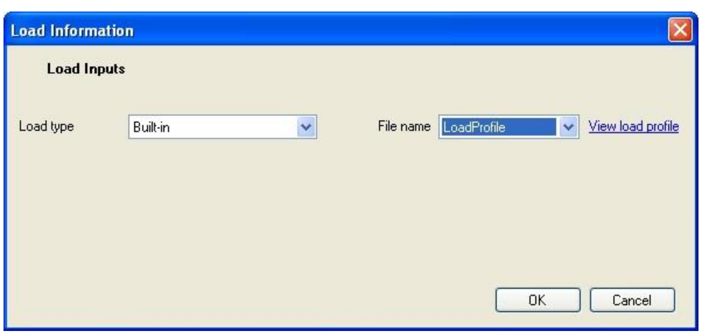

Fig. 5. Load Information input window.

After all of the required input data has been entered into the *Application Information*, *Simulation Information* and *Load Information* windows, the user initiates the LCCP calculation by pressing the green arrow button found on the main window of the LCCP Design Tool. After the calculations are complete, the results may be viewed by pressing the Results tab found on the main window of the LCCP Design tool.

A sample Results window is shown in Fig. 6. Details of the direct and indirect emissions are provided in tabular form, and the results are also summarized graphically in the form of pie charts. The graphical summary may be copied and pasted into word processing or presentation documents.

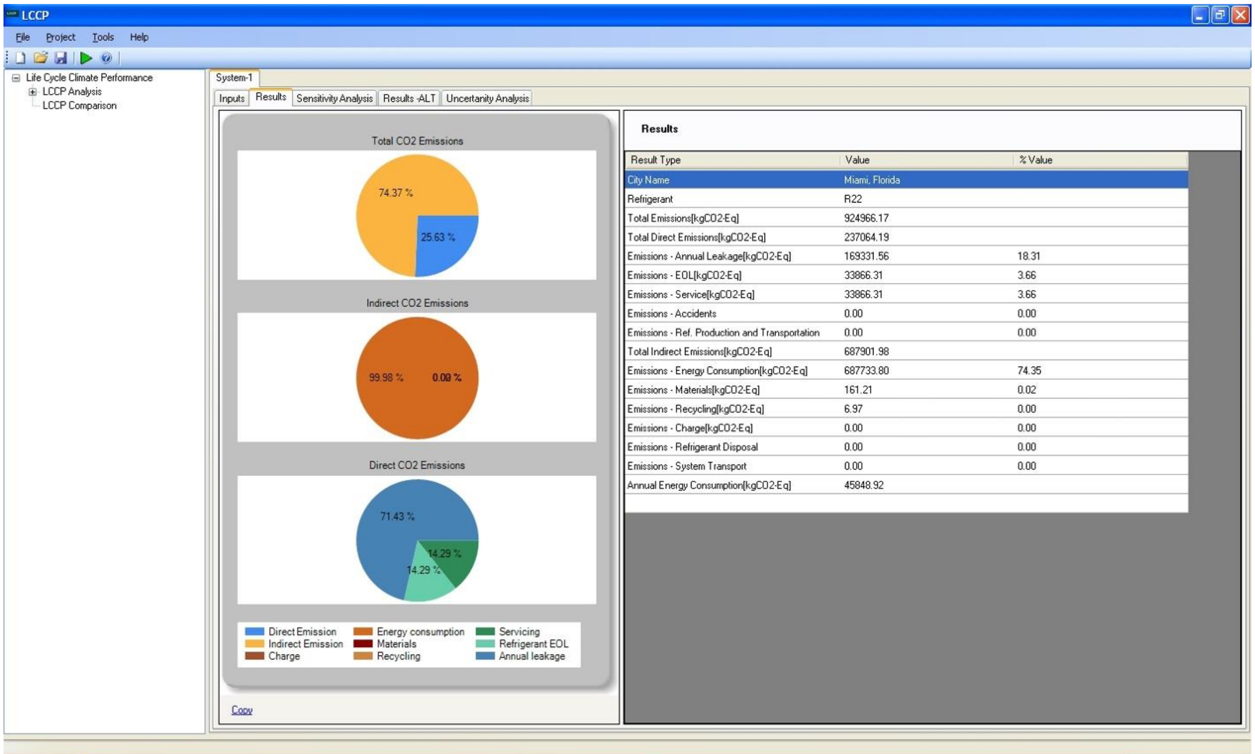

Fig. 6. Desktop LCCP Design Tool results window.

Finally, the Desktop LCCP Design Tool allows users to perform sensitivity analyses and uncertainty analyses. A sensitivity analysis can be used to determine the effect that a change in one or more of the input variables has on the resulting total CO_{2e} emissions of the system. Furthermore, since many of the input variables used in the LCCP analysis are not know precisely, an analysis can be performed in which the uncertainty of each of the various input parameters can be specified, and a bound on the resulting total CO_{2e} emissions of the system can be determined.

2.3.2 Web-based LCCP Design Tool

While the desktop version of the LCCP design tool allows for the detailed analysis of any HVAC&R system type, the web-based LCCP design tool allows users to more easily determine the lifetime carbon dioxide equivalent emissions from a few pre-selected system types, such as supermarket refrigeration systems and residential heat pump systems. The input screen of the web-based LCCP Design Tool is shown in Fig. 7 for a medium-temperature direct expansion (DX) refrigeration system for supermarket applications. From this screen, users can input detailed information regarding the refrigeration system to be modeled, including the following:

- System location
- System lifetime
- Type of refrigerant and refrigerant charge
- Annual refrigerant leak rate and refrigerant service leakage rate

- Degree of subcooling at the expansion device
- Degree of superheat at the evaporator outlet
- Nominal load
- Compressor efficiency and compressor displacement
- Condenser and evaporator tube diameter and length
- Air-side and refrigerant-side heat transfer coefficients

The results of the LCCP analysis are displayed in the output screen, an example of which is shown in Fig. 8. Summary plots show the breakdown of direct and indirect emissions as well as the components which make up the direct and indirect emissions. The detailed emissions data is also present in tabular form.

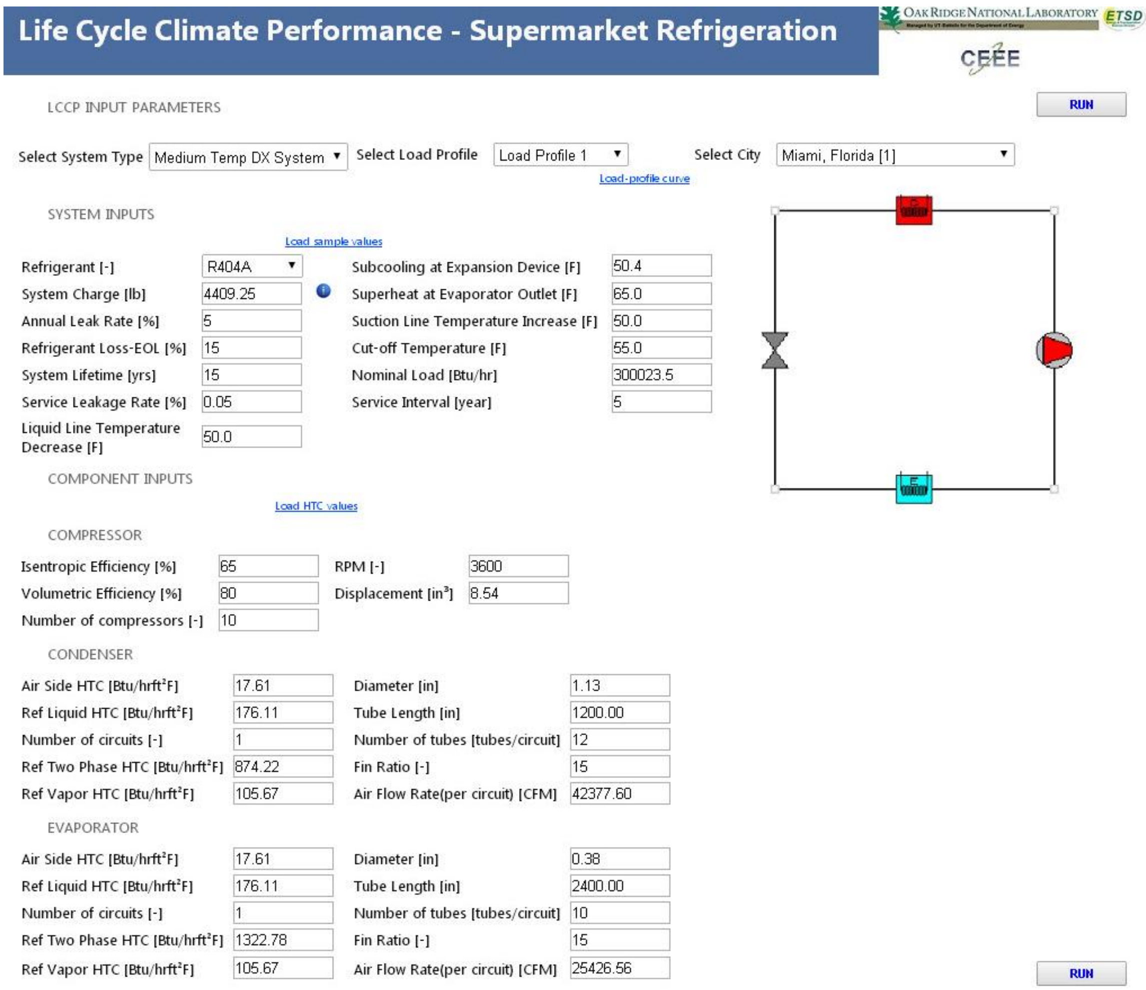

Fig. 7. Web-based LCCP Design Tool data input screen.

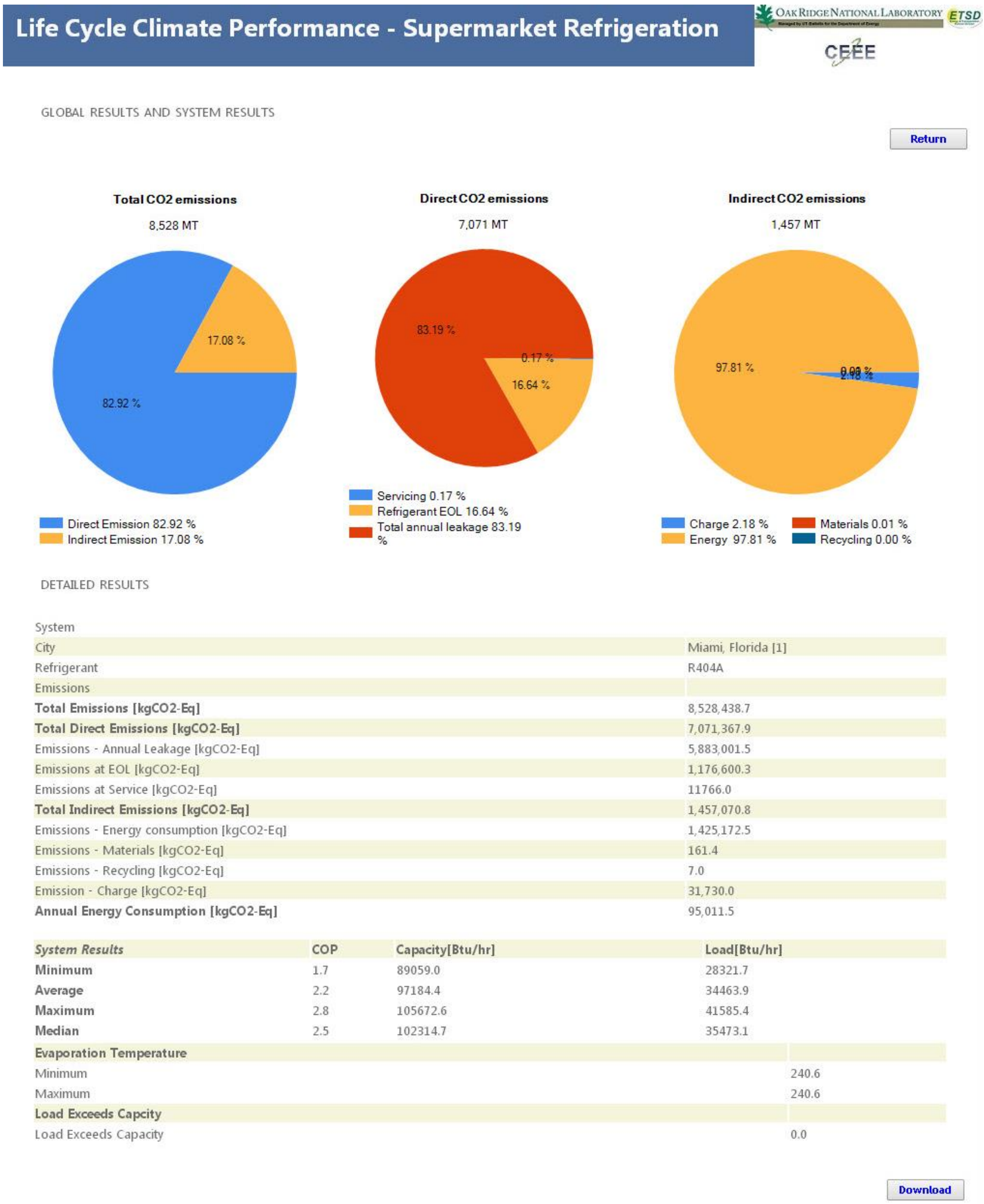

Fig. 8. Web-based LCCP Design Tool data output screen.

3. LCCP ANALYSIS OF REFRIGERATION SYSTEMS

The LCCP Design Tool described in Chapter 2 was used to determine the life-time carbon dioxide equivalent emissions from a variety of supermarket refrigeration system types using a variety of refrigerant options. The LCCP analyses are presented in several publications (Abdelaziz, Fricke, and Vineyard 2012; Fricke, Abdelaziz, and Vineyard 2013; Beshr, Aute, Fricke, et al. 2014; Beshr, Aute, Abdelaziz, et al. 2014; Beshr, Aute, Sharma, et al. 2014; Beshr et al. 2015). A summary of these analyses is provided in this section.

Because supermarket refrigeration systems typically have significant refrigerant charge leakage and high electricity consumption, they generate considerable direct and indirect greenhouse gas emissions. A typical 35,000 ft^2 (3300 m^2) grocery store can consume two to three million kilowatt-hours of energy which results in significant indirect CO_{2e} emissions. In addition, a typical supermarket with the traditional multiplex direct expansion (DX) refrigeration system requires approximately 3000 to 5000 lb (1400 to 2300 kg) of refrigerant, and the refrigeration system can have an average annual refrigerant charge loss of 30% (Faramarzi and Walker 2004). High or moderate GWP refrigerants in conjunction with these high refrigerant leak rates contributes to significant direct CO_{2e} emissions. Methods for reducing the negative environmental impact of commercial refrigeration systems include using low or zero GWP refrigerants, improving the energy efficiency of the refrigeration systems, and designing refrigeration systems with one of the primary performance criterion being to minimize the environmental impact.

Many efforts are underway to develop suitable low GWP alternative refrigerants for use in commercial refrigeration. The synthetic alternative refrigerants proposed thus far are mostly blends comprised of R-32, R-1234yf and/or R-1234ze(E), along with other HFC refrigerants, in an effort to obtain a balance between low GWP, affordability, safety, and system efficiency. One of the promising refrigerants for commercial systems is the zeotropic blend R-448A, which shows competitive performance with, and much lower GWP than, R-404A (Yana Motta and Spatz 2012).

The open-source LCCP Design Tool discussed in Chapter 2 was used to compare the LCCP of four commercial refrigeration system configurations, using four refrigerants (R-404A, R-448A, R-744 [CO_2], and L-40), in six US cities representing different climate zones. The four refrigeration systems include the multiplex direct expansion (DX) system, the cascade/secondary loop system, the transcritical CO_2 booster system, and the medium-temperature secondary loop/low-temperature DX system. Finally, a sensitivity analysis was performed to identify the relative importance of various input parameters on the calculated total CO_{2e} emissions of supermarket refrigeration systems.

3.1 ENERGY MODELING

To determine the emissions associated with the energy consumption of the refrigeration systems over their lifetimes, EnergyPlus (DOE 2012) was used to model a typical supermarket in relevant U.S. locations to incorporate the climate significance. EnergyPlus simulations provide estimates of the hourly electric energy consumption of the commercial refrigeration systems. The resulting hourly electric consumption was then multiplied by the hourly electric emission rate for the relevant interconnect region based on data published by Deru and Torcellini (2007). The annual emissions were then multiplied by the system lifetime to estimate the total indirect emissions.

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. 2011). This single-story supermarket model has a floor area of 45,000 ft^2 (4,181 m^2) with a floor-to-ceiling height of 20 ft (6.1 m), and is divided into six zones (office, dry storage, deli, sales, produce and bakery), as shown in Fig. 9. 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. Heating and air-conditioning are 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 5 and 6, respectively. These tables report the size, cooling capacity, power input (for evaporator fans, lighting, defrost heaters and anti-sweat heaters) and operating temperature of the display cases and walk-ins, based on representative manufacturers' information.

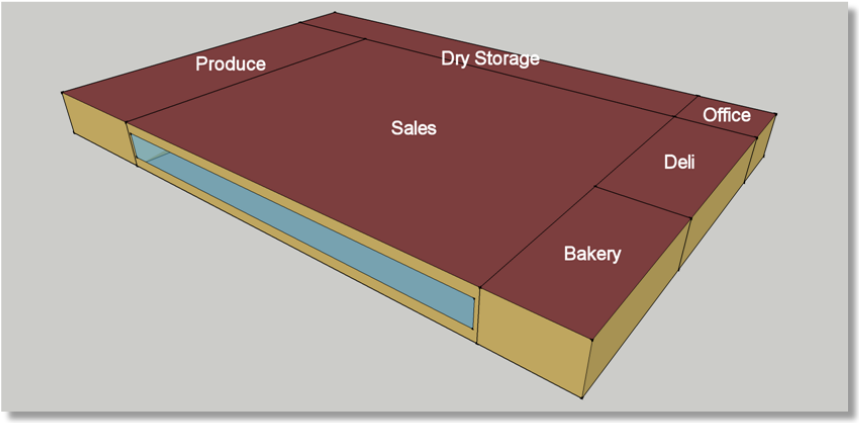

Fig. 9. EnergyPlus supermarket model.

Table 5. Refrigerated display case specifications used in supermarket energy modeling (Beshr et al. 2015)

Case Type	Length, ft (m)	Cooling Capacity, Btu/(h·ft) (W/m)	Evaporator Fan, W/ft (W/m)	Lighting, W/ft (W/m)	Defrost, W/ft (W/m)	Anti-Sweat Heater, W/ft (W/m)	Case Temperature, °F (°C)
Medium-Temperature							
Multi-deck meat	120 (36.6)	1500 (1442)	27 (88)	12 (39)	135 (443)	20 (66)	36 (2.2)
Multi-deck other	260 (79.2)	1500 (1442)	12 (41)	18 (60)	0	0	36 (2.2)
Low-Temperature							
Reach-in	268 (81.7)	560 (538)	20 (66)	33 (108)	400 (1312)	71 (233)	5.0 (-15.0)
Single-level open	128 (39.0)	550 (529)	10 (33)	0	420 (1378)	24 (79)	10 (-12.0)

Table 6. Refrigerated walk-in cooler/freezer specifications used in supermarket energy modeling (Beshr et al. 2015)

Walk-In Type	Area, ft^2 (m^2)	Cooling Capacity, $Btu/(h \cdot ft^2)$ (W/m^2)	Evaporator Fan, W/ft^2 (W/m^2)	Lighting, W/ft^2 (W/m^2)	Defrost, W/ft^2 (W/m^2)	Anti-Sweat Heater, W/ft^2 (W/m^2)	Walk-In Temperature, °F (°C)
Medium-Temperature							
Meat Cooler	400 (37.2)	60 (190)	4 (40)	1 (11)	20 (210)	0	36 (2.2)
Food Cooler	2600 (241.5)	60 (190)	2 (26)	1 (11)	0	0	36 (2.2)
Low-Temperature							
Food Freezer	1000 (92.9)	79 (250)	4 (43)	1 (11)	29 (310)	0	-10 (-23.3)

To determine the significance of climate on refrigeration system performance, the energy consumption of the model supermarket was estimated for six U.S. cities using EnergyPlus. These six cities, shown in Table 7, are representative of several of the climate zones in the U.S. (Baechler et al. 2010).

Table 7. Climate zones and cities used in the LCCP analysis

Climate Zone	City	Annual Average Temperature, °F (°C)	Average Hourly Emission Rate for Electricity Production, lb CO_2/kWh (kg CO_2/kWh)
1A	Miami, FL	76.8 (24.9)	1.49 (0.678)
2B	Phoenix, AZ	74.8 (23.8)	1.67 (0.757)
3B	Los Angeles, CA	63.1 (17.3)	0.77 (0.351)
4B	Albuquerque, NM	57.6 (14.2)	2.43 (1.10)
5A	Chicago, IL	50.0 (10.0)	1.41 (0.638)
8	Fairbanks, AK	28.2 (-2.1)	1.71 (0.774)

3.2 REFRIGERATION SYSTEMS

In this study, four different commercial refrigeration system designs were investigated that serve the medium-temperature and low-temperature loads of the model supermarket described in Section 3.1.

3.2.1 Multiplex Direct Expansion (DX) System (S1)

The first refrigeration system (designated as S1) is the typical multiplex direct expansion (DX) supermarket refrigeration system, as shown in Fig. 10. This refrigeration system consists of a combined medium-temperature (MT) and low-temperature (LT) direct expansion (DX) compressor rack with mechanical subcooling and an air-cooled condenser. Two such systems are used to satisfy the refrigeration loads of the model supermarket, and the refrigeration load and refrigerant charge for each system is summarized in Table 8. The compressors used for the two racks were modeled based on commercially available compressors with published performance maps.

In this study, the multiplex DX system using R-404A as the refrigerant is considered to be the baseline for comparison. The alternative refrigerant, R-448A, was also considered for use in this system, since R-448A has a GWP which is nearly two-thirds less than that of R-404A, and it is designed to be a near drop-in replacement for R-404A.

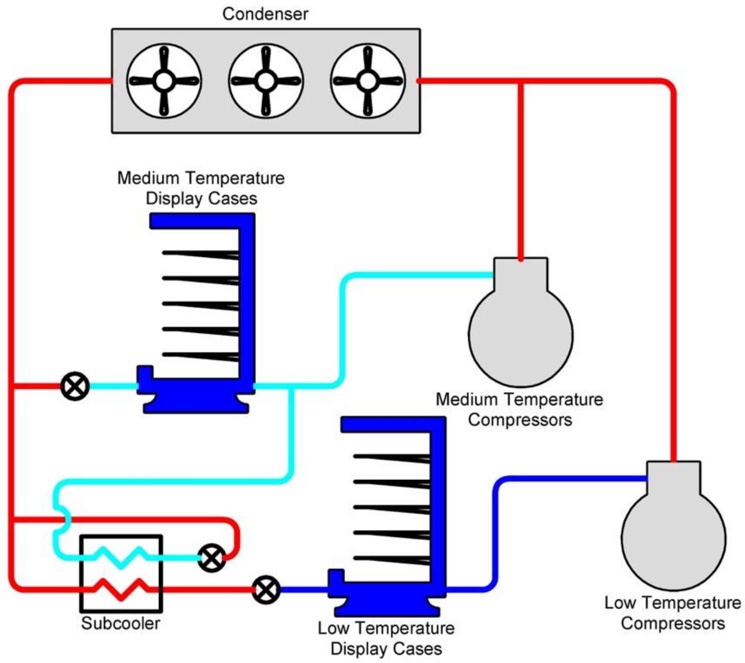

Fig. 10. Multiplex direct expansion (DX) refrigeration system (System S1).

Table 8. Multiplex DX refrigeration system configuration (System S1) (Beshr et al. 2015)

Compressor rack	MT capacity, Btu/h (kW)	LT capacity, Btu/h (kW)	Refrigerant charge, lb (kg)†
1	570 000 (167.1)	220 000 (64.6)	5050 (2290)
2	180 000 (52.8)	80 000 (23.4)	1650 (750)

† Charge values are estimated, based on data provided by PG&E (2011).

3.2.2 Cascade/Secondary Loop System (S2) using R-448A and CO_2

The second system (designated as S2) considered in this study was a cascade/secondary loop system, shown in Fig. 11, in which R-448A refrigerant is used in the high-temperature circuit of the cascade system and R-744 is used as the refrigerant in the low-temperature circuit of the cascade system. The LT loads are served by direct expansion of R-744 in the low-temperature circuit while the MT loads are served by a pumped liquid R-744 secondary loop, which is chilled by the R-448A circuit. In this study, two air-cooled cascade/secondary loop systems were modeled to satisfy the loads of the model supermarket, and the refrigeration load and refrigerant charge for each system are summarized in Table 9. Compressor maps for commercially available HFC and R-744 compressors were used for the energy simulations.

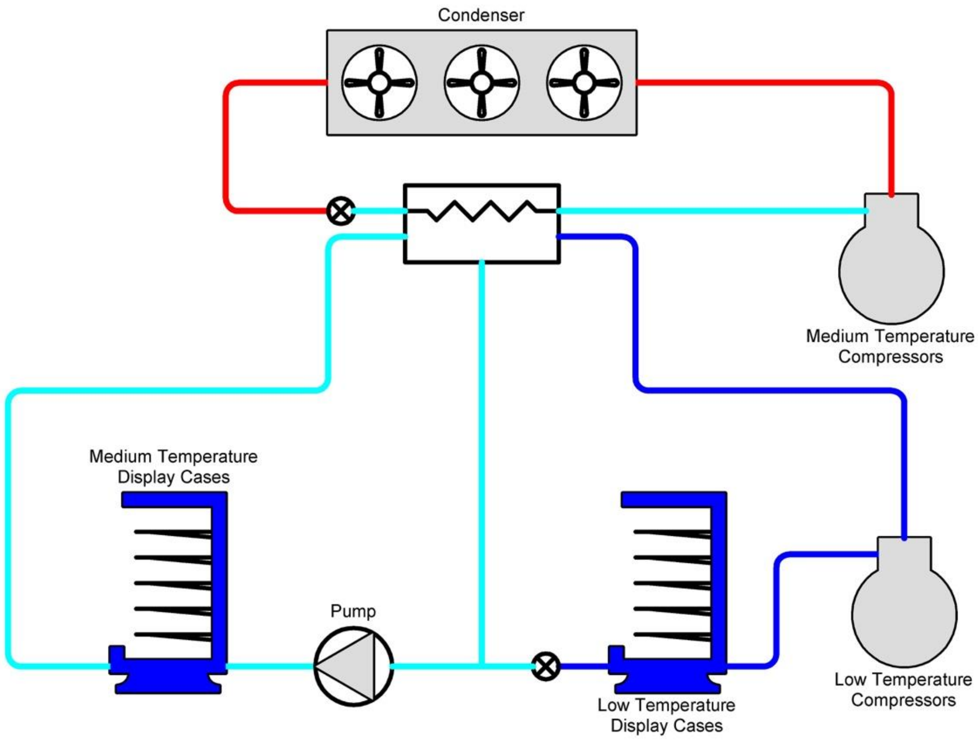

Fig. 11. Cascade/secondary loop system (System S2) using R-448A and CO_2.

Table 9. Cascade/secondary loop system (System S2) using R-448A and R-744 (Beshr et al. 2015)

Compressor rack	MT capacity, Btu/h (kW)	LT capacity, Btu/h (kW)	Refrigerant charge, lb (kg)
1	405 000 (118.6)	220 000 (64.6)	HFC: 2170 (985) R-744: 1150 (520)
2	345 000 (101.2)	80 000 (23.4)	HFC: 1850 (840) R-744: 420 (190)

3.2.3 Transcritical R-744 (CO_2) Booster System (S3)

The third system (designated as S3) considered in this study was the transcritical R-744 (CO_2) booster configuration, as shown in Fig. 12, where both the MT and LT loads are served by direct expansion of CO_2. The heat rejection from the system occurs either supercritically or subcritically, depending upon the outdoor ambient temperature, through an air-cooled gas cooler. Two such systems are used to satisfy the refrigeration loads of the model supermarket, and the refrigeration load and refrigerant charge for each system are summarized in Table 10.

When the compressor discharge conditions are such that the R-744 is in the supercritical region, then the high-side operating pressure is independent of the gas cooler exit temperature (Sawalha 2008). For a given gas cooler exit temperature, there is an optimum pressure to achieve the maximum coefficient of performance (COP) (Ge and Tassou 2011). Several researchers have developed correlations to determine the optimum high-side pressure in transcritical R-744 refrigeration systems, and in the energy simulations performed in this study, the optimal gas pressure control based on the correlation of Ge and Tassou (2011) was used.

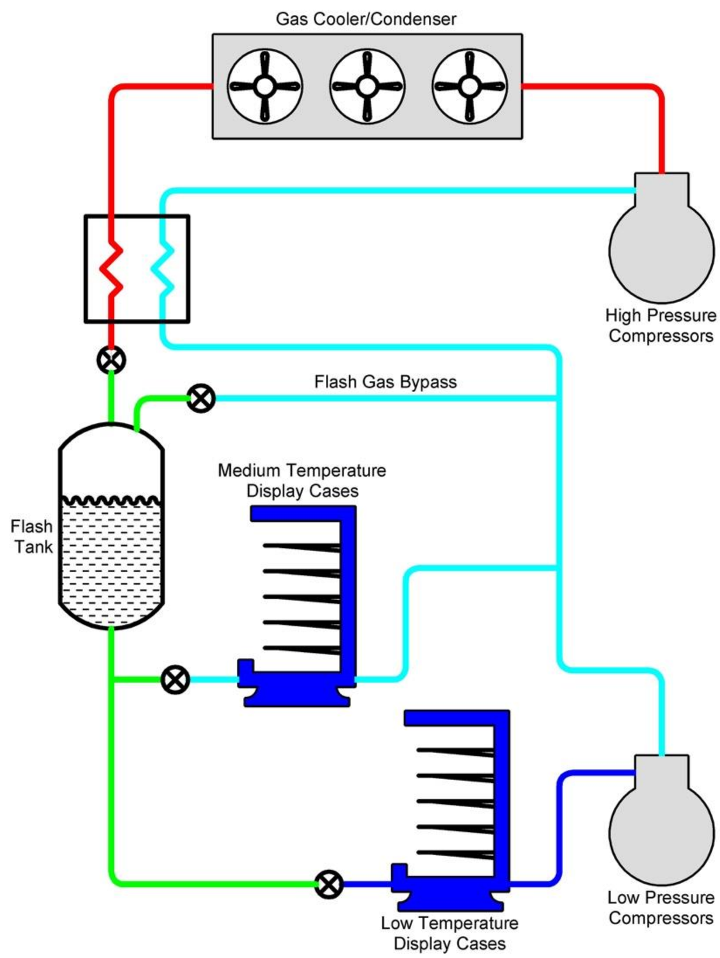

Fig. 12. Transcritical R-744 (CO_2) booster refrigeration system (System S3).

Table 10. Transcritical R-744 (CO_2) booster refrigeration system configuration (System S3) (Beshr et al. 2015)

Compressor rack	MT capacity, Btu/h (kW)	LT capacity, Btu/h (kW)	Refrigerant charge, lb (kg)
1	405 000 (118.6)	220 000 (64.6)	2000 (905)
2	345 000 (101.2)	80 000 (23.4)	1360 (615)

3.2.4 Secondary (MT) / Central DX (LT) System (S4)

As shown in Fig. 13, the fourth system (designated as S4) consists of two individual refrigeration systems: (1) a secondary loop system for cooling the medium-temperature loads, and (2) a direct expansion system for cooling the low-temperature loads. The MT loads are served by a secondary circuit which uses propylene glycol cooled by a DX system using L-40 refrigerant (a low-GWP A2L alternative for R-404A). The LT loads are satisfied with a separate centralized DX system utilizing R-448A refrigerant. The refrigeration load and refrigerant charge for each of the four compressor racks which serve the loads of the model supermarket are given in Table 11.

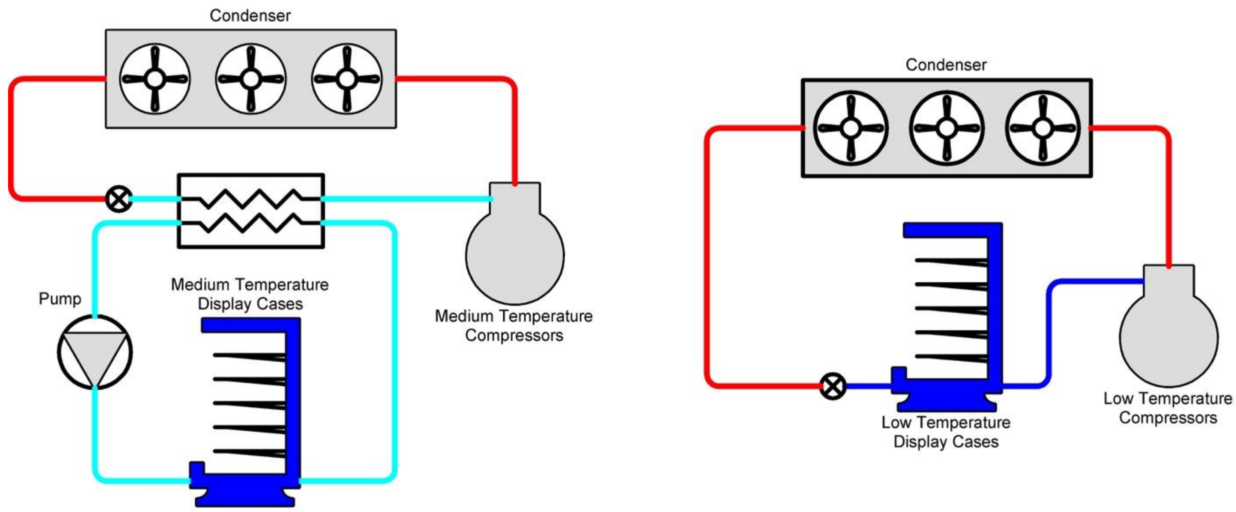

Fig. 13. Secondary (MT) / central DX (LT) refrigeration system (System S4).

Table 11. Secondary (MT) / central DX (LT) refrigeration system configuration (System S4) (Beshr et al. 2015)

Compressor rack	Refrigeration capacity, Btu/h (kW)	Refrigerant Charge, lb (kg)
MT1	570 000 (167.1)	460 (209)
MT2	180 000 (52.8)	150 (66)
LT1	220 000 (64.6)	640 (290)
LT2	80 000 (23.4)	230 (104)

3.2.5 Refrigeration System Modeling Assumptions

The saturated evaporating temperatures for the four refrigeration system types are shown in Table 12. It was assumed that the saturated evaporating temperatures of the CO_2-based refrigeration systems were 2.0°F (1.1°C) higher than that of the HFC-based systems due to the enhanced transport properties of carbon dioxide.

Table 12. Evaporating temperatures for the three refrigeration systems

System	Compressor rack	Temperature level	Saturated evaporating temperature, °F (°C)
S1	1	MT	27.0 (-2.80)
		LT	-4.0 (-20.0)
	2	MT	27.0 (-2.80)
		LT	-19.0 (-28.3)
S2	1	MT	29.0 (-1.70)
		LT	0.0 (-18.0)
	2	MT	29.0 (-1.70)
		LT	-17.0 (-27.2)
S3	1	MT	29.0 (-1.70)
		LT	0.0 (-18.0)
	2	MT	29.0 (-1.70)
		LT	-17.0 (-27.2)
S4	MT1	MT	27.0 (-2.80)
	LT1	LT	-4.0 (-20.0)
	MT2	MT	27.0 (-2.80)
	LT2	LT	-19.0 (-28.3)

Additional modeling assumptions for the refrigeration systems are as follows:

- For systems S1, S2 and S4, the suction line pressure drop was assumed to be 2.5 psi (17 kPa) while the discharge line pressure drop was assumed to be 1.2 psi (8.5 kPa). The suction gas temperature at the compressor inlet was assumed to be the saturated evaporating temperature plus the display case superheat. The display case superheat was assumed to be 7.2°R (4.0 K) for systems S1, S2 and S4 and 18°R (10 K) for system S3.
- System S1 contains a subcooler for each LT compressor rack. The temperature of the liquid refrigerant exiting the subcooler was set to 50°F (10°C). No subcoolers were used in systems S2 and S4. System S3 contains a liquid-suction heat exchanger just after the gas cooler. The effectiveness of this heat exchanger was assumed to be 0.4.
- The difference between the saturated condensing temperature and the ambient air for systems S1, S2 and S4 were approximately 7.2°R (4.0 K) for ambient temperatures greater than or equal to 68°F (20°C). In addition, the minimum condensing temperature for Systems S1, S2 and S4 was set at 72°F (22°C). No subcooling was assumed at the exit of the condensers.
- For system S3, it was assumed that the temperature difference between the gas cooler outlet and the ambient air was 5.4°R (3 K) for transcritical operation and 18°R (10 K) for subcritical operation. During subcritical operation, the minimum condensing temperature for System S3 was set at 50°F (10°C).
- For all systems, the performance of the compressors was determined from manufacturers' compressor maps. Inefficiencies occurring during part load operation were ignored. The performance of the R-448A compressors was based on R-404A compressor performance by assuming that R-448A capacity was 100% of R-404A capacity and R-448A power was 93% of R-404A power.

The composition and global warming potential of the refrigerants used in the energy modeling and LCCP analysis are shown in Table 13.

Table 13. Refrigerant composition and GWP values (ASHRAE 2016; IPCC 2013)

Refrigerant	Composition (Mass %)	ASHRAE Safety Classification	GWP
R-744 (CO_2)	R-744 (100.0)	A1	1
R-448A	R-32/125/1234yf/134a/1234ze(E) (26.0/26.0/20.0/21.0/7.0)	A1	1273
L-40	R-32/152a/1234yf/1234ze(E) (40.0/10.0/20.0/30.0)	A2L	285
R-404A	R-125/143a/134a (44.0/52.0/4.0)	A1	3943

Finally, the refrigeration system lifetime was assumed to be 20 years with a service interval of two years. The annual leakage rate, refrigerant loss at end-of-life, service leakage rate, and reused refrigerant were assumed to be 10%, 10%, 5%, and 85%, respectively (Abdelaziz, Fricke, and Vineyard 2012).

3.3 RESULTS AND DISCUSSION

The energy consumption as well as the direct, indirect and total CO_{2e} emissions for the four refrigeration systems operating in each of the six cities listed in Table 7 were determined. In addition, the effect of various input parameters on the calculated CO_{2e} emissions was determined. These results of these analyses are presented below.

3.3.1 LCCP Analysis

The direct emissions of the four refrigeration systems (S1, S2, S3 and S4) are shown in Fig. 14. The direct emissions are dependent on the GWP of the system refrigerant and the refrigerant charge mass. Direct emissions are not affected by the location of the refrigeration system. It can be seen that the transcritical CO_2 booster system (system S3) has the lowest direct emissions due to the very low GWP of its refrigerant and its low refrigerant charge mass. Although system S4 has slightly lower refrigerant charge mass than system S3, it has higher direct emissions than S3 because S4 utilizes a refrigerant with a much higher GWP. The baseline system, S1 with R-404A, has the highest direct emissions since it has the largest refrigerant charge using and the highest GWP refrigerant of the four systems investigated. For system S1, it can be seen that replacing R-404A with R-448A can reduce the direct emissions of the system by over 67%.

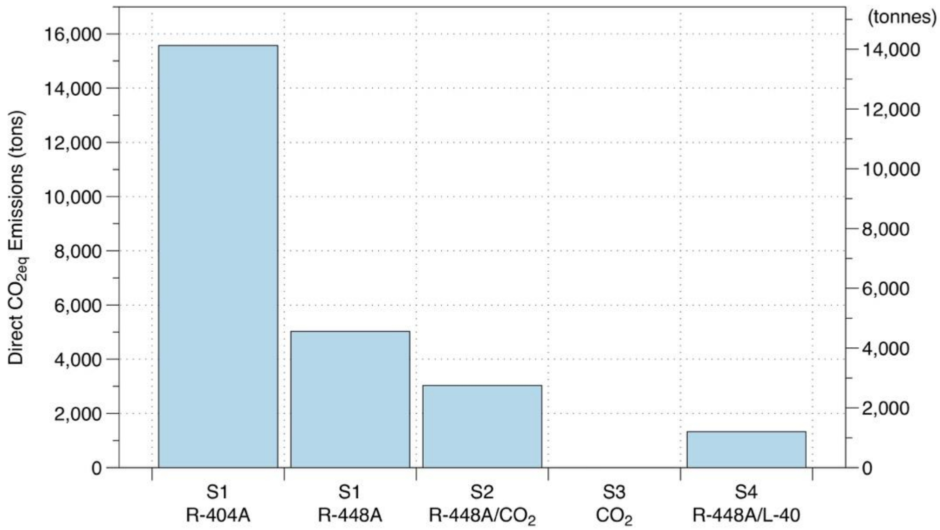

Fig. 14. Total direct emissions from the four refrigeration systems.

The annual energy consumption and total indirect emissions associated with the four refrigeration systems, located in the six cities listed in Table 7, are shown in Fig. 15. It was assumed that the location of the refrigeration system does not affect its direct emissions; however, it can be seen that location does dramatically affect the indirect emissions of the system. The energy consumption of a refrigeration system is dependent upon the weather conditions, while the indirect emissions are dependent upon both the refrigeration system energy consumption and the CO_2 emission rate for electricity production. As shown in Fig. 15, it can be seen that in general, the annual energy consumption of the refrigeration systems decreases from warm to cold climates. In addition, in cold climates such as Fairbanks, the energy consumption of the transcritical CO_2 booster system (system S3) is much closer to that of the other systems than in warmer climates such as Miami. This is because system S3 does not perform as efficiently in warm climates as it does in cold climates. Thus, in warmer climates, system S3 tends to have higher indirect emissions as compared to the other systems. Also, it is worth noting that the indirect emissions in Los Angeles are much lower than that in the other cities for all four refrigeration systems, even though the annual electricity consumption of the systems in Los Angeles is relatively high. This behavior is due to the low hourly emission rate for electricity production in Los Angeles, as shown in Table 7. It can be seen that for all climates, the secondary (MT)/central DX (LT) refrigeration system (system S4) tends to have the lowest electricity consumption and hence, the lowest indirect emissions. On the other hand, for all but the coldest climates, the transcritical CO_2 booster system (system S3) results in higher electricity consumption and higher indirect emissions.

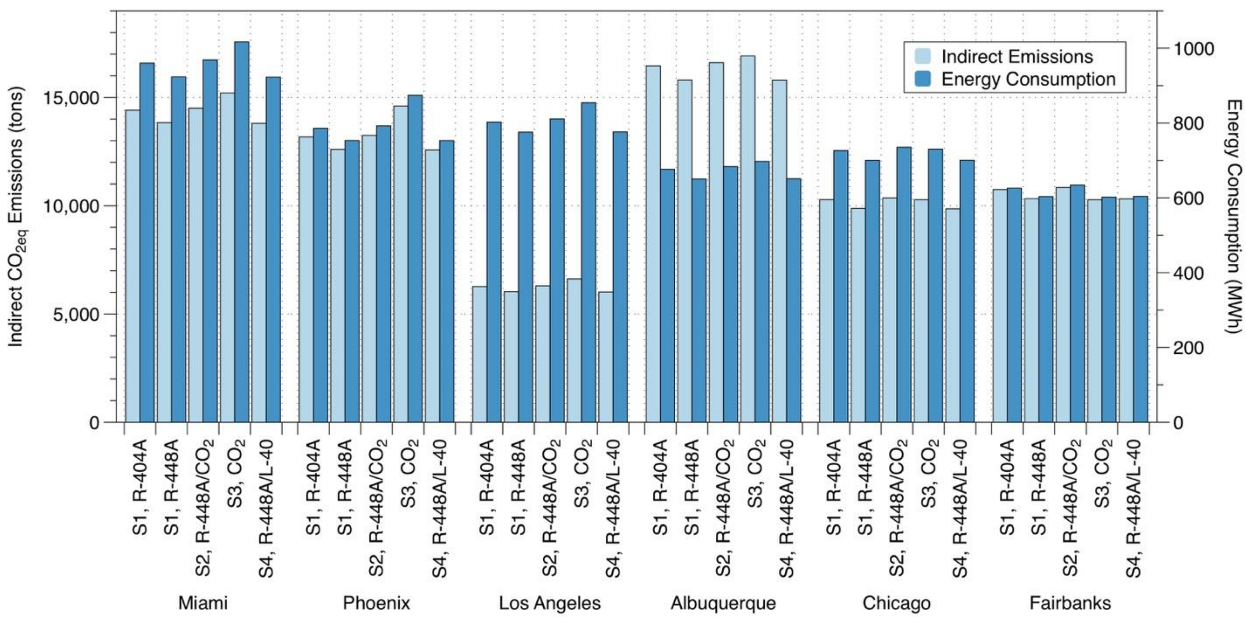

Fig. 15. Total indirect emissions and annual electricity consumption for the four refrigeration systems.

The total emissions from the four refrigeration systems, for the six cities, is shown in Fig. 16. For cities in cold and temperate climates, the slightly higher indirect emissions of the transcritical CO_2 booster system are outweighed by its lower direct emissions. Thus, for these climates, system S3 has the lowest total emissions. On the other hand, for cities in hot climates, such as Miami and Phoenix, the higher direct emissions of system S4, as compared to S3, are outweighed by its lower indirect emissions. Hence, system S4 is more environmentally friendly in the hot climates. Overall, S4 offers a good trade-off between emissions and electricity consumption making it more attractive for most climates. Finally, the multiplex DX system (system S1) utilizing R-404A has the highest total emissions.

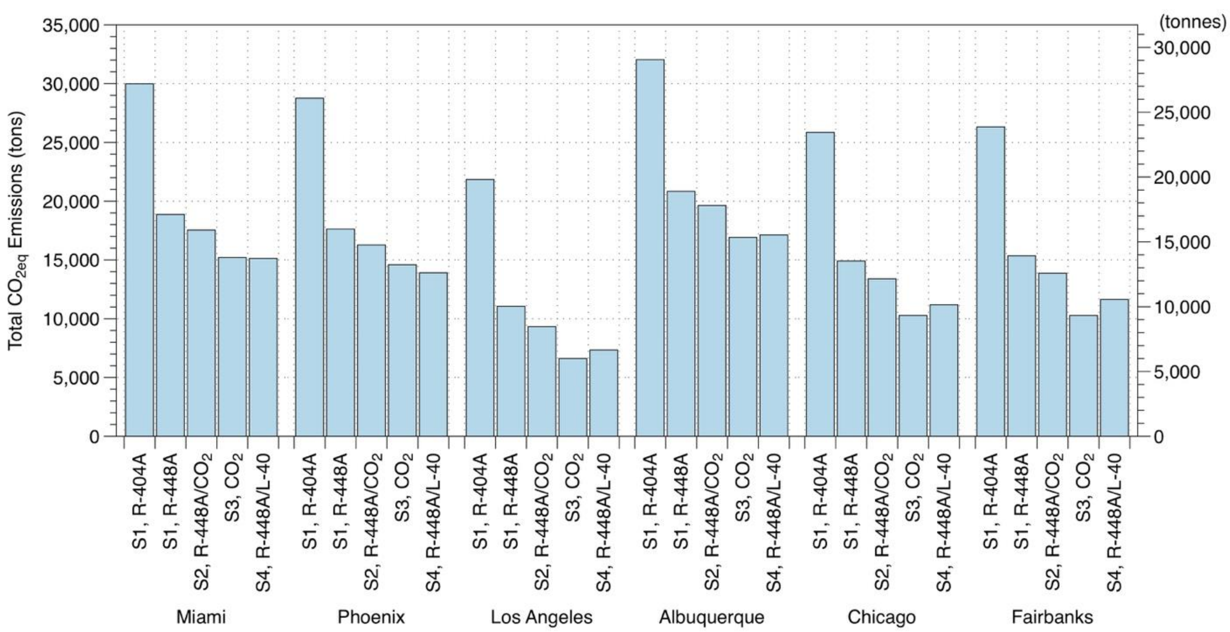

Fig. 16. Total emissions (direct and indirect) from the four refrigeration systems.

3.3.2 Effect of Annual Refrigerant Leak Rate

For a supermarket refrigeration system, direct emissions can be the major contributor to the total CO_{2e} emissions of the system. Therefore, using lower GWP refrigerants, or ensuring the leak-tightness of the system can result in a considerable decrease in the system's total CO_{2e} emissions. The effect of the annual refrigerant leak rate on total CO_{2e} emissions is shown in Fig. 17. The analysis presented in Fig. 17 was performed for Systems S1, S3 and S4 in Chicago. For system S3, which is the transcritical CO_2 booster system, the impact of increasing or decreasing the annual leak rate is negligible due the very low GWP of CO_2. On the other hand, for refrigeration systems using moderate or high GWP refrigerants, such as system S1 utilizing R-404A, reducing the annual leak rate has a large impact on the total emissions. As previously discussed, for the baseline annual leakage rate of 10% in Chicago, system S4 has higher total emissions than system S3 although system S4 has lower indirect emissions than system S3. This is due to the much lower direct emissions of system S3 as compared to system S4. As shown in Fig. 17, if the annual refrigerant leak rate is lower than 2%, the higher direct emissions of system S4 compared to system S3 is offset by its lower indirect emissions. Hence, for very low annual refrigerant leak rates, system S4 tends to be more environmentally friendly than all of the other refrigeration systems investigated, in all of the climate zones.

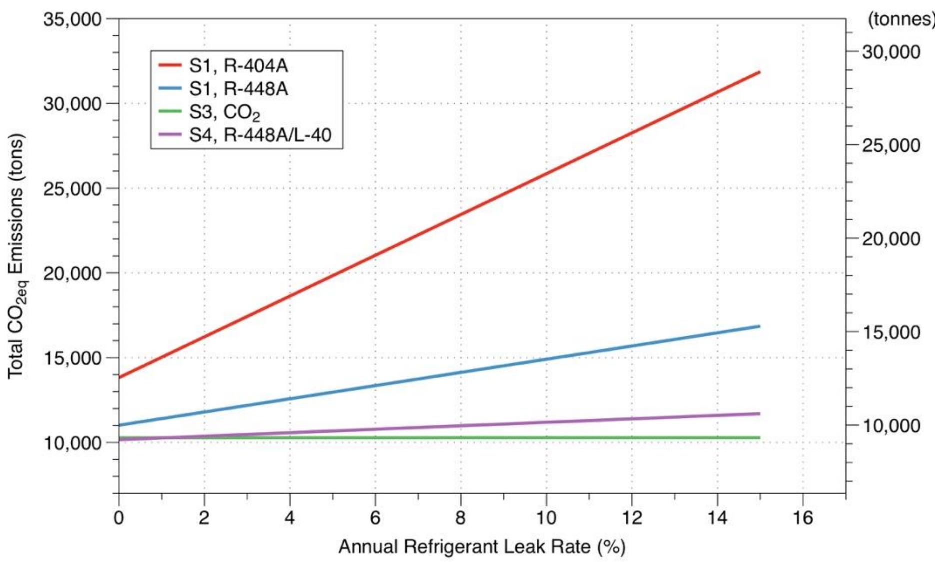

Fig. 17. Effect of refrigerant leak rate on total CO_{2e} emissions.

3.3.3 Effect of Refrigerant Charge or Hourly Emission Rate for Electricity Production

A sensitivity analysis was performed to determine the effect of a 10% increase in either the refrigerant charge or the hourly emission rate for electricity production on the total CO_{2e} emissions of systems S1 and S3 in Miami and Los Angeles. As noted earlier for supermarket refrigeration systems, refrigerant GWP and charge mass have a strong impact on total CO_{2e} emissions. Thus, the total CO_{2e} emissions for systems with higher direct emissions (i.e., systems utilizing refrigerants with higher GWP, or having larger charge mass) tend to be more sensitive to charge variation than systems with lower direct emissions (i.e. utilizing lower GWP refrigerants or having smaller charge mass), as shown in Fig. 18.

Moreover, using low GWP refrigerants or smaller system charge reduces the direct emissions, hence the impact of indirect emissions on total CO_{2e} emissions becomes more prevalent. As such, the sensitivity of total CO_{2e} emissions to variation in electricity production emission rates becomes more noticeable for low GWP refrigerants or smaller charge mass systems, as shown in Fig. 19. Moreover, for cities with low hourly emission rates for electricity production, such as Los Angeles, the contribution of direct emissions to total emissions is higher than for other cities. Thus, the total CO_{2e} emissions for cities such as Los Angeles tend to have higher sensitivity to charge variation.

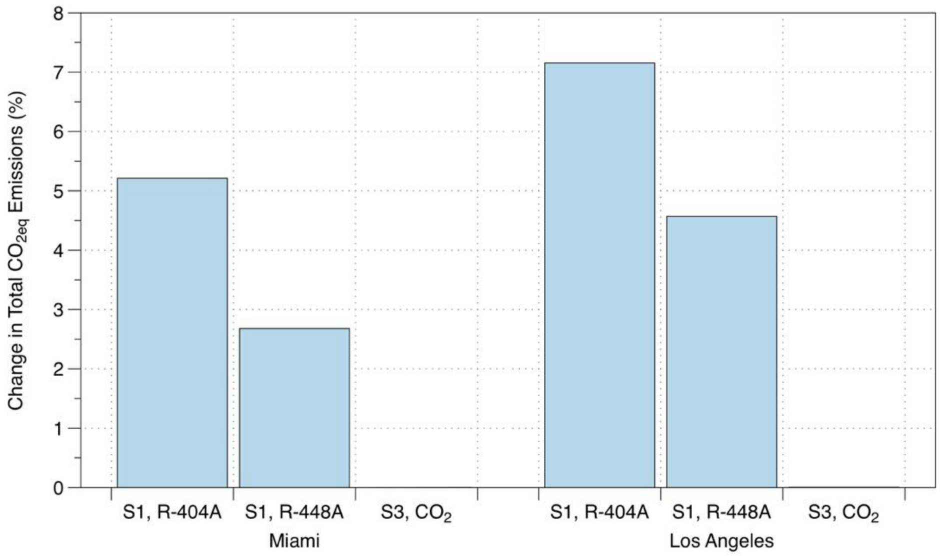

Fig. 18. Sensitivity of total CO_{2e} emissions due to a 10% increase in refrigerant charge.

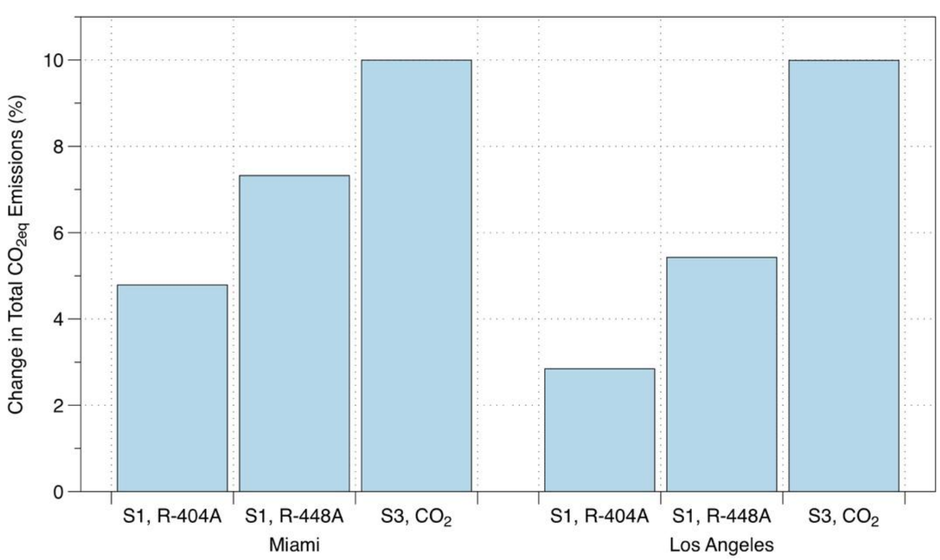

Fig. 19. Sensitivity of total CO_{2e} emissions due to a 10% increase in electricity production emissions.

3.3.4 Uncertainty Analysis

To determine the effect of the uncertainty of various input parameters on the total CO_{2e} emissions, an uncertainty analysis was performed using the multiplex DX system (S1) located in Chicago and Los Angeles, for the refrigerants R-404A and R-448A. The input parameters included in the uncertainty analysis included the service leakage rate, the annual leakage rate, the refrigerant loss at end-of-life, the percentage of reused refrigerant, the system charge, the hourly emission rate for electricity production, and the refrigerant's GWP. The analysis was performed for three different cases. For all three cases, the uncertainty in the values for reused refrigerant, service leakage rate, refrigerant loss at end-of-life, and annual leakage rate were assumed to be 20%, while the refrigerant charge uncertainty was assumed to be 5%. The power plant emissions and the refrigerant GWP uncertainties were varied. For cases 1, 2, and 3, the uncertainty in energy production emissions was 5%, 5%, and 20%, respectively, while the refrigerant GWP uncertainty was 20%, 5%, and 5%, respectively. The range of uncertainty for these various parameters is summarized in Table 14.

Table 14. Summary of uncertainty in LCCP parameters

Parameter	**Uncertainty (%)**		
	Case 1	**Case 2**	**Case 3**
Reused refrigerant	20	20	20
Service leakage rate	20	20	20
Refrigerant loss at end-of-life	20	20	20
Annual leakage rate	20	20	20
Charge	5	5	5
Power plant emission	5	5	20
Refrigerant GWP	20	5	5

The resulting uncertainties for the total CO_{2e} emissions of the multiplex DX system (S1), due to uncertainties in the input parameters, are shown in Table 15. The partial derivatives of the total emissions with respect to each of the input parameters are shown in Table 16, accompanied by the percentage difference between the derivatives values when utilizing R-448A compared to R-404A. The partial derivatives do not change for any of the three cases. This is because the derivatives do not depend on the magnitude of uncertainty of the input parameters, but rather on the baseline design itself. Thus, changing the location from Chicago to Seattle only causes a change in the partial derivative which depends on the location, such as the derivative of the total emissions with respect to the hourly emission rate for electricity production. Also, the results show that shifting to a low GWP refrigerant causes a noticeable drop in the impact of the uncertainty in the input parameters related to the direct emissions (i.e., service leakage rate, refrigerant loss at end-of-life, annual leakage rate, and system refrigerant charge) on the total emissions. This results in a more dominant impact of the power plant emissions uncertainty on the uncertainty of the total system emissions. Thus, for the low GWP refrigerant (R-448A) in Chicago for case 3, the uncertainty of the system's total emissions is higher than for R-404A. Finally, performing the same uncertainty analysis for S3 in Los Angeles results in LCCP uncertainty of 5%, 5%, and 19.98% for cases 1, 2, and 3, respectively. This is due to the small direct emissions of this system which makes the uncertainty in the power plant emissions much more dominant than in the case of other systems. This causes the uncertainty in the transcritical CO_2 booster system's LCCP to follow the value of the uncertainty in the power plant emissions.

Table 15. Uncertainties in the total CO_{2e} emissions of the multiplex DX system (S1)

Case	City	Refrigerant	Uncertainty in total CO_{2e} emissions (%)
Case 1	Chicago	R-404A	15.82
		R-448A	9.40
	Los Angeles	R-404A	18.64
		R-448A	12.17
Case 2	Chicago	R-404A	10.69
		R-448A	6.76
	Los Angeles	R-404A	12.52
		R-448A	8.40
Case 3	Chicago	R-404A	13.15
		R-448A	14.47
	Los Angeles	R-404A	13.68
		R-448A	13.45

Table 16. Partial derivatives of the total emissions with respect to each of the input parameters

Parameter	Chicago			Los Angeles		
	Partial derivative			Partial derivative		
	R-404A	R-448A	Difference, %	R-404A	R-448A	Difference, %
Reused refrigerant	-2.30E+04	-1.10E+04	-52.1	-2.30E+04	-1.10E+04	-52.1
Service leakage rate	5.43E+07	1.75E+07	-67.7	5.43E+07	1.75E+07	-67.7
Refrigerant loss at end-of-life	5.43E+06	1.75E+06	-67.7	5.43E+06	1.75E+06	-67.7
Annual leakage rate	1.09E+08	3.53E+07	-67.6	1.09E+08	3.53E+07	-67.6
Refrigerant charge	1.03E+04	3.33E+03	-67.7	1.03E+04	3.33E+03	-67.7
Power plant emission	1.45E+07	1.40E+07	-3.6	1.61E+07	1.55E+07	-3.4
Refrigerant GWP	3.58E+03	3.58E+03	0.0	3.58E+03	3.58E+03	0.0

3.4 SUMMARY

A flexible LCCP design tool was developed for the design and evaluation of supermarket refrigeration systems. The LCCP framework is open-source and can be easily extended to the analysis of other refrigeration and vapor compression technologies. This LCCP design tool was used to compare the environmental impact of four different supermarket refrigeration systems in six US cities. Comparing the total emissions for different cities suggests that the transcritical CO_2 booster system has the lowest CO_2 equivalent emissions, according to this analysis, in cold and temperate. Also, the R-448A/L-40 secondary circuit refrigeration system was found to offer a good balance between emissions and electricity consumption for hot climates. The parametric analysis showed that shifting towards low GWP refrigerants decreases the effect of the annual leak rate on the total system emissions. Moreover, the sensitivity analysis showed that shifting towards low GWP refrigerants, or more charge conservative systems increases the effect of the hourly emission rate for electricity production on the total system emissions. Finally, an uncertainty analysis was performed showing that using low GWP refrigerants, or more charge conservative systems causes a noticeable drop in the impact of the uncertainty in the inputs related to the direct emissions.

4. LABORATORY EVALUATION OF ALTERNATIVE REFRIGERANTS

The energy performance of an alternative lower global warming potential refrigerant was evaluated in a laboratory-scale commercial refrigeration system, and its performance was compared to that of a commonly used higher GWP refrigerant found in the commercial refrigeration industry. The aim is to identify safe, energy efficient, low GWP alternatives to today's commonly used high GWP refrigerants. The two refrigerants investigated were R-404A and R-448A. R-404A is a commonly used refrigerant in the commercial refrigeration industry and has a relatively high global warming potential, while R-448A is a new lower GWP alternative for R-404A.

R-404A is a zeotropic blend of several hydrofluorocarbon (HFC) refrigerants, and consists of 44.0% (by mass) R-125, 52.0% (by mass) R-143a and 4.0% (by mass) R-134a. The resulting mixture exhibits a low temperature glide of less than 1.1°F (0.6°C) under typical operating conditions. The global warming potential of R-404A is relatively high, with a value of 3,943 (IPCC 2013). R-404A has an ASHRAE safety classification of A1, indicating that it has low toxicity and is non-flammable (ASHRAE 2016).

R-448A is a non-flammable zeotropic blend of several HFC and hydrofluoro-olefin (HFO) refrigerants, including R-32 (26.0% by mass), R-125 (26.0% by mass), R-1234yf (20.0% by mass), R-134a (21.0% by mass) and R-1234ze(E) (7.0% by mass). R-448A exhibits a larger temperature glide than R-404A, with a value no more than 11°F (6°C) under typical operating conditions. The global warming potential of R-448A is significantly lower than that of R-404A, with a value of approximately 1,273 (IPCC 2013). R-448A has an ASHRAE safety classification of A1, indicating that it has low toxicity and is non-flammable (ASHRAE 2016).

4.1 COMMERCIAL REFRIGERATION SYSTEM DESCRIPTION

The laboratory-scale commercial refrigeration system installed in the environmental test chambers at the Oak Ridge National Laboratory, that was used to investigate the energy performance of R-404A and R-448A, consists of components typically found in most U.S. supermarket refrigeration systems. These components include a compressor rack, an air-cooled condenser and several medium-temperature (MT) and low-temperature (LT) refrigerated display cases. A schematic of the laboratory-scale refrigeration system, showing these various components as well as several instrumentation points, is shown in Fig. 20.

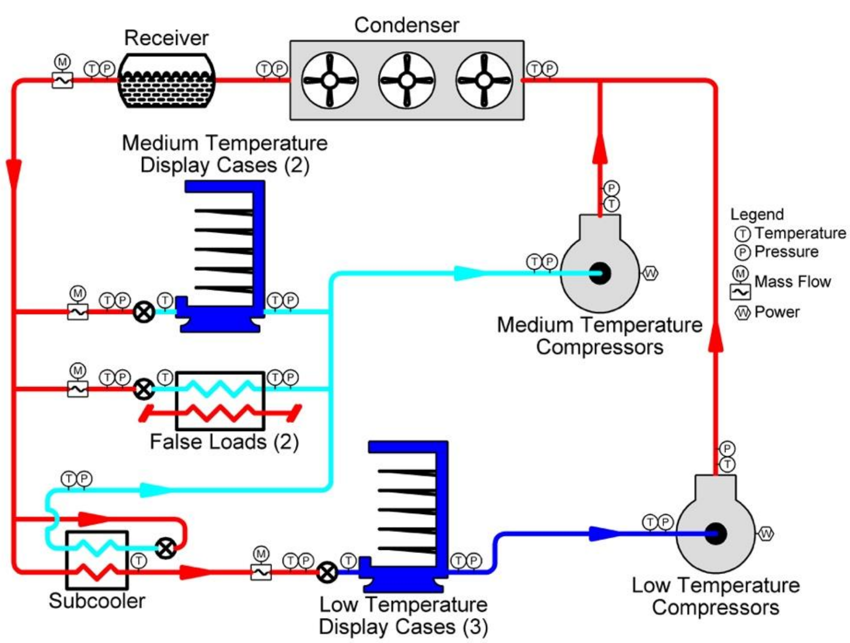

Fig. 20. Schematic of laboratory-scale commercial refrigeration system.

The refrigeration system has a low-temperature cooling capacity of approximately 5 tons at −20°F (18 kW at −29°C) and a medium-temperature cooling capacity of approximately 10 to 15 tons at 25°F (35 to 53 kW at −4°C). Three open vertical display cases, each 12 ft (3.7 m) in length, constitute the low-temperature load. The medium-temperature load consists of two open vertical display cases, each 12 ft (3.7 m) in length, as well as a "false" load provided by a plate heat exchanger and glycol loop.

The compressor rack and air-cooled condenser are installed in a temperature and humidity controlled "outdoor" environmental chamber while the refrigerated display cases are installed in a separate temperature and humidity controlled "indoor" environmental chamber. For both chambers, the temperature can be controlled between 0 to 130°F (−18 to 54°C) and the humidity can be controlled between 30 to 90%. Thus, the air-cooled condenser can be exposed to typical outdoor ambient conditions while the refrigerated display cases operate in an environment typical of that found in the sales area of a supermarket.

4.1.1 Compressor Rack

The compressor rack, shown in Fig. 21, consists of several refrigerant compressors and the associated piping that forms the low-temperature and medium-temperature liquid headers and suction headers as well as the common discharge header. The refrigeration loads (i.e., the refrigerated display cases) are connected to the low-temperature and medium-temperature liquid and suction headers of the compressor rack and the condenser is connected to the discharge header of the compressor rack.

Fig. 21. Compressor rack.

The compressor rack of the commercial refrigeration system is unique in that it contains two types of compressors that are commonly found in supermarket refrigeration systems: reciprocating compressors and scroll compressors. Note that only one type of compressor (i.e., reciprocating or scroll) is operated at one time. In essence, the compressor rack is two racks in one: a reciprocating compressor rack and a scroll compressor rack. Having both reciprocating and scroll compressors allows for system performance tests to be performed using the two common compressor types employed in supermarket refrigeration systems.

For each compressor type, the refrigeration system contains two low-temperature compressors and two medium-temperature compressors. In addition, for each compressor type and temperature level, the compressor rack contains a primary compressor that is variable capacity (capable of modulating its capacity from 10% to 100%), and a secondary compressor that is fixed capacity. The primary compressor is used first to satisfy the refrigeration load, and it can modulate its capacity to match the applied load. If the primary compressor is not sufficient to satisfy the load, then the secondary compressor operates as well, with the primary compressor modulating its capacity to match the load.

The refrigerating capacity and power requirements for both the reciprocating and scroll compressors are given in Table 17. In addition, the total refrigerating capacity of the compressor rack is given in Table 18.

Table 17. Compressor specifications for laboratory-scale refrigeration system

Compressor type	Temperature level	Capacity control	Refrigerating capacity, Btu/h (kW)*	Power, kW*
Reciprocating	LT	Variable	32 000 (9.38)	5.70
Reciprocating	LT	Fixed	32 000 (9.38)	5.70
Reciprocating	MT	Variable	92 300 (27.1)	8.50
Reciprocating	MT	Fixed	92 300 (27.1)	8.50
Scroll	LT	Variable	32 300 (9.47)	5.20
Scroll	LT	Fixed	32 300 (9.47)	5.20
Scroll	MT	Variable	56 300 (16.5)	5.23
Scroll	MT	Fixed	56 300 (16.5)	5.23

*Refrigerating capacity and power are given for the following operating conditions using R-404A:
LT: −20°F (−29°C) evaporating temperature, 105°F (41°C) condensing temperature
MT: 25°F (−3.9°C) evaporating temperature, 105°F (41°C) condensing temperature

Table 18. Compressor rack capacity for laboratory-scale refrigeration system

Compressor type	Temperature level	Compressor rack capacity, Btu/h (kW)
Reciprocating	LT	64 000 (18.8)
	MT	184 600 (54.1)
Scroll	LT	64 600 (18.9)
	MT	112 600 (33.0)

4.1.2 Refrigerated Display Cases

The three low-temperature and two medium-temperature display cases of the commercial refrigeration system are open, vertical multi-deck models, each 12 ft (3.7 m) in length. The rated capacity of each LT case is 18 100 Btu/h (5.30 kW) while the rated capacity of each MT case is 17 100 Btu/h (5.01 kW), according to ASHRAE Standard 72 (2005). Fig. 22 shows representative photos of the low-temperature and medium-temperature display cases.

(a) (b)

Fig. 22. Low-temperature (a) and medium-temperature (b) display cases.

Each low-temperature display case has one evaporator with one corresponding mechanical thermostatic expansion valve. Each medium-temperature display case contains two evaporators, with each evaporator having its own thermostatic expansion valve. Thus, each MT display case contains two thermostatic expansion valves. The LT and MT display cases utilize fluorescent lighting and shaded-pole evaporator fan motors. In addition, the LT display cases have anti-sweat heaters. Finally, the LT display cases utilize electric defrost heaters while the MT display cases utilize off-cycle defrost. The specifications for the low-temperature and medium-temperature display cases are shown in Table 19.

In addition to the refrigerated display cases, a medium-temperature "false" load is also incorporated into the refrigeration system. This false load consists of a plate heat exchanger with two medium-temperature refrigerant circuits on one side of the heat exchanger and a glycol circuit on the other side. The false load provides an additional load of approximately 50 000 Btu/h (14.7 kW) to the medium-temperature side of the refrigeration system.

Table 19. Refrigerated display case specifications

Display case parameter	Low-temperature display case	Medium-temperature display case
Type	Open, vertical multi-deck	Open, vertical multi-deck
Length	12 ft (3.7 m)	12 ft (3.7 m)
Rated Capacity	18 100 Btu/h (5.30 kW)	17 100 Btu/h (5.01 kW)
Fan Amperage	7.2 A	0.56 A
Lighting Amperage	4.2 A	0.72 A
Anti-Sweat Heater Amperage	8.6 A	--
Defrost Type	Electric	Off-cycle
Defrost Amperage	20.4 A	--

4.1.3 Air-Cooled Condenser

As shown in the system schematic in Fig. 20, an air-cooled condenser is used to reject heat from the refrigeration system. The condenser accepts the discharge refrigerant vapor from the compressor rack, condenses the refrigerant, and discharges the condensed refrigerant into a liquid receiver. The air-cooled condenser has four variable speed fans and its rated capacity, with R-404A, is 351 000 Btu/h (103 kW) at a temperature difference of 10°F (5.6°C) between the saturated condensing temperature and the ambient air. Fig. 23 shows a photo of the air-cooled condenser, mounted above the liquid receiver, as installed in the "outdoor" environmental chamber.

Fig. 23. Air-cooled condenser (top) mounted above liquid receiver (bottom).

4.1.4 Refrigeration System Controls

The refrigeration system controller provides the following control:

- Compressor control to maintain suction pressure setpoints for the LT and MT circuits
- Condenser fan speed control to maintain condensing pressure
- Display case temperature control via electronic evaporator pressure regulators
- Display case defrost initiation and termination

In addition to the system controller, the refrigeration system also has electronic display case superheat controllers, and an electronic subcooler controller. Note that since the refrigerated display cases currently utilize mechanical thermostatic expansion valves, the superheat controllers are not required. However, superheat controllers are provided in the system in the event that electronic expansion valves are to be used in the display cases. Also, since the false load utilizes electronic expansion valves, electronic superheat controllers are required for the two false load refrigerant circuits.

4.1.5 Instrumentation

The laboratory-scale commercial refrigeration system was fully instrumented to monitor its performance. Refrigerant temperature, pressure and flow rate were measured at various locations within the system. Compressor and display case fan/lighting and defrost heater power were measured. In addition, display case discharge and return air temperatures were measured. Table 20 lists the specifications of the instrumentation used to measure these various quantities.

Table 20. Instrumentation specifications

Instrument	Quantity measured	Range	Accuracy
Differential pressure transducer (Kele DPW-692)	Refrigerant pressure drop across condenser	0 to 25 psi (0 to 172 kPa)	<±0.5% full scale
Mass flow meter, coriolis (Micro Motion CMF series)	Refrigerant mass flow rate (receiver exit, MT liquid line, LT liquid line, display case liquid lines)	0 to 125 lb/min (0 to 0.94 kg/s)	Liquid flow: ±0.10% Gas flow: ±0.50%
Mass flow meter, positive displacement (Omega FPD2000)	Refrigerant mass flow rate (display case liquid lines, MT false load liquid lines)	0.03 to 7.0 GPM (0.0018 to 0.44 L/s)	±0.5%
Temperature/humidity sensor (Vaisala HMT120)	Indoor and outdoor chamber temperature and relative humidity	Temperature: −40 to 176°F (−40 to 80°C) Relative humidity: 0 to 100% RH	Temperature: ±0.36°F for 59 to 77°F (±0.2°C for 15 to 25°C) Relative humidity: (for 32 to 104°F or 0 to 40°C) 0 to 90% RH: ±1.7% RH 90% to 100% RH: ±2.5% RH
Temperature/humidity sensor (Vaisala HMT330)	Display case discharge/return temperature and relative humidity	Temperature: −40 to 176°F (−40 to 80°C) Relative humidity: 0 to 100% RH	Temperature: ±0.36°F at 68°F (±0.2°C at 20°C) Relative humidity: (for 59 to 77°F or 15 to 25°C) 0 to 90% RH: ±1% RH 90 to 100% RH: ±1.7% RH
Type-T thermocouple	Refrigerant temperature, product temperature, display case discharge/return temperature	−328 to 662°F (−250 to 350°C)	Greater of 1.8°F (1.0°C) or or 0.75% above 32°F (0°C)

Instrument	Quantity measured	Range	Accuracy
Pressure transducer (Wika AC-1)	Refrigerant pressure	0 to 300 psig (0 to 2.17 MPa) 0 to 500 psig (0 to 3.55 MPa)	≤1.0% of span
Watt transducer (Ohio Semitronics GW5)	Display case power, compressor power	0 to 2000 W 0 to 60 000 W	±0.2% of reading

4.2 TEST PLAN

For both refrigerants (R-404A and R-448A), the performance of the laboratory-scale commercial refrigeration system, using the reciprocating compressors, was determined at four ambient temperature conditions: 60°F, 75°F, 95°F and 105°F (16°C, 24°C, 35°C, and 41°C). For each refrigerant, the refrigeration system setpoints were adjusted so that one LT and one MT reciprocating compressor operated continuously at full capacity, and the condenser fan speed was adjusted to maintain approximately a 10°F (5.6°C) temperature differential between the saturated condensing temperature and the condenser entering air temperature. The system setpoints and operating conditions used for each refrigerant are shown in Table 21 for operation with reciprocating compressors. In addition, the refrigerated display case thermostatic expansion valves were adjusted to achieve a superheat of approximately 10°F (5.6°C) at the exit of each display case.

Table 21. Refrigeration system setpoints and operating conditions when operating with reciprocating compressors

Variable	R-404A	R-448A
LT suction pressure setpoint	12 psig (184 kPa)	8 psig (156 kPa)
LT compressor capacity range	80% to 100% capacity, least cycling	80% to 100% capacity, least cycling
MT suction pressure setpoint	45 psig (412 kPa)	38 psig (363 kPa)
MT compressor capacity range	80% to 100% capacity, least cycling	80% to 100% capacity, least cycling
Condenser pressure control setpoint	126 psig (970 kPa)	126 psig (970 kPa)
Condenser fan speed	Variable	Variable
LT display case EPR setting	100% (fully open)	100% (fully open)
MT display case EPR setting	100% (fully open)	100% (fully open)
False load circuit 1 EPR setting	50%	50%
False load circuit 2 EPR setting	2%	2%

Similarly, when using the scroll compressors, the performance of the laboratory-scale commercial refrigeration system was determined at four ambient temperature conditions using both refrigerants (R-404A and R-448A): 60°F, 75°F, 95°F and 105°F (16°C, 24°C, 35°C, and 41°C). However, the system setpoints and operating parameters were set differently than that used for the reciprocating compressor tests. The suction pressure setpoints were set at values typical of that found in actual operating supermarket refrigeration systems, and the condenser fan speed was adjusted to maintain approximately a 10°F (5.6°C) temperature differential between the saturated condensing temperature and the condenser entering air temperature. The system setpoints and operating conditions used for each refrigerant are shown in Table 22 for operation with scroll compressors. In addition, the refrigerated display case thermostatic expansion valves were adjusted to achieve a superheat of approximately 10°F (5.6°C) at the exit of each display case.

Table 22. Refrigeration system setpoints and operating conditions when operating with scroll compressors

Variable	R-404A	R-448A
LT suction pressure setpoint	17 psig (219 kPa)	14 psig (198 kPa)
LT compressor capacity range	80% to 100% capacity, least cycling	80% to 100% capacity, least cycling
MT suction pressure setpoint	50 psig (446 kPa)	46 psig (419 kPa)
MT compressor capacity range	80% to 100% capacity, least cycling	80% to 100% capacity, least cycling
Condenser pressure control setpoint	126 psig (970 kPa)	126 psig (970 kPa)
Condenser fan speed	Variable	Variable
LT display case EPR setting	Automatic	Automatic
MT display case EPR setting	Automatic	Automatic
False load circuit 1 EPR setting	Automatic	Automatic
False load circuit 2 EPR setting	Automatic	Automatic

4.2.1 Data Acquisition

After the refrigeration system achieved steady-state operation at each of the test conditions, system performance data was collected for a 24-hour period. Measurement points were sampled and recorded once per second, and 30-second averaged data was used for the calculation of thermodynamic properties and performance metrics.

4.2.2 Data Analysis

From the data collected during testing, system performance characteristics, as functions of ambient conditions, were determined, including system coefficient of performance (COP), compressor isentropic efficiency, compressor discharge temperature and pressure, suction header operating temperature and pressure, and display case discharge air temperature.

Using the 30-second averaged data for the 24-hour test periods (consisting of 2881 data points per test), the thermodynamic properties at the inlet and outlet of each major system component, including compressors, display cases, false load, and condenser, were determined using REFPROP (Lemmon, McLinden, and Huber 2013). These properties include specific volume, specific enthalpy and specific entropy. The average value of these thermodynamic properties, as well as the average values of refrigerant mass flow and compressor power, were then determined for the 24-hour test period. Note that for the refrigerated display cases, the average values of the thermodynamic properties and refrigerant mass flow excluded those periods during which the display cases were undergoing defrost.

The average refrigeration load, $\dot{Q}_i$, for each display case and false load for the 24-hour test period was calculated as follows:

$$\dot{Q}_i = \dot{m}_i\left(h_{i,out} - h_{i,in}\right) \tag{7}$$

where $\dot{m}_i$ is the average refrigerant mass flow through load *i*, $h_{i,out}$ is the average refrigerant enthalpy at the exit of load *i*, and $h_{i,in}$ is the average refrigerant enthalpy at the inlet of load *i*. The average total refrigeration load for the refrigeration system during the 24-hour test period then becomes:

$$\dot{Q}_{total} = \sum_{i=1}^{n} \dot{Q}_i \tag{8}$$

where *n* is the number of independent loads connected to the refrigeration system. Note that performance data collected during defrost periods was ignored in the calculation of refrigeration load.

The average total compressor power, $\dot{W}_{total}$, for the 24-hour test period was determined as follows:

$$\dot{W}_{total} = \sum_{i=1}^{n} \dot{W}_{i,LT} + \sum_{j=1}^{m} \dot{W}_{j,MT} \tag{9}$$

where $\dot{W}_{i,LT}$ is the average input power to low-temperature compressor *i*, $\dot{W}_{j,MT}$ is the average input power to medium-temperature compressor *j*, and *n* and *m* are the number of low- and medium-temperature compressors, respectively.

The system COP was then determined from the average total refrigeration load and the average total compressor power for the 24-hour test period:

$$COP = \frac{\dot{Q}_{total}}{\dot{W}_{total}} \tag{10}$$

The average uncertainty in the calculated COP values was estimated to be ±0.016 (Taylor and Kuyatt 1994).

The average isentropic efficiency of the i^{th} compressor, $\eta_{i,th}$, for the 24-hour test period was calculated as follows:

$$\eta_{i,th} = \frac{\dot{m}_i\left(h_{c,out,s} - h_{c,in}\right)}{\dot{W}_{i,a}} \tag{11}$$

where $\dot{m}_i$ is the average refrigerant mass flow rate through compressor *i*, $h_{c,out,s}$ is the average compressor outlet enthalpy assuming isentropic compression between the inlet and outlet pressures, $h_{c,in}$ is the average compressor inlet enthalpy determined from measured refrigerant temperature and pressure and $\dot{W}_{i,a}$ is the average measured power for compressor *i*. The average uncertainty in the calculated isentropic efficiency values was estimated to be ±0.036 (Taylor and Kuyatt 1994).

4.3 RESULTS

4.3.1 Performance with Reciprocating Compressors

The total refrigerating capacity of the system, operating with reciprocating compressors using either R-448A or R-404A, is shown in Fig. 24. It can be seen that system capacity decreases as the ambient temperature increases. Furthermore, the capacity of the system when operating with R-404A is lower than that when operating with R-448A. On average, the R-448A capacity is 4.0% higher than that of the R-404A capacity.

The total reciprocating compressor power for the refrigeration system, operating with either R-448A or R-404A, is shown in Fig. 25. It can be seen that total compressor power increases as the ambient temperature increases. Furthermore, the compressor power when operating with R-448A is lower than that when operating with R-404A. On average, the R-448A compressor power is 5.7% lower than that of the R-404A compressor power.

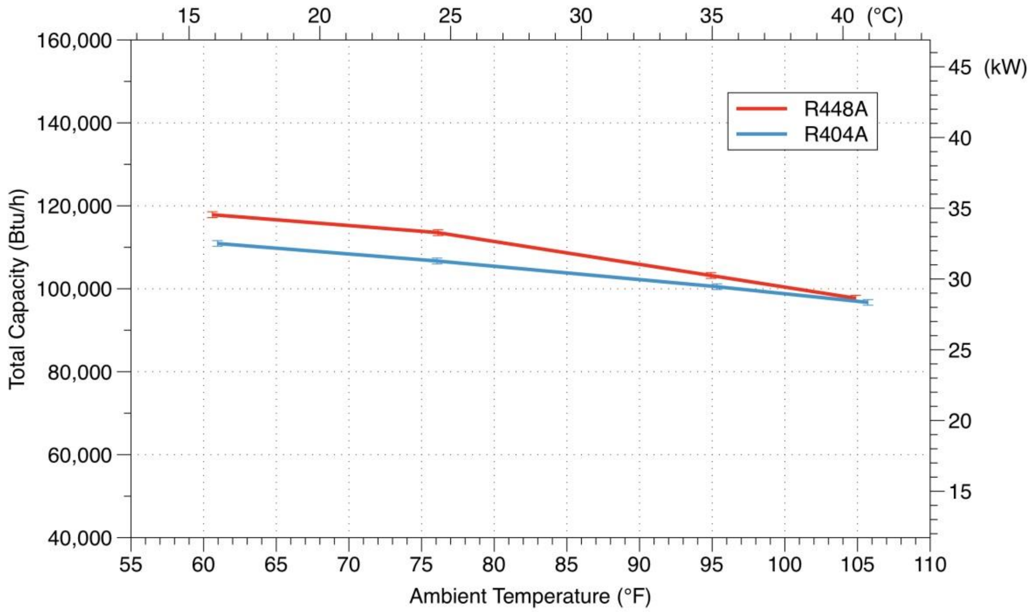

Fig. 24. Total system capacity for R-448A and R-404A, using reciprocating compressors.

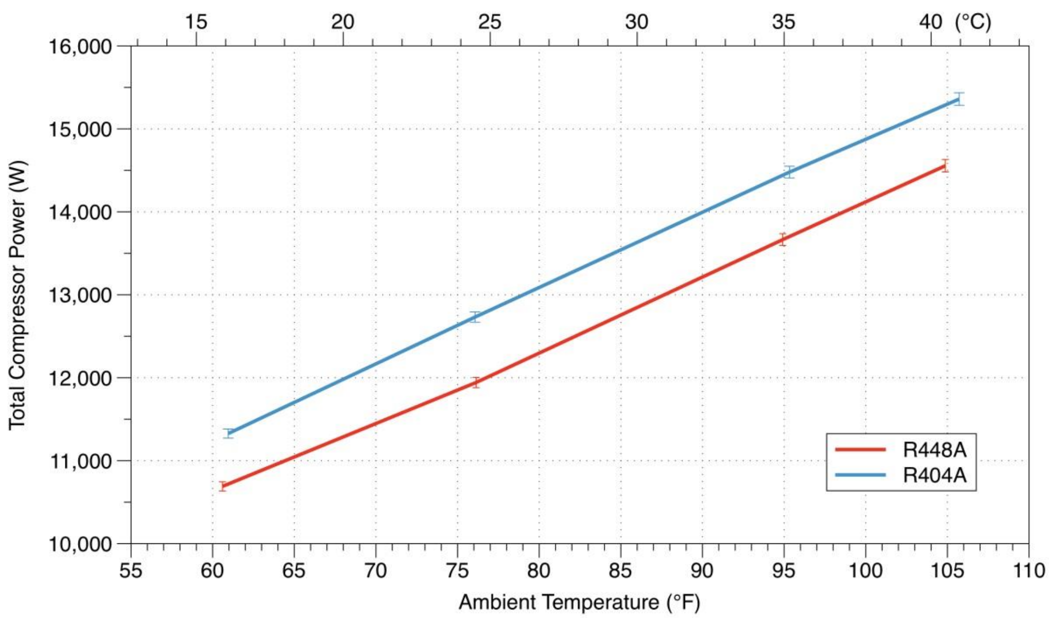

Fig. 25. Total compressor power for R-448A and R-404A, using reciprocating compressors.

The COP of the commercial refrigeration system, operating with either R-448A or R-404A, is shown in Fig. 26. As noted above, the capacity of R-448A is slightly higher than that of R-404A and the compressor power associated with R-448A is lower than that of R-404A. Overall then, the COP of the system operating with R-448A was greater than that when operating with R-404A. The maximum deviation between the R-448A COP and the R-404A COP was 13%. On average, the COP of the refrigeration system operating with R-448A was 10% higher than that when operating with R-404A.

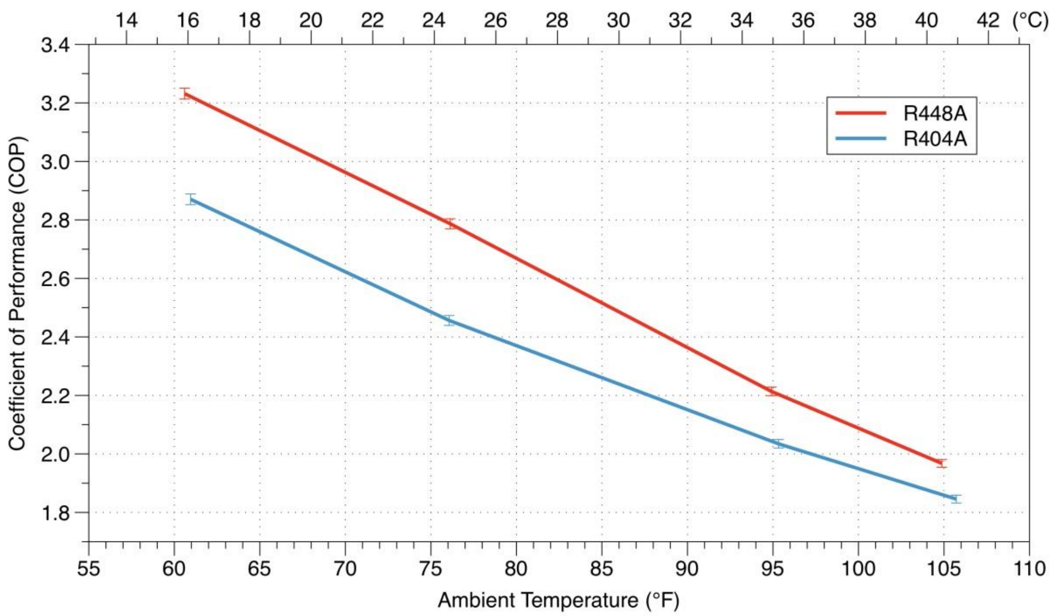

Fig. 26. Refrigeration system COP for R-448A and R-404A, using reciprocating compressors.

The total refrigerant mass flow rate for the commercial refrigeration system, operating with either R-448A or R-404A, is shown in Fig. 27. It can be seen that as the ambient temperature increases, the total refrigerant mass flow rate increases. On average, the total refrigerant mass flow rate of the system operating with R-448A was found to be approximately 25% lower than that when operating with R-404A.

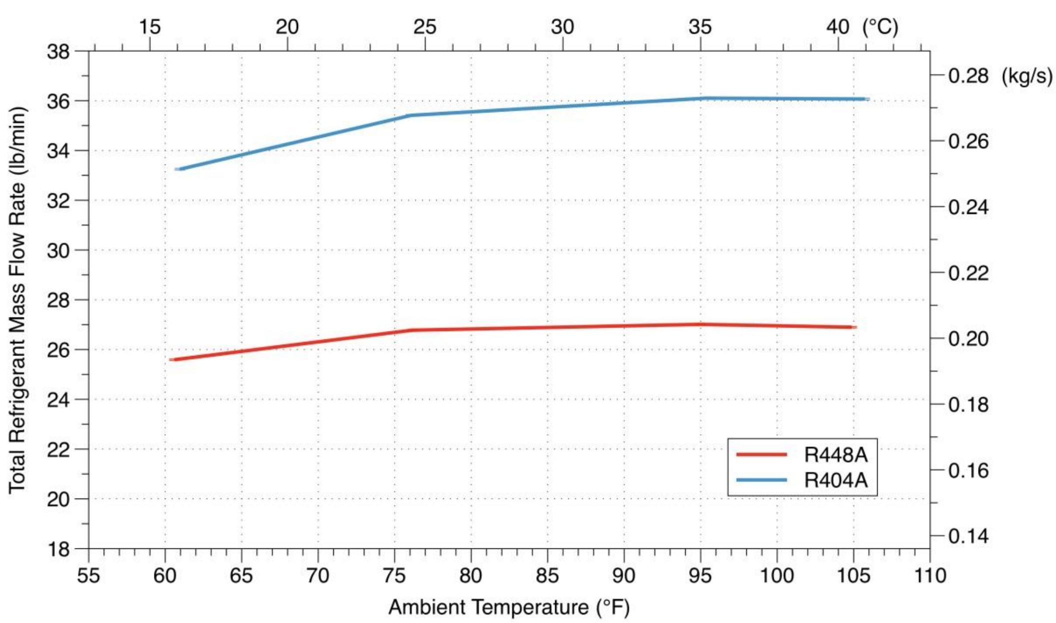

Fig. 27. Total refrigerant mass flow for R-448A and R-404A, using reciprocating compressors.

The low-temperature and medium-temperature compressor isentropic efficiencies for R-448A and R-404A are shown in Fig. 28, for the system operating with reciprocating compressors. It can be seen that as the ambient temperature increases, the isentropic efficiency of the compressors increases. In addition, it can be seen that the isentropic efficiency of the compressors when operating with R-448A is less than that when operating with R-404A. The isentropic efficiency of the low-temperature compressors, when operating with R-448A, is on average 3.9% lower than that when operating with R-404A. Similarly, the isentropic efficiency of the medium-temperature compressors, when operating with R-448A, is on average 4.2% lower than that when operating with R-404A.

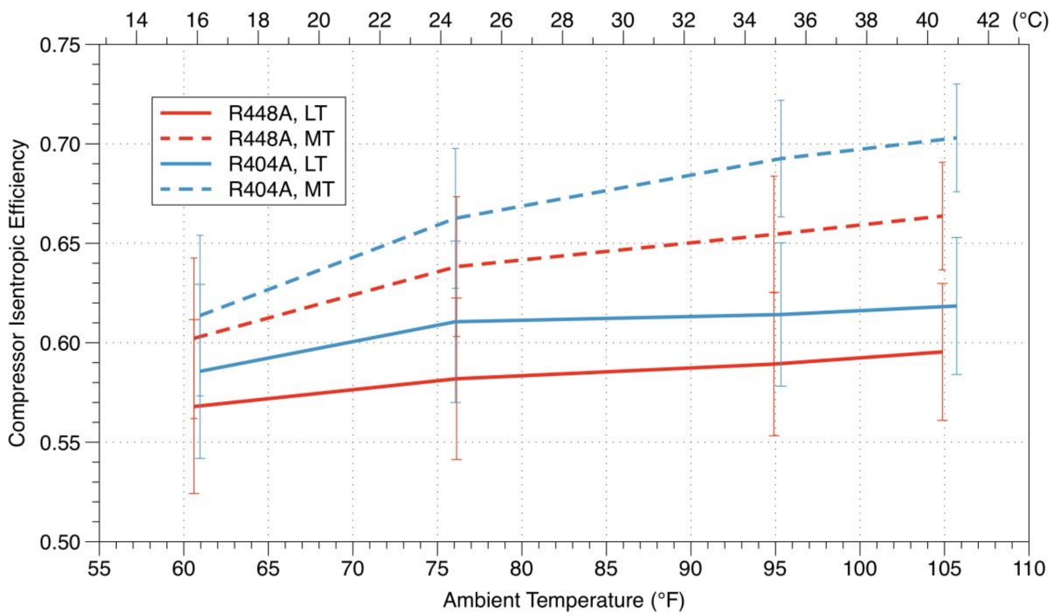

Fig. 28. Low-temperature (LT) and medium-temperature (MT) compressor isentropic efficiencies for R-448A and R-404A, using reciprocating compressors.

The low-temperature and medium-temperature compressor discharge temperatures, when operating with either R-448A or R-404A, are shown in Fig. 29. It can be seen that the compressor discharge temperature increases as ambient temperature increases. In general, the compressor discharge temperature of R-448A is between 16°F to 20°F (8.9°C to 11°C) greater than that of R-404A.

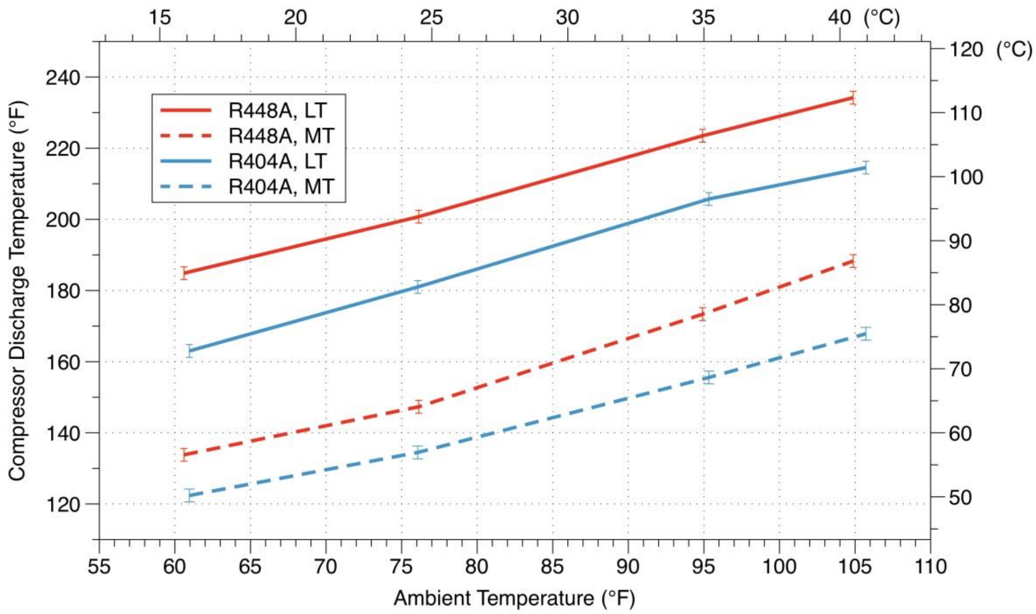

Fig. 29. Low-temperature (LT) and medium-temperature (MT) compressor discharge temperatures for R-448A and R-404A, using reciprocating compressors.

The low-temperature and medium-temperature suction header pressures for the refrigeration system, when operating with either R-448A or R-404A, are shown in Fig. 30, for the system operating with reciprocating compressors. It can be seen that the suction header pressure increases slightly as the ambient temperature increases. On average, the low-temperature suction header pressure is 2.5 psi (17 kPa) lower with R-448A as compared to R-404A, while the medium-temperature suction header pressure is 7.5 psi (51 kPa) lower with R-448A as compared to R-404A.

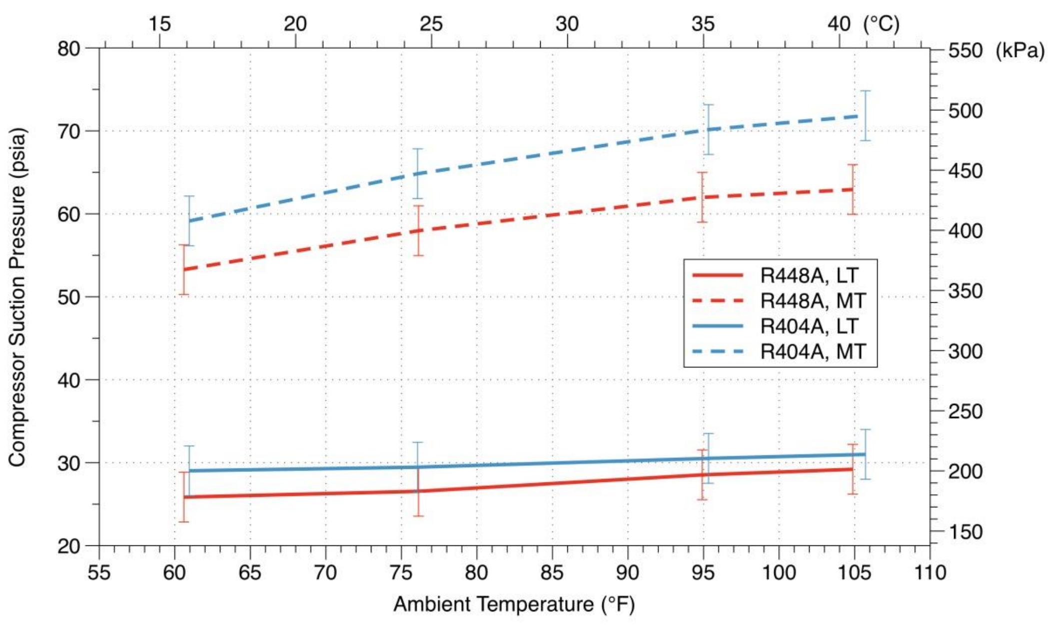

Fig. 30. Low-temperature (LT) and medium-temperature (MT) suction header pressures for R-448A and R-404A, using reciprocating compressors.

Finally, Fig. 31 shows the average low-temperature and medium-temperature display case discharge air temperatures for the commercial refrigeration system operating with either R-448A or R-404A, using reciprocating compressors. It can be seen that the display case discharge air temperature increases slightly as ambient temperature increases. In addition, the display case discharge air temperatures for R-448A are lower than those for R-404A. On average, the low-temperature display case discharge air temperature was 1.8°F (1.0°C) lower for R-448A as compared to R-404A, while the discharge air temperature for the medium-temperature display cases was 4.4°F (2.4°C) lower when operating with R-448A versus R-404A.

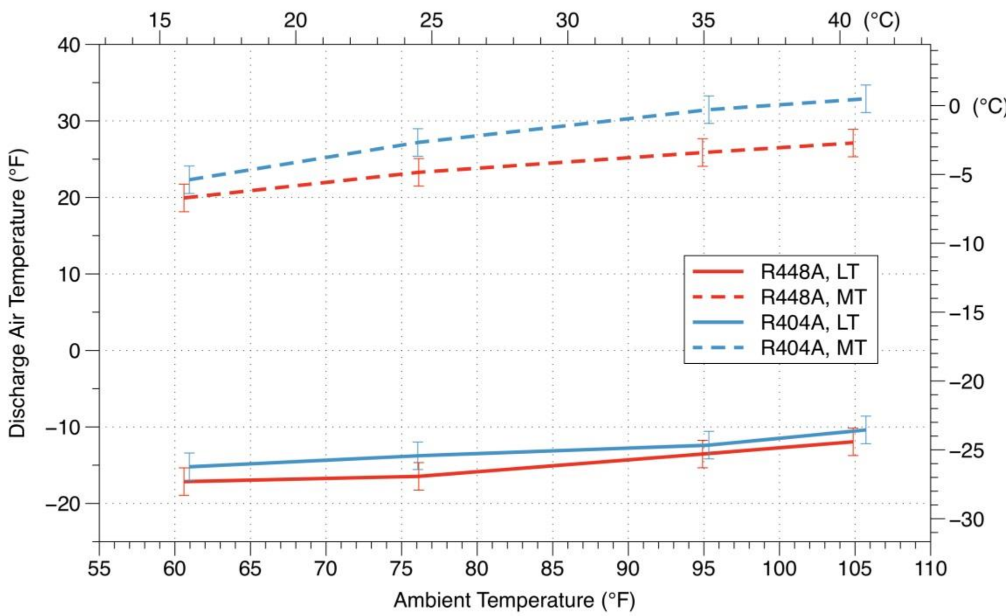

Fig. 31. Low-temperature (LT) and medium-temperature (MT) display case discharge air temperatures for R-448A and R-404A, using reciprocating compressors.

4.3.2 Summary of Results with Reciprocating Compressors

From the data presented above, it can be seen that R-448A and R-404A performed similarly in the laboratory-scale commercial refrigeration system while operating with the reciprocating compressors. Compared to R-404A, R-448A exhibited an energy benefit since system COP was increased and compressor power was decreased. On average, the efficiency of the system increased by 10% when operating with R-448A as compared to R-404A. The refrigeration capacity of R-448A was found to be slightly higher than that of R-404A. For the same saturated evaporating and condensing temperatures, compressor suction pressures of R-448A were lower than that of R-404A (2.5 psi to 7.5 psi lower) while compressor discharge temperatures of R-448A were higher (16°F to 20°F higher). Since system performance differences between R-448A and R-404A are small, it can be presumed that R-448A would be a suitable drop-in replacement refrigerant for R-404A. Only minor changes in the system would be required to retrofit R-404A with R-448A. Suction pressure setpoints would need to be reduced and expansion valve superheat settings would require adjustment.

4.3.3 Performance with Scroll Compressors

The total refrigerating capacity of the system, operating with scroll compressors using either R-448A or R-404A, is shown in Fig. 32. It can be seen that the system capacity decreases slightly as the ambient temperature increases. The capacity of the system when operating with R-448A and R-404A is approximately the same. This is not surprising, given that the display case temperatures and indoor chamber temperature were the same during both refrigerant evaluations.

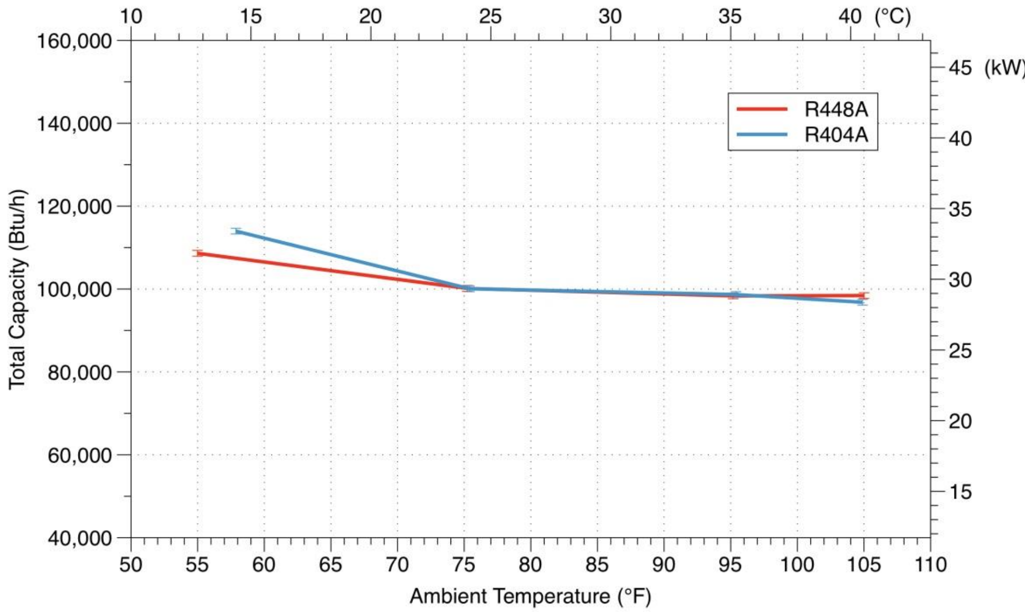

Fig. 32. Total system capacity for R-448A and R-404A, using scroll compressors.

The total scroll compressor power for the refrigeration system, operating with either R-448A or R-404A, is shown in Fig. 33. It can be seen that the total compressor power increases as the ambient temperature increases. The total compressor power when operating with R-448A is approximately 15% lower on average than that when operating with R-404A.

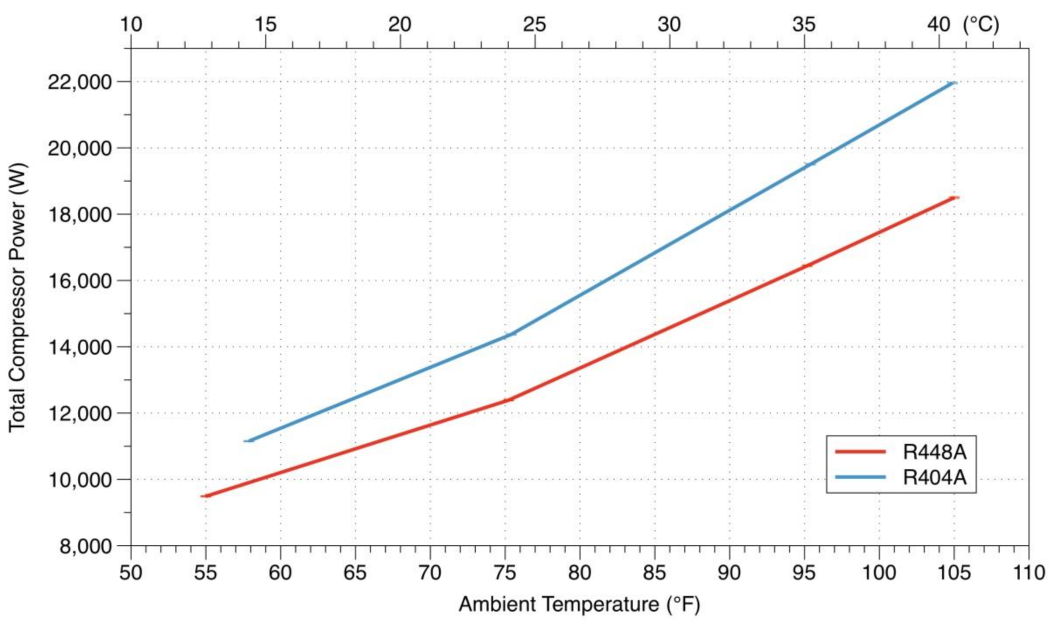

Fig. 33. Total compressor power for R-448A and R-404A, using scroll compressors.

The COP of the commercial refrigeration system, operating with either R-448A or R-404A, is shown in Fig. 34. As noted above, the capacity of the refrigeration system is essentially the same for the R-448A and R-404A tests. However, the compressor power associated with R-448A is lower than that of R-404A.

Overall, the COP of the system operating with R-448A was found to be approximately 16% greater on average than that when operating with R-404A. Since the load was the same, the energy savings, shown in Fig. 33 is a close match to the increase in COP illustrated in Fig. 34.

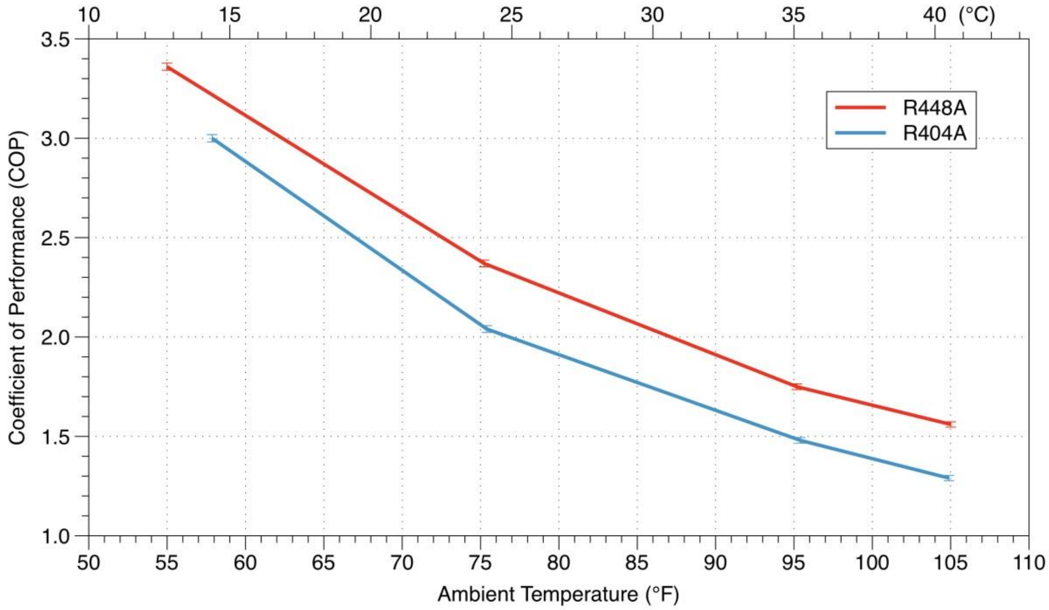

Fig. 34. Refrigeration system COP for R-448A and R-404A, using scroll compressors.

The total refrigerant mass flow rate for the commercial refrigeration system, operating with either R-448A or R-404A, is shown in Fig. 35. It can be seen that as the ambient temperature increases, the total refrigerant mass flow rate increases. On average, the total refrigerant mass flow rate of the system operating with R-448A was found to be approximately 28% lower than that when operating with R-404A.

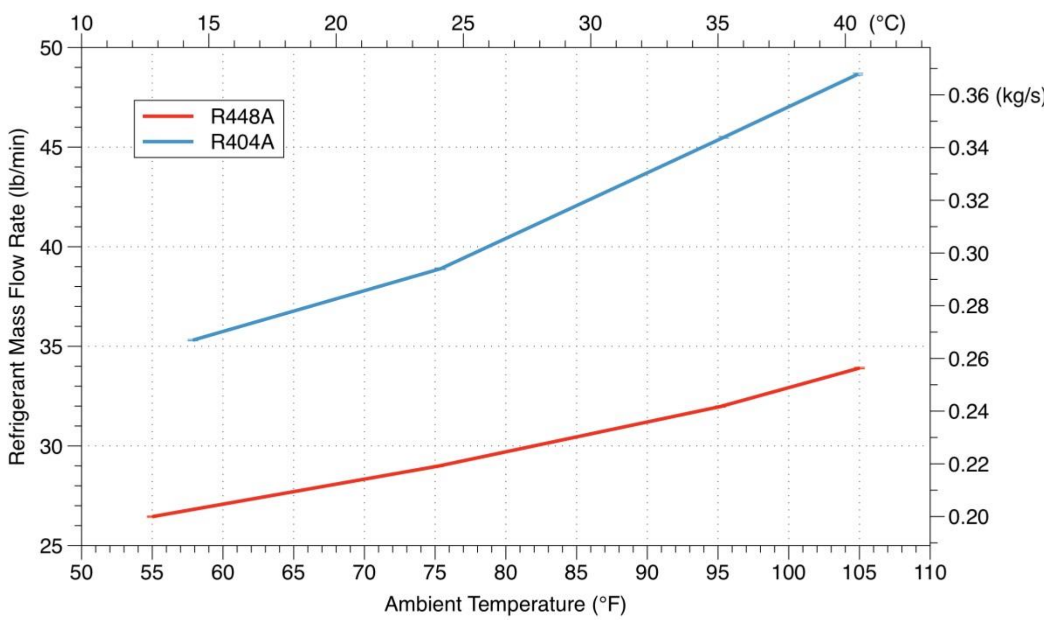

Fig. 35. Total refrigerant mass flow for R-448A and R-404A, using scroll compressors.

The low-temperature and medium-temperature compressor isentropic efficiencies for R-448A and R-404A are shown in Fig. 36, for the system operating with reciprocating compressors. It can be seen that as ambient temperature increases, the isentropic efficiency of the medium-temperature scroll compressors increases while the isentropic efficiency of the low-temperature scroll compressors decreases. In general, the isentropic efficiency of the scroll compressors when operating with R-448A was greater than that when operating with R-404A. The isentropic efficiency of the low-temperature scroll compressors, when operating with R-448A, was on average 19% greater than that when operating with R-404A. The difference in isentropic efficiency of the medium-temperature compressors between R-448A and R-404A was within the experimental uncertainty, and thus, the isentropic efficiency of the medium temperature compressors, when operating with R-448A, was essentially equal to that when operating with R-404A.

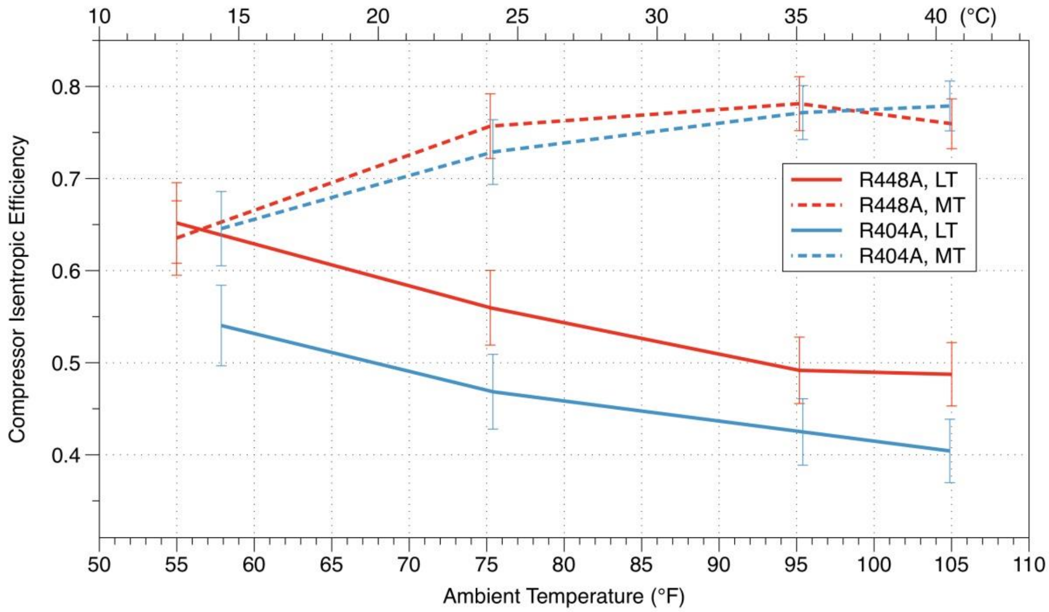

Fig. 36. Low-temperature (LT) and medium-temperature (MT) compressor isentropic efficiencies for R-448A and R-404A, using scroll compressors.

The low-temperature and medium-temperature compressor discharge temperatures, when operating with either R-448A or R-404A, are shown in Fig. 37. It can be seen that the compressor discharge temperature increases as ambient temperature increases. On average, the low-temperature scroll compressor discharge temperature using R-448A was 32°F (18°C) greater than that of R-404A, while the medium-temperature scroll compressor discharge temperature using R-448A was 20°F (11°C) greater than that of R-404A.

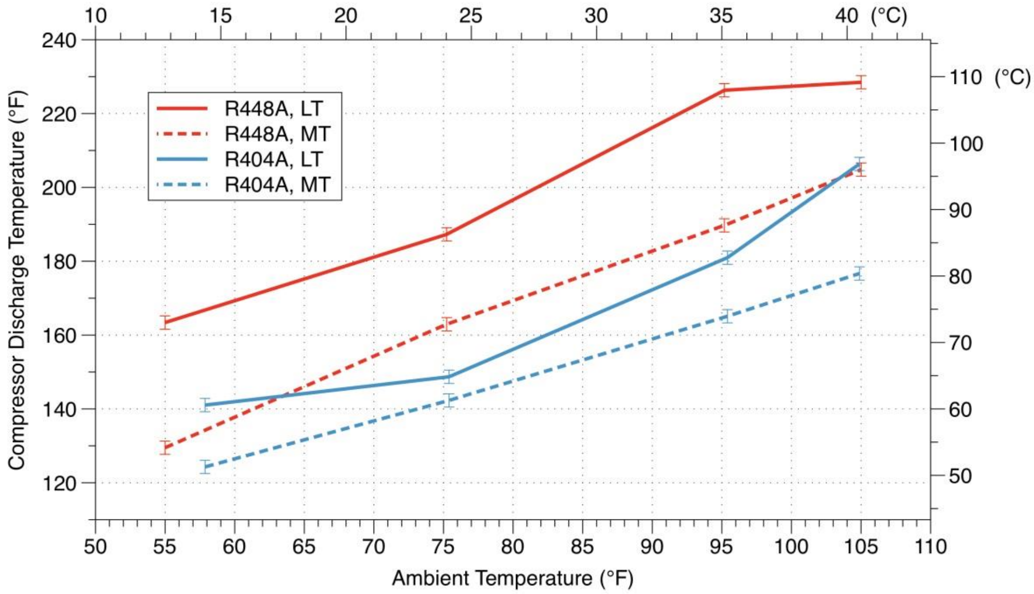

Fig. 37. Low-temperature (LT) and medium-temperature (MT) compressor discharge temperatures for R-448A and R-404A, using scroll compressors.

The low-temperature and medium-temperature suction header pressures for the refrigeration system, when operating with either R-448A or R-404A, are shown in Fig. 38, for the system operating with scroll compressors. On average, the low-temperature suction header pressure was 3 psi (21 kPa) lower with R-448A as compared to R-404A, while the medium-temperature suction header pressure was 5 psi (34 kPa) lower with R-448A as compared to R-404A.

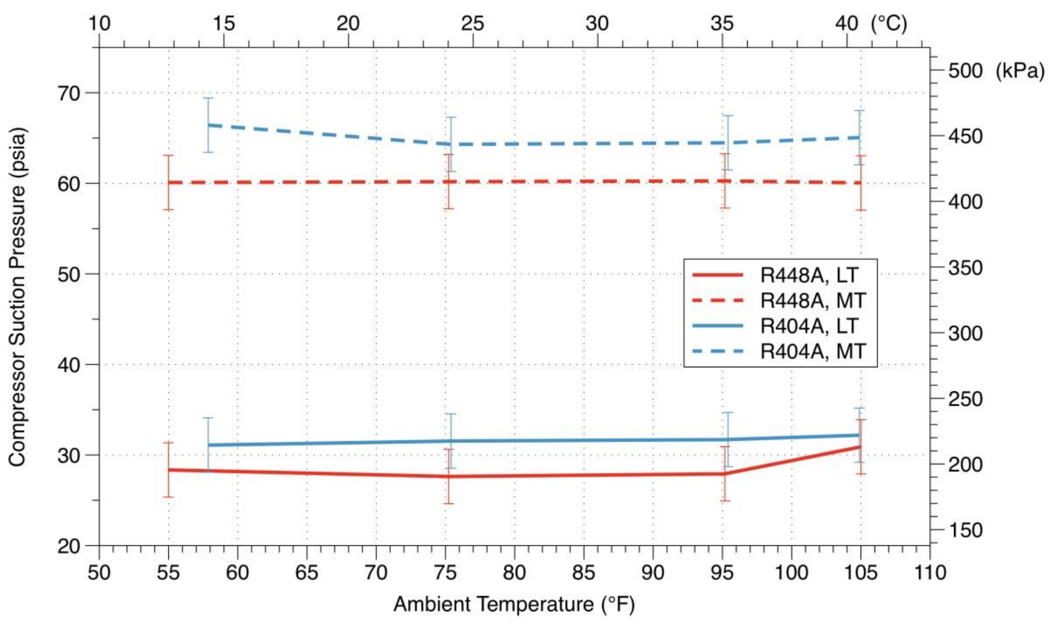

Fig. 38. Low-temperature (LT) and medium-temperature (MT) suction header pressures for R-448A and R-404A, using scroll compressors.

4.3.4 Summary of Results with Scroll Compressors

From the data presented above, it can be seen that R-448A and R-404A performed similarly in the laboratory-scale commercial refrigeration system while operating with scroll compressors. Compared to R-404A, R-448A exhibited an energy benefit since system COP was increased and compressor power was decreased. On average, the efficiency of the system increased by approximately 16% when operating with R-448A as compared to R-404A. The refrigeration capacity of R-448A and R-404A were the same since the display case operating temperatures and indoor chamber temperature were the same for both refrigerant evaluations. For the same saturated evaporating and condensing temperatures, compressor suction pressures of R-448A were lower than that of R-404A (3 psi to 5 psi lower) while compressor discharge temperatures of R-448A were higher (20°F to 32°F higher). Since system performance differences between R-448A and R-404A are small, it can be presumed that R-448A would be a suitable drop-in replacement refrigerant for R-404A. Only minor changes in the system would be required to retrofit R-404A with R-448A. Suction pressure setpoints would need to be reduced and expansion valve superheat settings would require adjustment.

4.3.5 Discussion

From the data presented above, it can be seen that R-448A and R-404A performed similarly in the laboratory-scale commercial refrigeration system. Compared to R-404A, R-448A exhibited an energy benefit since system COP was increased and compressor power was decreased. The refrigeration capacity of R-448A was found to be similar to that of R-404A. For the same saturated evaporating and condensing temperatures, compressor suction pressures of R-448A were lower than that of R-404A while compressor discharge temperatures of R-448A were higher. Since system performance differences between R-448A and R-404A are small, it can be presumed that R-448A would be a suitable drop-in replacement refrigerant for R-404A. Only minor changes in the system would be required to retrofit R-404A with R-448A. Suction pressure setpoints would need to be reduced and expansion valve superheat settings would require adjustment.

5. FIELD EVALUATION OF ALTERNATIVE REFRIGERANTS IN A COMMERCIAL REFRIGERATION SYSTEM

Due to concerns related to ozone depletion and global climate change, refrigerants such as R-22 and R-404A, which are commonly used in commercial refrigeration systems, have been, or may potentially be, phased out due to their detrimental environmental effects. Alternative refrigerants with no ozone depleting potential and lower global warming potential have been developed. The ideal alternative refrigerant should be environmentally safe, while at the same time, providing the same or better system performance, both in terms of energy consumption and refrigeration capacity. Thus, the main objective of this investigation is to determine the change in energy consumption of a commercial refrigeration system due to retrofitting a new refrigerant with lower environmental impact.

5.1 DESCRIPTION

Two similarly sized supermarkets will be identified. The stores will be located within close proximity to each to ensure that climate, weather, time-of-year and economic conditions of the shoppers are comparable. These existing stores will utilize R-22 or R-404A as the refrigerant. Details of each refrigeration system will be collected, such as:

- Basic system layout and refrigerant type
- MT and LT compressor specifications (capacity, power, quantity)
- Condenser specifications (capacity, fan power, quantity)
- Connected loads: display case and walk-in specifications (capacity, power, length/size)
- System control strategy (pressure and temperature setpoints, floating condensing/evaporating pressures, liquid subcooling, etc.)

The performance data which has already been collected from these stores will serve as the baseline for the refrigerant retrofit comparison. Thus, the quality of the existing data will be evaluated to ensure that comparisons to the baseline data are valid and meaningful. If the existing instrumentation and data acquisition equipment is found to be insufficient, additional instrumentation and data acquisition equipment will be added to the refrigeration systems.

Once the system performance data using the baseline refrigerant has been deemed sufficient, the alternative refrigerant. Following the refrigerant retrofit, refrigeration system operating parameters such as pressure setpoints and expansion valve superheat will be adjusted to obtain optimal performance. Finally, the above mentioned quantities will be recorded for a period sufficient to cover at least a cold season and a warm season.

The alternative refrigerant to be tested is R-448A (GWP = 1274), designed to replace R-404A (GWP = 3943) and R-22 (GWP = 1760) in existing equipment at a significantly lower GWP. Based on performance evaluations by the Oak Ridge National Laboratory and manufacturers in small scale supermarket facilities, R-448A is expected to maintain similar or better capacity and efficiency relative to R-404A and R-22.

A comparison of system performance (energy consumption) will be made between the baseline refrigerant and R-448A normalized to ambient conditions, operating conditions, normalized to system capacity. Other operating parameters such as suction and discharge pressures, refrigeration-side and case temperatures, refrigerant level and system parameters setpoints will be compared.

It is anticipated that the performance of the refrigeration systems will be determined for a period of no less than six months, with an effort to ensure that the system performance data is collected for both a cold and warm season.

5.2 DATA ACQUISITION

The state of the existing data acquisition system connected to each store's refrigeration system will be assessed to determine the following:

- Type of quantities measured
- Location and number of quantities measured
- Frequency of measurements
- Suitability of measured quantities to assess effect of refrigerant retrofit

It will be assumed that at a minimum, the following quantities should be measured to ensure that meaningful comparisons can be made between the baseline refrigerant and the alterative refrigerant:

- Compressor and condenser fans power
- Condensing pressure and temperature
- MT (medium-temperature) suction pressure and temperature
- LT (low-temperature) suction pressure and temperature
- Display case discharge air temperatures
- MT and LT liquid flow rate
- Refrigerant level
- Outdoor ambient temperature and relative humidity
- Store indoor temperature and relative humidity

Instrumentation and data acquisition equipment will be added to the refrigeration systems for those quantities which are not sufficiently measured or recorded.

5.3 FIELD EVALUATION REPORT

A detailed final report will be written to describe the results of the refrigerant retrofit comparison. This report will include the following:

1. High Level Objective
2. Description of Test Systems
    - Clear description of the new refrigeration technology
    - Clear description of the benchmark. Whenever possible, this baseline system should use reasonable-cost and up-to-date refrigeration technologies and fluids
3. Performance
    - Overall assumptions that should include operating conditions. Special attention should be paid to testing systems under comparable ambient conditions and/or locations
    - Identification of sources of differences using both theoretical and well documented field measurement based arguments
    - Energy measurements including both annualized energy and peak summer energy consumption, and a "bin-hour" ambient temperature graph for typical year for the location of the test stores
    - Inclusion of any reliability considerations
4. Economics
    - Main assumptions that could affect return on investment (ROI)
    - Realistic maintenance costs
5. Conclusions

5.4 SITE SELECTION

A major food retailer has agreed to participate in the field evaluation of R-448A using the refrigeration systems available at a supercenter located in Pineville, MO. The refrigerant in the existing refrigeration system is R-404A. Three central rack systems provide cooling to the store's refrigerated display cases and walk-ins, as shown in Table 23.

Table 23. Refrigeration rack specifications at field site

Refrigeration Rack	Cooling Capacity, Btu/h (kW)	Suction Temperature, °F (°C)
Rack A	305,000 (89.4)	-27 (-33)
Rack B	456,000 (134)	+13 (-11)
Rack C	555,000 (163)	+20 (-6.7)

Details of the existing refrigeration system configuration have been discussed with the store operators, including compressor racks, attached refrigeration loads, system controls and instrumentation. Additional instrumentation will be required, including compressor suction and discharge temperature, refrigerant liquid temperature after the condenser and power measurement for the compressor racks and condenser fans. The store operators will coordinate the installation of the necessary additional instrumentation required for the field study, and then data collection of baseline performance from the refrigeration system, using R-404A, will commence.

6. CONCLUSIONS

Environmental concerns are driving regulations and the HVAC&R industry towards lower GWP alternatives to HFC refrigerants. The selection of an appropriate alternative refrigerant for a given refrigeration application should be based on several factors, including the GWP of the refrigerant, the energy consumption of the refrigeration system over its operating lifetime, and leakage of refrigerant over the system lifetime. For example, focusing on energy efficiency alone may overlook the significant environmental impact of refrigerant leakage; while focusing on GWP alone might result in lower efficiency systems that result in higher indirect impact over the equipment lifetime.

Thus, the objective of this CRADA project between Honeywell and ORNL was to develop an LCCP modeling tool for optimally designing HVAC&R equipment with lower life cycle greenhouse gas emissions, and the selection of alternative working fluids to reduce the greenhouse gas emissions of HVAC&R equipment. In addition, an experimental evaluation program was utilized to measure the COP and refrigerating capacity of various refrigerant candidates.

An LCCP methodology was presented to determine the direct and indirect emissions associated with the lifetime operation of refrigeration systems, from system construction through system operation and system dismantling. The methodology was then incorporated into an open-source LCCP Design Tool which can be used to estimate the lifetime direct and indirect carbon dioxide equivalent gas emissions of various commercial refrigeration system designs and refrigerant options, with the goal of providing guidance on lower GWP refrigerant solutions with improved LCCP compared to baseline systems. The LCCP Design Tool is available in a desktop computer version and a simplified web-based version. Both versions may be accessed from http://lccp.umd.edu/.

The open-source LCCP Design Tool was used to compare the lifetime emissions of four commercial refrigeration system configurations, using four refrigerants (R-404A, R-448A, R-744 [CO_2], and L-40), in six US cities representing different climate zones. The four refrigeration systems include the multiplex direct expansion (DX) system, the cascade/secondary loop system, the transcritical CO_2 booster system, and the medium-temperature secondary loop/low-temperature DX system. Finally, a sensitivity analysis was performed to identify the relative importance of various input parameters on the calculated total CO_{2e} emissions of supermarket refrigeration systems.

From the LCCP analysis, it was found that the transcritical CO_2 booster system has the lowest CO_2 equivalent emissions in cold and temperate climates. Also, the R-448A/L-40 secondary circuit refrigeration system was found to offer a good balance between emissions and electricity consumption for hot climates. The parametric analysis showed that shifting towards low GWP refrigerants decreases the effect of the annual leak rate on the total system emissions. Moreover, the sensitivity analysis showed that shifting towards low GWP refrigerants, or more charge conservative systems increases the effect of the hourly emission rate for electricity production on the total system emissions. Finally, an uncertainty analysis was performed showing that using low GWP refrigerants, or more charge conservative systems causes a noticeable drop in the impact of the uncertainty in the inputs related to the direct emissions.

The energy performance of an alternative lower global warming potential refrigerant, R-448A, was evaluated in a laboratory-scale commercial refrigeration system, and its performance was compared to that of the commonly used higher GWP refrigerant, R-404A, found in the commercial refrigeration industry. The laboratory-scale commercial refrigeration system installed in the environmental test chambers at the Oak Ridge National Laboratory (ORNL) consists of components typically found in most U.S. supermarket refrigeration systems, including a compressor rack, an air-cooled condenser and several medium-temperature (MT) and low-temperature (LT) refrigerated display cases. The refrigeration system has a low-temperature cooling capacity of approximately 5 tons at −20°F (18 kW at −29°C) and a medium-temperature cooling capacity of approximately 10 to 15 tons at 25°F (35 to 53 kW at −4°C). The compressor rack of the commercial refrigeration system is unique in that it contains two types of compressors that are commonly found in supermarket refrigeration systems: reciprocating compressors and scroll compressors. Having both reciprocating and scroll compressors allows for system performance tests to be performed using the two common compressor types employed in supermarket refrigeration systems.

Using each refrigerant (R-404A and R-448A), the performance of the laboratory-scale commercial refrigeration system was determined at four ambient temperature conditions: 60°F, 75°F, 95°F and 105°F (16°C, 24°C, 35°C, and 41°C). It was found that R-448A and R-404A performed similarly in the laboratory-scale commercial refrigeration system. Compared to R-404A, R-448A exhibited an energy benefit since system COP was increased and compressor power was decreased. The refrigeration capacity of R-448A was found to be similar to that of R-404A. For the same saturated evaporating and condensing temperatures, compressor suction pressures of R-448A were lower than that of R-404A while compressor discharge temperatures of R-448A were higher. Since system performance differences between R-448A and R-404A are small, it can be presumed that R-448A would be a suitable drop-in replacement refrigerant for R-404A. Only minor changes in the system would be required to retrofit R-404A with R-448A. Suction pressure setpoints would need to be reduced and expansion valve superheat settings would require adjustment.

Future efforts related to this project include completing a field evaluation of the lower GWP alternative refrigerant in third-party supermarkets. The main objective of the field evaluation is to compare the energy consumption of refrigeration systems using incumbent refrigerants and alternative refrigerants in actual, operating supermarkets, thereby providing motivation to supermarket owners and operators to retrofit existing systems with the lower GWP alternative. Honeywell and ORNL are currently negotiating the site selection and logistics for the field evaluation of R-448A with a major food retailer.

Motivated by the outstanding energy and environmental performance of the alternative lower GWP refrigerant R-448A, the CRADA partner, Honeywell has commercialized the refrigerant (Solstice® N40) as an alternative to R-404A in both new and existing commercial refrigeration systems.

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