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Laboratory Evaluation of a Commercial CO_2 Booster Refrigeration System

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ABSTRACT

The traditional multiplex direct expansion (DX) refrigeration system used in commercial applications is prone to significant refrigerant leakage, especially older existing systems. The use of high Global Warming Potential (GWP) refrigerants in these systems, combined with high refrigerant leakage, can result in considerable direct carbon dioxide equivalent (CO_{2eq}) emissions. In addition, commercial refrigeration systems consume a substantial amount of electrical energy, resulting in high indirect CO_{2eq} emissions. Thus, there are ongoing efforts to reduce both 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.

Based on prior energy and life cycle climate performance (LCCP) analyses, it was determined that a transcritical CO_2 booster refrigeration system for supermarket applications has the potential to reduce carbon emissions and increase energy efficiency. To that end, a lab-scale transcritical CO_2 booster refrigeration system was fabricated and installed in the environmental test chambers at the Oak Ridge National Laboratory (ORNL). The performance of the transcritical CO_2 booster refrigeration system was evaluated over an outdoor ambient temperature range of 15.6 to 32.2°C. Compressor power was found to increase by approximately 78% over this same temperature range while the refrigeration load remained relatively constant. The resulting coefficient of performance (COP) of the system was found to vary from 2.2 (at an outdoor ambient temperature of 32.2°C) to 4.1 (at an outdoor ambient temperature of 15.6°C). Based on the laboratory evaluation, the transcritical CO_2 booster refrigeration system demonstrates promise as a low emission, high efficiency alternative to the traditional multiplex DX systems currently in use.

1. INTRODUCTION

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

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and the high refrigerant leakage rates leads to significant direct emissions of greenhouse gases into the atmosphere. Hence, these multiplex refrigeration systems can directly contribute to the increase in global warming. Furthermore, the operation of multiplex DX refrigeration systems also contributes to global warming indirectly. On an annual basis, the refrigeration system of a typical supermarket may consume approximately 1 million kWh (Zhang, 2006). Thus, the indirect impact on the environment results from the release of greenhouse gases (mainly CO_2) associated with the generation and transmission of the electrical energy used by the refrigeration system.

The direct environmental impact of the refrigeration system can be reduced by using refrigerants with lower GWP. Refrigerants such as R32, R134a, R717, R744, R290, R600a and R1234yf could be potential alternative refrigerants to the commonly used R404A. However, due to toxicity and/or flammability, some of these refrigerant options may not be permissible under various municipal safety codes. Cascade systems and secondary loop systems (discussed later in Section 2) using CO_2 as a refrigerant can be used to reduce the direct impact on the environment due to their lower HFC refrigerant charge.

The indirect environmental impact of the refrigeration system can be reduced by increasing the energy efficiency of the system. One option for increasing energy efficiency is to reduce the load on the refrigeration system. This can be done by replacing open display cases with doored display cases. Several studies have shown that doored display cases can reduce refrigeration system energy consumption by up to 50% when combined with high efficiency display case components such as LED lighting, demand defrost, electronically commutated evaporator fan motors and humidity controlled anti-sweat heaters (Rauss *et al.*, 2008; Fricke and Becker, 2010). Other energy efficiency measures that can be utilized include variable speed drives for compressors and condenser fan motors, as well as floating condensing and suction pressure controls.

Carbon dioxide has recently received considerable attention as an alternative to the commonly used synthetic refrigerants in supermarket refrigeration systems, in an effort to develop systems with lower environmental impact (Bansal, 2012; Getu and Bansal, 2008). Although CO_2 has a high critical pressure (7.38 MPa) and a low critical temperature (31.06°C), its high operating pressure leads to a high vapor density and thus a high volumetric refrigerating capacity. The volumetric refrigerating capacity of CO_2 (22,545 kJ m^{-3} at 0°C) is 3 to 10 times larger than CFC, HCFC, HFC and HC refrigerants (Kim et al. 2004). In addition, carbon dioxide has no Ozone Depletion Potential (ODP); a GWP of one; and is nontoxic, nonflammable and inexpensive – all attractive characteristics when compared to synthetic refrigerants.

Carbon dioxide has successfully been used as a refrigerant in the low-temperature circuit of cascade systems, in secondary loop systems, and in transcritical systems (Bansal, 2012; Girotto *et al.*, 2004; Hinde and Zha, 2009). However, transcritical CO_2 systems tend to be more popular in moderate climates such as Northern Europe where the refrigeration system operates a majority of the time in the more efficient subcritical mode (Denecke *et al.*, 2012; Sawalha and Palm, 2003).

In an effort to increase the efficiency of the transcritical CO_2 system and to make it applicable to warmer climates, several researchers have investigated the energy performance of the transcritical refrigeration system in various regions across Europe (Bell, 2004; Ferrandi and Orlandi, 2012; Ge and Tassou, 2009, 2010; Mazzola et al., 2012; Sarkar and Agrawal, 2010; Sawalha, 2007; Winter and Murin, 2012). However, these studies were based on thermodynamic analyses and lacked experimental results to validate system performance. In this paper, the performance of a laboratory-scale transcritical CO_2 booster refrigeration system was evaluated for supermarket applications in the United States. In addition, the system performance was compared against that of a similar HFC-based multiplex DX system.

2. REFRIGERATION SYSTEM

The high-efficiency, low-emission commercial refrigeration system installed in the environmental test chambers at the Oak Ridge National Laboratory (ORNL) consisted of a transcritical CO_2 compressor rack, an air-cooled gas cooler/condenser, medium-temperature (MT) and low-temperature (LT) refrigerated display cases, and MT and LT "false" loads.

A piping diagram for the transcritical CO_2 compressor rack and gas cooler is shown in Figure 1. The liquid line at the exit of the flash tank supplies liquid CO_2 to the MT and LT display cases and false loads. The superheated CO_2

from the exit of the LT display case and false load returns to the compressor rack at the LT suction header, which feeds the LT compressors. The superheated CO_2 from the MT display case and false load returns to the compressor rack at the MT suction header and mixes with the LT compressor discharge and the flash gas from the flash tank. This vapor is then compressed in the MT compressors and the discharge is cooled via the air-cooled gas cooler/condenser.

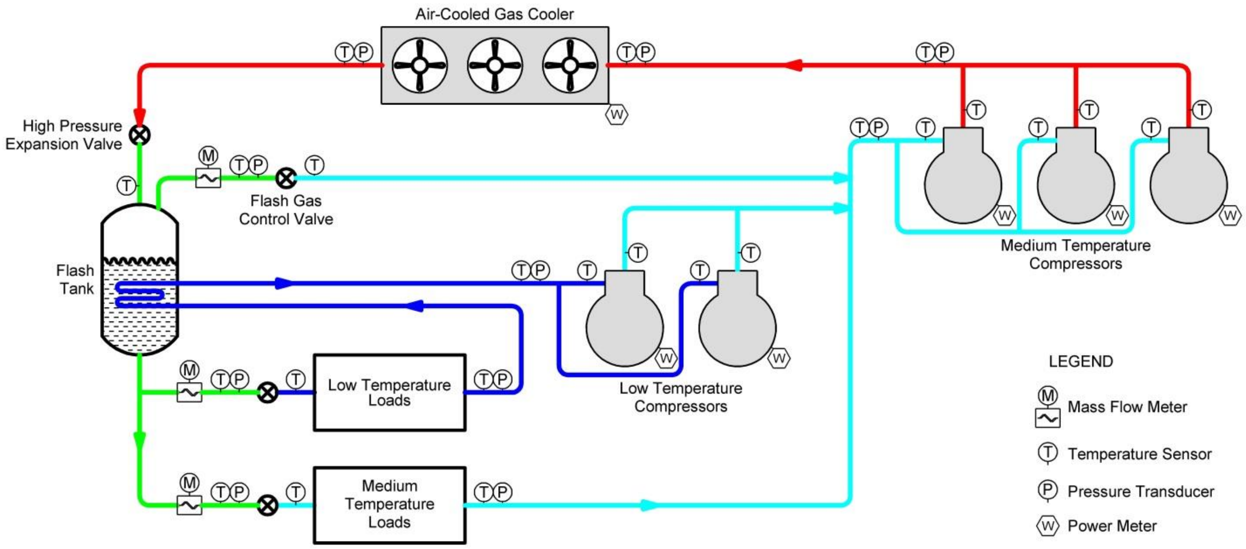

Figure 1: Schematic of the laboratory-scale transcritical CO_2 booster refrigeration system

The laboratory-scale refrigeration system has a low-temperature cooling capacity of approximately 9.1 kW at a saturated evaporating temperature of −30°C and a medium-temperature cooling capacity of approximately 34 kW at at a saturated evaporating temperature −6.7°C. One 4-door vertical display case, 3.0 m in length, as well as a "false" load provided by a plate heat exchanger and a glycol loop, constitutes the low-temperature load. The medium-temperature load consists of one open vertical display case, 2.4 m in length, as well as a "false" load provided by a plate heat exchanger and glycol loop.

The air-cooled gas cooler/condenser is installed in a temperature and humidity controlled "outdoor" environmental chamber while the compressor rack and refrigerated display cases are installed in a separate temperature and humidity controlled "indoor" environmental chamber. For both chambers, the temperature can be controlled between −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.

The compressor rack consists of several refrigerant compressors and the associated piping and valving which forms the liquid headers and low-temperature and medium-temperature suction headers as well as the discharge header. The refrigeration loads (i.e., the refrigerated display cases) are connected to the liquid and suction headers of the compressor rack and the gas cooler/condenser is connected to the discharge header of the compressor rack.

The compressor rack contains two low-temperature reciprocating compressors and three medium-temperature reciprocating compressors. The low-temperature compressors operate sub-critically, while the medium-temperature compressors can operate either sub-critically or super-critically, depending upon the ambient conditions. In addition, for each temperature level (i.e., LT or MT), the compressor rack contains a primary compressor that is variable capacity (via a variable frequency drive), and one or two secondary compressors that are 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(s) operate as well, with the primary compressor modulating its capacity to match the load. The manufacturer's model, evaporator capacity and power requirements for the reciprocating CO_2 compressors are given in
Table **1**.

Table 1: Compressor specifications

Compressor Type	Temperature Level	Model	Capacity Control	Evaporator Capacity (kW)*	Power (kW)*
Reciprocating	LT	2KSL-1K	Variable	5.57	1.34
Reciprocating	LT	2MSL-07K	Fixed	3.52	0.82
Reciprocating	MT	4MTC-10K	Variable	11.1	9.66
Reciprocating	MT	4MTC-10K	Fixed	11.4	9.72
Reciprocating	MT	4MTC-7K	Fixed	11.3	9.40

*Evaporator capacity and power are given for the following operating conditions using R-744 (CO_2):
LT: −30°C saturated evaporating temperature, −6.7°C saturated condensing temperature
MT: −6.7°C saturated evaporating temperature, 38°C gas cooling temperature

The low-temperature display case is a 4-door model with a length of 3.0 m, with a rated capacity of 1.67 kW. The medium-temperature display case is an open, vertical multi-deck model, 2.4 m in length, with a rated capacity of 2.81 kW. The rated capacities are determined by the manufacturer according to ASHRAE Standard 72 (2005).

Each display case contains one evaporator and one electronic expansion valve (EEV). Furthermore, each display case is controlled by its own case controller (EEV, temperature set-point, defrost, etc.). The LT and MT display cases utilize LED lighting and electronically commutated (EC) evaporator fan motors. In addition, the LT display case has anti-sweat heaters. Finally, the LT display case utilizes electric defrost heaters while the MT display case utilizes off-cycle defrost. The specifications for the low-temperature and medium-temperature display cases are shown in Table 2.

Table 2: Refrigerated display case specifications

Case Parameter	Low-Temperature Display Case	Medium-Temperature Display Case
Model Number	6RZLH	O5DM-NRG
Type	4-door, vertical multi-deck	Open, vertical multi-deck
Length	3.0 m	2.4 m
Rated Capacity	1.67 kW	2.81 kW
Fan Amperage	0.93 amps	0.75 amps
Lighting Amperage	0.90 amps	0.40 amps
Anti-Sweat Heater Amperage	7.99 amps	--
Defrost Type	Electric	Off-cycle
Defrost Amperage	16.29 amps	--

In addition to the refrigerated display cases, low-temperature and medium-temperature “false” loads are also incorporated into the refrigeration system. The false loads consist of plate heat exchangers with one refrigerant circuit on one side of the heat exchanger and a glycol circuit on the other side. The LT false load can provide an additional load of up to approximately 6.4 kW to the low -temperature side of the refrigeration system, while the MT false load can provide an additional load of up to approximately 27.0 kW to the medium-temperature side of the refrigeration system.

As shown in the system schematic in Figure 1, an air-cooled gas cooler/condenser is used to reject heat from the refrigeration system. The condenser accepts the discharge refrigerant vapor from the compressor rack, cools or condenses the refrigerant, and discharges the cooled refrigerant into a flash tank. The air-cooled gas cooler/condenser has two variable speed fans and its rated heat rejection capacity, with R-744, is 78.5 kW at an entering gas temperature of 117°C and a leaving gas temperature of 36.4°C.

The refrigeration system controller provides the following control for the system:

- Compressor control to maintain suction pressure setpoints for the LT and MT circuits
- Condenser fan speed control to maintain condensing pressure

In addition to the system controller, each display case has an individual case controller which communicates with the system controller and regulates expansion valve opening, display case air temperature, defrost operation, and lighting and fan operation. Also, since the false loads utilize electronic expansion valves, electronic controllers are required for the two false loads.

The lab-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. The specifications of the instrumentation are given in Table 3.

Table 3: Specifications of instrumentation

Instrument	Measurement	Measurement Range	Accuracy
Watt transducer	Compressor power, gas cooler fan power	0 to 4,000 W 0 to 8,000 W 0 to 80,000 W	±0.5% of reading
Type-T thermocouple	Refrigerant temperature, display case discharge and return air temperature	-270 to 400°C	1.0°C or 0.75% (whichever is greater) for 0 to 350°C
Temperature/humidity sensor	"Indoor" and "outdoor" chamber temperature and humidity	Humidity: 0 to 100% RH Temperature: -40 to 80°C	±1.7% RH for 0 to 90% RH ±0.2°C for 15 to 25°
Pressure transducer	Refrigerant pressure	0 to 7 MPa 0 to 14 MPa	±0.25% full scale
Coriolis mass flow meter	Refrigerant mass flow	0 to 10 kg/min	±0.05%
Positive displacement flow meter	Refrigerant mass flow, glycol mass flow	0.11 to 26.4 L/min	±0.5% of reading

3. LABORATORY PERFORMANCE DATA

3.1 Refrigeration Capacity

The medium-temperature and low-temperature refrigeration loads for the transcritical CO_2 refrigeration system as a function of outdoor ambient temperature are shown in Figure 2. The data presented in Figure 2 is for the combination of LT and MT display cases and false loads. For each display case and false load, the refrigeration load was determined from the inlet and outlet enthalpies and refrigerant mass flow rate through the load as follows:

$$\dot{Q}_i = \dot{m}_i\left(h_{i,out} - h_{i,in}\right) \tag{1}$$

where $\dot{Q}_i$ is the refrigeration load of load *i*, $\dot{m}_i$ is the refrigerant mass flow rate through load *i*, $h_{i,out}$ is the enthalpy of the refrigerant exiting load *i*, and $h_{i,in}$ is the enthalpy of the refrigerant entering load *i*. The enthalpies of the refrigerant were determined from the NIST Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) Version 9.1 (Lemmon *et al.*, 2013), using measured refrigerant pressures and temperatures at the corresponding locations.

It can be seen that the MT and LT refrigeration loads remain fairly constant over the outdoor temperature range from 15.6°C to 32.2°C. This is not unexpected, since the "indoor" conditions surrounding the display cases were fixed at 23.9°F and 55% RH, and thus, the load on the display cases did not vary. On average, the total LT load was approximately 7.6 kW while the total MT load was approximately 18.2 kW.

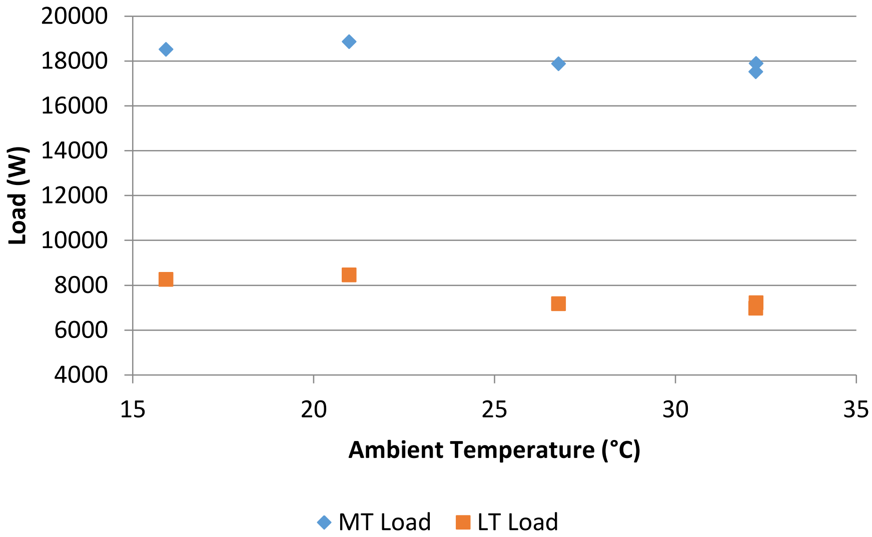

Figure 2: Medium-temperature (MT) and low-temperature (LT) refrigeration loads

3.2 Total Compressor Power

The total compressor power for the transcritical CO_2 refrigeration system as a function of outdoor ambient temperature is shown in Figure 3. The total compressor power is the sum of the power supplied to the MT and LT compressors, as measured by the individual power transducers on each compressor, and this performance data was obtained with the combination of LT and MT display cases and false loads as shown in Figure 2. It can be seen that as the outdoor ambient temperature increases, the total compressor power increases. Over the outdoor ambient temperature range from 15.6°C to 32.2°C, the total compressor power increased from approximately 6.50 kW to 11.5 kW, an increase of approximately 76%.

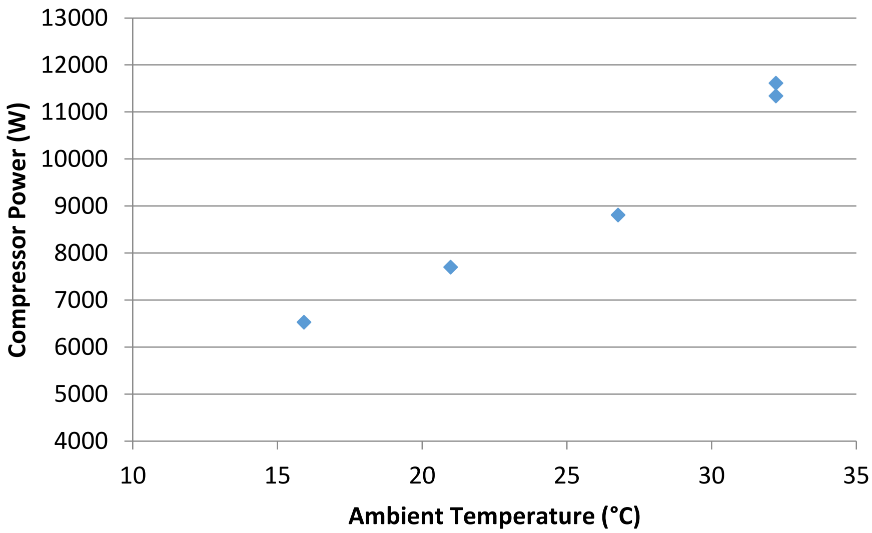

Figure 3: Total compressor power

3.3 Coefficient of Performance (COP)

Combining the refrigeration loads and compressor power, the coefficient of performance (COP) of the transcritical CO_2 booster refrigeration system can be calculated as a function of outdoor ambient temperature, according to:

$$COP = \frac{\dot{Q}_{total}}{\dot{W}_{total}} \tag{2}$$

where $\dot{Q}_{total}$ is the total refrigerating capacity and $\dot{W}_{total}$ is the total compressor power. As shown in Figure 4, the COP, or efficiency, of the system is greatest at lower outdoor ambient temperatures, and the COP was found to vary from 4.1 to 2.1 over the outdoor ambient temperature range from 15.6°C to 32.2°C.

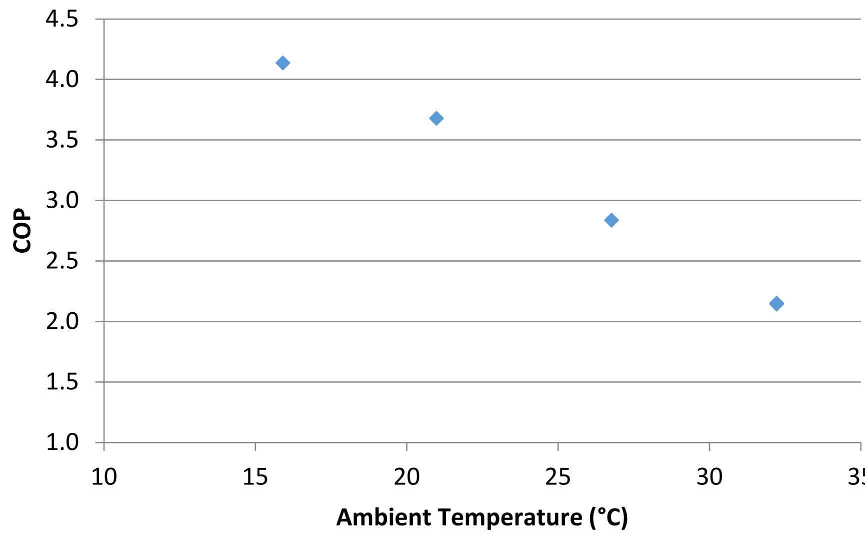

Figure 4: Coefficient of performance (COP) for the transcritical CO_2 refrigeration system

3.4 Gas Cooler Pressure

The refrigeration system controller maintains the gas cooler pressure at a value which optimizes the COP of the system for a given outdoor ambient temperature. At an outdoor ambient temperature of 32.2°C, the gas cooler pressure is approximately 9.1 MPa, while at an outdoor ambient temperature of 15.6°C, the gas cooler pressure is 6.0 MPa. Contrast these high operating pressures with that of an HFC multiplex DX system using R404A, which would have a condensing pressure of approximately 1.8 MPa at an outdoor ambient temperature of 32.2°C.

3.5 Compressor Discharge and Gas Cooler Temperatures

The refrigerant temperatures at the discharge of the MT compressors, as well as at the inlet and outlet of the gas cooler were found to increase as the outdoor ambient temperature increases. The gas cooler inlet temperature was found to range from 71 to 99°C, while the gas cooler outlet temperature ranged from 19 to 34°C, over the outdoor ambient temperature range of 15.6°C to 32.2°C. The high discharge temperature (or gas cooler inlet temperature) during supercritical operation indicates an opportunity for utilizing the rejected heat to offset some or all of the water heating or space heating needs of a supermarket.

3.6 Comparison of CO_2 and HFC-Based Refrigeration Systems

The performance of the transcritical CO_2 booster refrigeration system was compared to that of a similar sized laboratory-scale HFC-based multiplex DX system using R-404A as the refrigerant. The HFC-based refrigeration system has a low-temperature cooling capacity of approximately 18 kW at a saturated evaporating temperature of −29°C and a medium-temperature cooling capacity of approximately 35 to 53 kW at a saturated evaporating temperature of −4°C. Three open vertical display cases, each 3.7 m in length, constitute the low-temperature load. The medium-temperature load consists of two open vertical display cases, each 3.7 m in length, as well as a “false” load provided by a plate heat exchanger and glycol loop. The system contains two LT and two MT reciprocating compressors as well as an air-cooled condenser. Furthermore, this HFC-based refrigeration system is similarly instrumented to determine its performance.

The coefficients of performance of both the transcritical CO_2 booster and the HFC-based refrigeration systems are shown in Figure 5, as a function of outdoor ambient temperature. It can be seen that over the outdoor ambient temperature range of 15.6ºC to approximately 31.1ºC, the COP of the transcritical CO_2 booster system is greater than that of the HFC system. At 15.6ºC, the COP of the transcritical CO_2 booster system was nearly 20 % higher than that of the HFC system, and on average between 15.6ºC and 31.1ºC, the COP of the CO_2 booster system was 15% greater than the HFC system. Extrapolating the trends above 31.1ºC, it is expected that the HFC-based refrigeration system will have a greater COP than the transcritical CO_2 booster system.

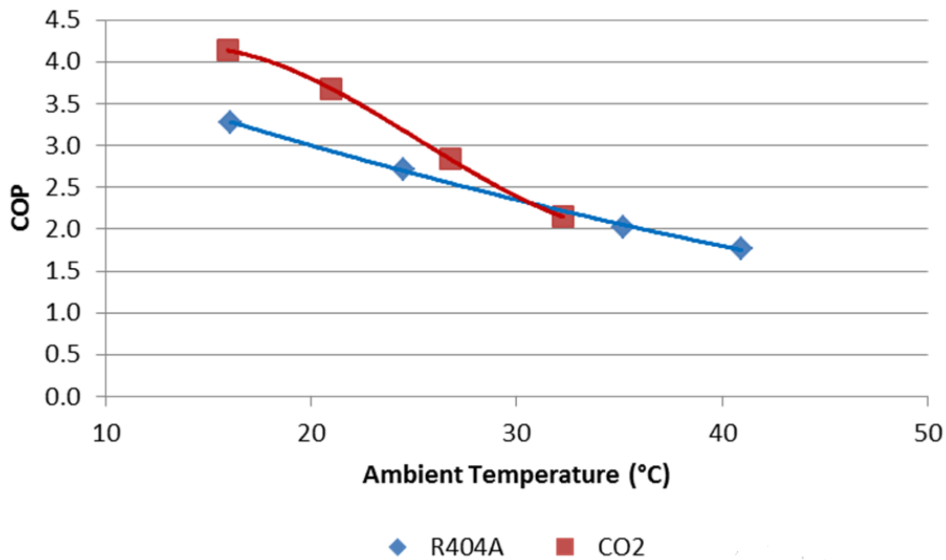

Figure 5: Comparison of COP for R404A multiplex DX and transcritical CO_2 booster refrigeration systems

For those climate zones with ambient temperatures that fall mainly below 31.1ºC, the transcritical CO_2 booster refrigeration system would offer an energy benefit compared to the traditional HFC-based multiplex DX system.

CONCLUSION

In this paper, the performance of a laboratory-scale transcritical CO_2 booster refrigeration system was evaluated under controlled operating conditions. This system consisted of a transcritical CO_2 compressor rack, an air-cooled gas cooler/condenser, medium-temperature (MT) and low-temperature (LT) refrigerated display cases, and MT and LT "false" loads. The lab-scale refrigeration system has a low-temperature cooling capacity of approximately 9.1 kW at a saturated evaporating temperature of −30°C and a medium-temperature cooling capacity of approximately 34 kW at a saturated evaporating temperature of −6.7°C. The air-cooled gas cooler/condenser is installed in a temperature and humidity controlled "outdoor" environmental chamber while the compressor rack and refrigerated display cases were installed in a separate temperature and humidity controlled "indoor" environmental chamber.

The performance of the transcritical CO_2 booster refrigeration system was determined at four outdoor ambient temperature conditions ranging from 15.6°C to 32.2°C. Over the outdoor ambient temperature range of 15.6 to 32.2°C, the total load on the system was found to remain relatively constant. In addition, the compressor power was found to increase by approximately 78% over this same temperature range. Thus, the resulting coefficient of performance (COP) of the system was found to vary from 2.2 (at an outdoor ambient temperature of 32.2°C) to 4.1 (at an outdoor ambient temperature of 15.6°C). In addition, the coefficients of performance of both the transcritical CO_2 booster and an HFC-based refrigeration systems were compared, and it was found that over the outdoor ambient temperature range of 15.6ºC to approximately 31.1ºC, the COP of the transcritical CO_2 booster system was on average 15% greater than that of the HFC system. Based on the laboratory evaluation, the transcritical CO_2 booster refrigeration system demonstrates promise as a low emission, high efficiency alternative to the traditional multiplex DX systems currently in use.

NOMENCLATURE

CFC	chlorofluorocarbon	
CO_{2eq}	carbon dioxide equivalent emissions	
COP	coefficient of performance	
DX	direct expansion	
EC	electronically commutated	
EEV	electronic expansion valve	
GWP	global warming potential	
HC	hydrocarbon	
HCFC	hydrochlorofluorocarbon	
HFC	hydrofluorocarbon	
LCCP	life cycle climate performance	
LED	light emitting diode	
LT	low-temperature	
MT	medium-temperature	
ODP	ozone depletion potential	
ORNL	Oak Ridge National Laboratory	
h	enthalpy of the refrigerant	(kJ/kg)
$\dot{m}$	refrigerant mass flow rate	(kg/s)
$\dot{Q}$	refrigeration load	(W)
$\dot{W}$	compressor power	(W)

Subscript

i	i^{th} load
in	inlet
out	outlet
total	total

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