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Building Technologies Office 03.02.04.100 Milestone Report—Evaluation of Low Environmental Impact Distributed Scroll Booster Technology for Supermarket Refrigeration

Samuel Yana Motta
Junjie Luo
Bo Shen
Vishal Sharma

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BUILDING TECHNOLOGIES OFFICE 03.02.04.100 MILESTONE REPORT—EVALUATION OF LOW ENVIRONMENTAL IMPACT DISTRIBUTED SCROLL BOOSTER TECHNOLOGY FOR SUPERMARKET REFRIGERATION

Samuel F. Yana Motta
Junjie Luo
Bo Shen
Vishal Sharma

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OAK RIDGE NATIONAL LABORATORY
Oak Ridge, TN 37831
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ABBREVIATIONS

COP	coefficient of performance
DOE	US Department of Energy
DX	direct expansion
GHG	greenhouse gas
GWP	global warming potential
HFC	hydrofluorocarbon
HPDM	Heat Pump Design Model
LCCP	life cycle climate performance
LT	low temperature
MT	medium temperature
ORNL	Oak Ridge National Laboratory
VI	vapor injection

1. INTRODUCTION

A distributed scroll booster refrigeration system is proposed as an alternative for current refrigeration systems used in supermarket applications. The distributed scroll booster system is configured to enable the use of low-pressure refrigerants such as R-1234yf, which has a global warming potential (GWP) of less than 1, in a cycle configuration that employs a two-stage compression process. This scroll booster system is composed of medium-temperature (MT) compressors (high stage) coupled with low-temperature (LT) compressors (low stage). This two-stage system architecture allows both LT and MT refrigerated display cases to be integrated in the same system. By design, this system reduces the refrigerant charge, produces higher efficiencies (two-stage compression), uses existing components to minimize cost impact, and will be able to use ultra-low GWP class A2L (low toxicity and low flammability) refrigerants such as R-1234yf when standards and building codes are updated (these updates are expected to occur by 2025).

Supermarkets and other large food retail stores commonly use multiplex direct expansion (DX) refrigeration systems in conjunction with hydrofluorocarbon (HFC) refrigerants such as R-404A or R-407C. These systems contain refrigerant charges on the order of several thousands of pounds with extremely long refrigerant piping runs to and from the refrigerated display cases. The estimated annual refrigerant leak rate for these systems ranges from 5% to 35% for in-use equipment, with the higher annual leak rates (>25%) more characteristic of older equipment and the lower rates (<15%) more characteristic of newer equipment (ICF Consulting, 2005). The high GWP of the HFC refrigerants commonly used in these systems, coupled with the large refrigerant charge, long refrigerant piping runs, and high refrigerant leakage rates, leads to significant direct emissions of greenhouse gases (GHGs) into the atmosphere.

In this project, a novel distributed scroll booster refrigeration system (shown in Figure 1) will be investigated for its potential to reduce both energy consumption and GHG emissions. Rather than a large centralized system approach, a distributed system architecture is proposed in which several smaller refrigeration systems are distributed throughout the store to serve both the MT and LT refrigeration loads. The distributed nature of the system results in reduced refrigerant charge in any one refrigeration system, as well as reduced lengths of refrigerant piping runs to and from the display cases. Thus, the magnitude and probability of refrigerant leaks will be reduced compared with leaks in the centralized system approach. In addition, the two-stage compression configuration of the system, in combination with the use of a low GWP, low-pressure refrigerant (R-1234yf), will result in an increase in energy efficiency. Finally, the distributed nature of the design allows refrigeration to be easily scaled up or down to meet the needs of large- and small-format stores.

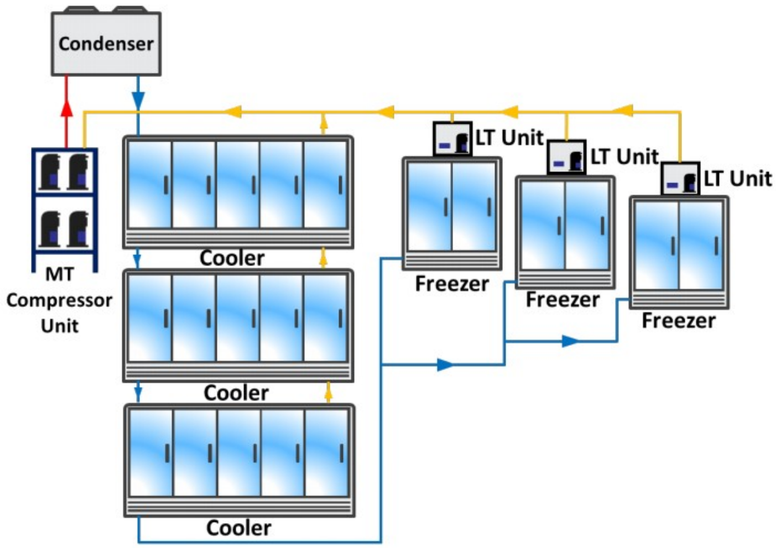

Figure 1. Distributed scroll booster equipment. (Source: Saunders et al., 2021.)

1.1 SCOPE OF THIS REPORT

The main objectives of this proposed research project can be met through the following three tasks:

1. Modeling and simulation of the system
2. Performance testing at controlled ambient conditions
3. Field evaluations

This report will cover modeling and a description of the experimental setup. The results of performance testing and field evaluations will be provided in a second report.

2. SYSTEM MODELING

The modeling work was based on the Heat Pump Design Model (ORNL, 2024) from the US Department of Energy's (DOE's) Oak Ridge National Laboratory (ORNL). HPDM is a public-domain modeling tool for use in steady-state and quasi-steady-state design analyses of extensive thermal system configurations. HPDM is a flagship software developed with DOE support to model and design building equipment in the heating, ventilation, air conditioning, refrigeration, and water heating sectors. The detailed equipment design tool is used widely by international scholars and US engineers, has been adopted by many equipment manufacturers, and has supported development of thousands of products. It has supported many DOE initiatives to assess feasibility of products such as high-efficiency rooftop units, cold climate heat pumps, and low GWP refrigerants. It is the most used and most influential public-domain building equipment modeling software in the industry and academia.

Following are descriptions of some of the HPDM features and capabilities:

- Comprehensive knowledge-based tool contains state-of-the-art developments in object-oriented programming, numerical recipes, advanced solver and optimization algorithms, property handling, and fundamental thermal science at component and system levels.

- Hardware-based equipment design tool promotes energy saving technologies and provides technical support to transfer new technologies from ORNL to US industry.

- Flexible component-based platform models extensive building equipment types and complicated configurations.

- Comprehensive library simulates details of most heating, ventilation, air conditioning, refrigeration, and water heating components.

- Extensive fluid properties capability handles air, water, glycol, hydrochlorofluorocarbon, HFC, new hydrofluoro-olefin, and natural (CO_2, propane) refrigerants.

- Extensive co-simulation capability integrates with EnergyPlus and Modelica to facilitate building energy co-simulations, control development, fault diagnosis, and energy storage.

In addition to an online version, HPDM is also provided as a free, downloadable desktop version with a Python wrapper.

Using the HPDM, we modeled a supermarket scroll-booster system as shown in Figure 2. It uses five vapor injection (VI) compressors. Two compressors control the LT capacity and refrigerant flow rate, which discharge vapor mixed with the refrigerant flow out of the MT evaporators; this mixture then enters the three MT VI compressors. Because a vapor injection system is considered a two-stage compression, the overall system is a four-stage compression. The system uses a microchannel condenser, with a liquid receiver at the exit to control a near-zero subcooling degree. The VI compressors are coupled with economizers to control intermediate vapor flow to the injection ports at a constant superheat degree regulated by upstream electronic expansion valves. The two-evaporator pressure regulating valves control the liquid at a constant temperature, entering the MT evaporators at 60°F and entering the LT evaporators at 30°F.

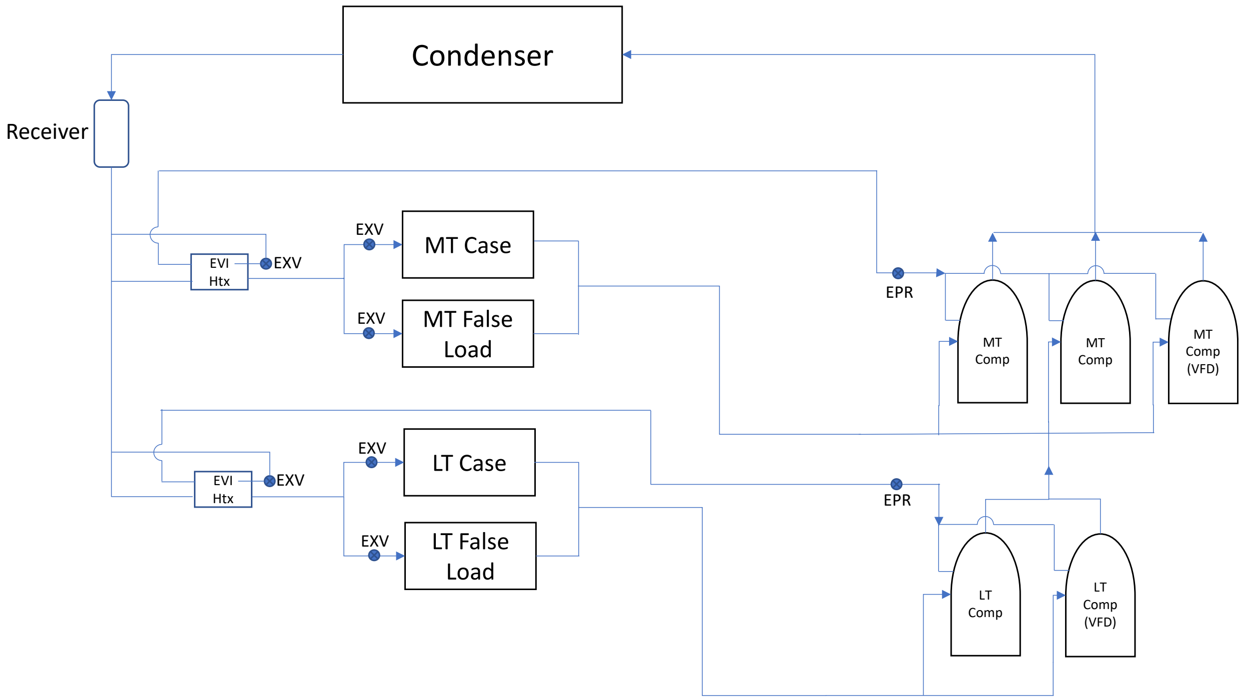

Figure 2. Flow diagram of the scroll booster system studied by ORNL.
(Comp = compressor; EVI Htx = Heat exchanger for vapor injected compressor;
EXV = electronic expansion valve; VFD = variable frequency drive)

HPDM modeled the vapor injection compressors as a two-stage compression with interstage injection. The mass flow and energy balances are illustrated in Figure 3. The displacement volumes of the low and high stages should be given according to the compressor hardware.

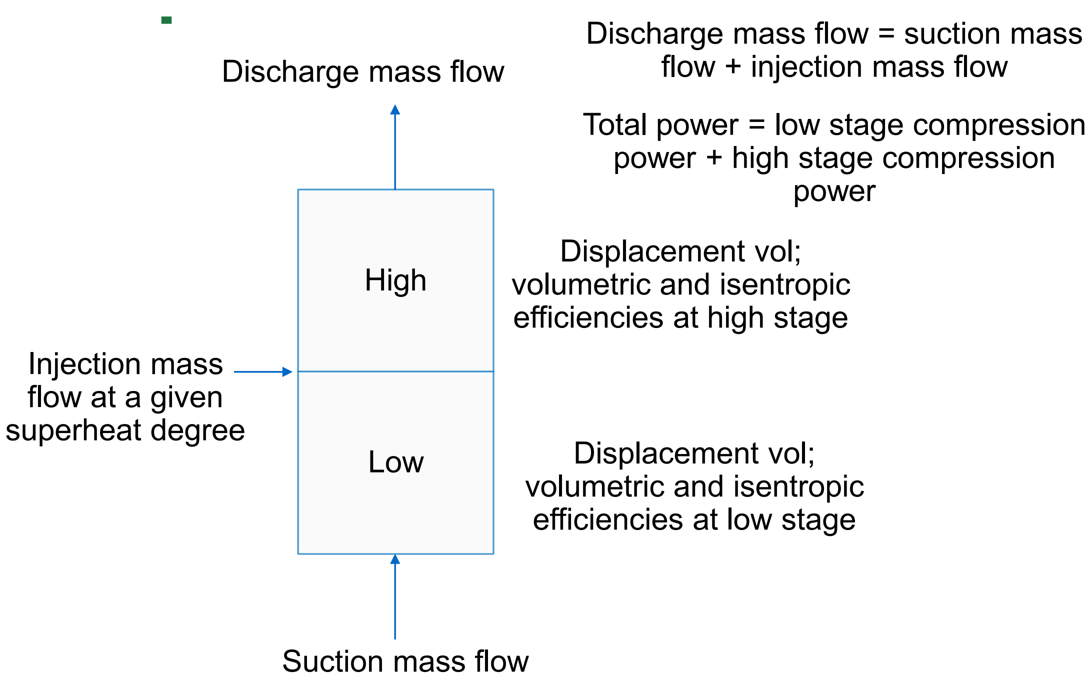

Figure 3. Two-stage compression with interstage injection.

Compressor manufacturers provides 10-coefficient compressor maps for vapor injection compressors, to correlate the compressor suction mass flow rate, power consumption, and injection saturation temperature as a function of the suction and discharge saturation temperatures. In combination with the standard conditions to produce the map via measured data from a compressor calorimeter (e.g., suction and injection superheat degrees such as 10R), standard subcooling degree of 10R, and temperature approach of the economizer (e.g., 10R), the isentropic and volumetric efficiencies can be reduced from the three sets of map coefficients.

Microchannel heat exchangers can be modeled using a segment-to-segment approach, by dividing each single port into mini-segments (Figure 4). Heat transfer, pressure drop, and charge inventory are calculated segment by segment. Air flows by passing port-to-port along the microchannel tube and fin surface.

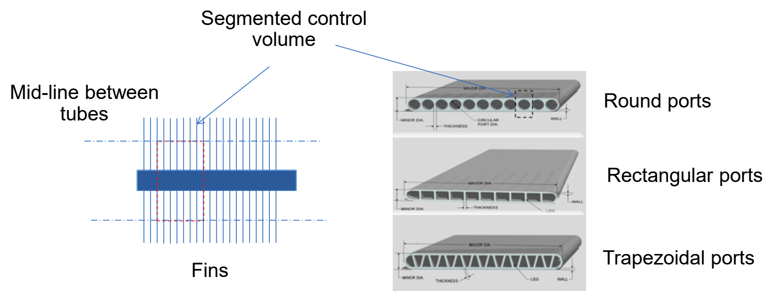

Figure 4. Segmented control volume of a mini-port.

There are many port shapes, such as round, rectangular, trapezoidal, and triangular. The port hydraulic inside diameter is treated as the characteristic dimension to calculate heat transfer, pressure drop, and void fraction. The real shape of a port is used to calculate its cross-sectional flow area for determining the inner volume, charge inventory, and inside heat transfer surface area. Each port's outside heat transfer surface area is calculated using the microchannel tube total outside surface area divided by its number of ports. The fins are treated as straight fins.

The heat transfer rate $\dot{Q}$ in each mini-segment is calculated using the $\varepsilon - NTU$ method. The pressure drop of the refrigerant $\Delta P_{r,k}$ is determined by the frictional pressure drop and momentum pressure drop within the port segment. In addition, the refrigerant mass in the tube segment can be determined with the known refrigerant properties and inner volume. The essence of using finite segments is that the substance properties can be treated as constant within each segment. Certainly, smaller segments lead to better accuracy. However, this is at the expense of calculation time.

For elements where no moisture condenses, the effectiveness correlations for sensible heat transfer are used to calculate the heat transfer between the refrigerant and air. The fin efficiency is calculated as Equation (1):

$$\eta_F = \frac{\tanh(mL_f)}{mL_f} \tag{1}$$

where η_F is the fin efficiency for heat transfer relative to the maximum heat transfer if the whole fin temperature is the same as the fin base temperature, and L_f is the equivalent fin length in each control volume. Annular fin for fin-and-tube and straight fin for microchannel heat exchangers can be assumed to have the same surface area. In addition, $m = \sqrt{\frac{2h_o}{k_f \delta_f}}$, k_f is the fin thermal conductivity, h_o is the airside heat transfer coefficient, and δ_f is the fin thickness.

The overall efficiency relative to the entire surface area is calculated as,

$$\eta_o A_o = (A_o - A_F) + \eta_F A_F , \tag{2}$$

where A_F is the fin surface area of each divided port, and A_o is the exposed port surface area, including the port outside surface and the fin surface.

The overall heat transfer conductance UA is composed of three parts: the air side, the tube wall, and the refrigerant side:

$$\frac{1}{UA} = R_o + R_{port} + R_i = \frac{1}{\eta_o A_o h_o} + R_{port} + \frac{1}{h_i A_i} , \tag{3}$$

where h_i is the refrigerant side heat transfer coefficient, A_i is the port inside heat transfer area, and R_{port} is the tube wall thermal resistance, assuming each port is a mini round tube.

The single-row microchannel heat exchanger in the laboratory system has 61 tubes, with 45 tubes in the condensing section and 16 tubes in the subcooler. The tube length is 78 in., tube width is 1 in., and tube pitch is 0.37 in. The fin density is 276 fins/ft, singe fin thickness is 0.00315 in., and each microchannel flat tube has 26 rectangular channels with an inner hydronic diameter of 0.0278 in.

We simulated the four-stage compression system with the following refrigerants: R-134a, R-1234yf, and R-513A. Copeland provided MT compressor maps for the three refrigerants. However, no maps are available for the LT compressors, so constant isentropic efficiency (60%) and volumetric efficiency (95%) are assumed for the two separate stages of the LT VI compressors. The condenser fan consumes 500 W to drive 6,700 cfm air flow. The suction saturation temperature of the MT compressors was set at 30°F, and the saturation temperature entering the LT compressors was set at −25°F.

Figures 5 and 6 present the LT and MT capacities of the three refrigerants. Figure 7 shows the total coefficients of performance (COPs), or the sum of LT capacity and MT capacity divided by the total power consumption (excluding the evaporator fan power). Refrigerants R-1234yf and R-513A lead to higher LT capacities but lower MT capacities. The resultant total COP of R-134a is higher than the COPs of R-1234yf and R-513A.

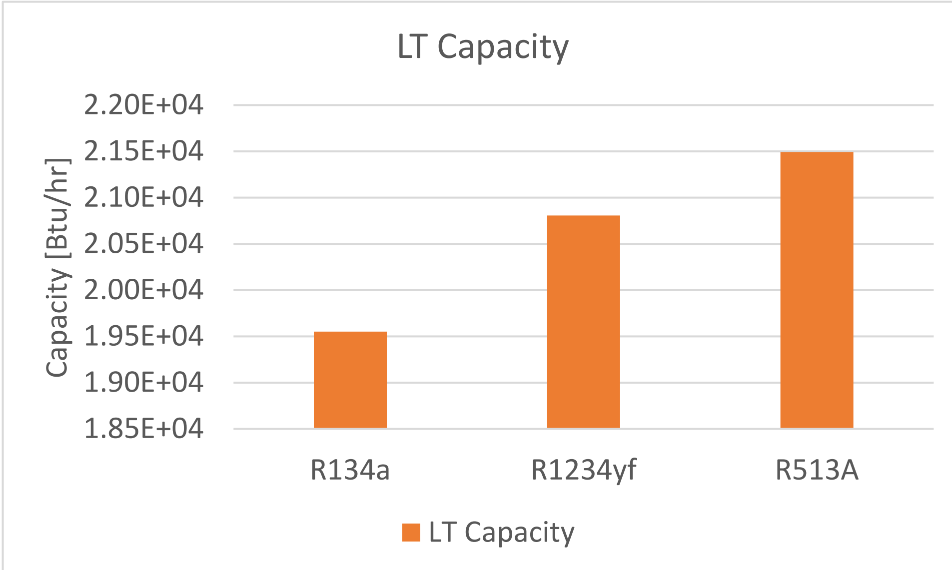

Figure 5. Predicted low temperature (LT) capacity of three refrigerants.

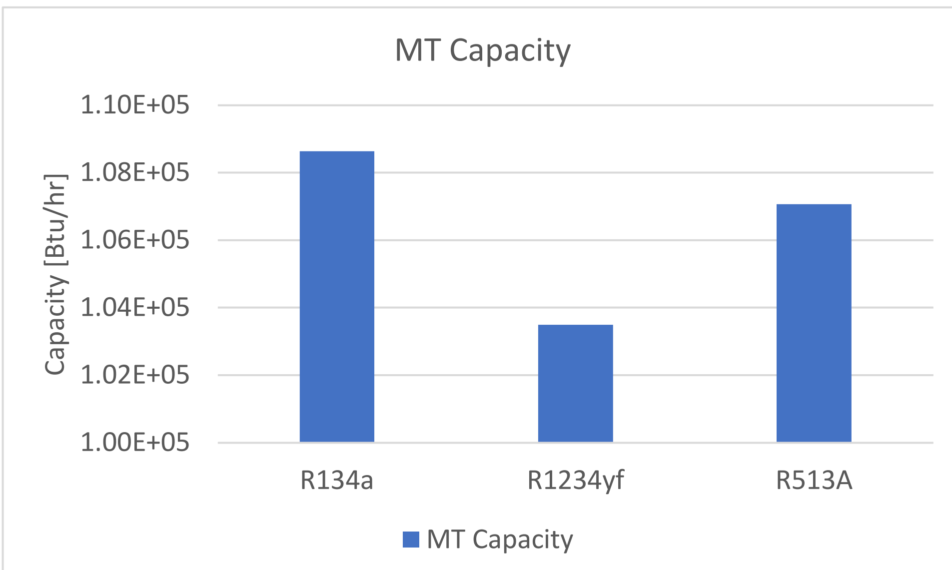

Figure 6. Predicted medium temperature (MT) capacity of three refrigerants.

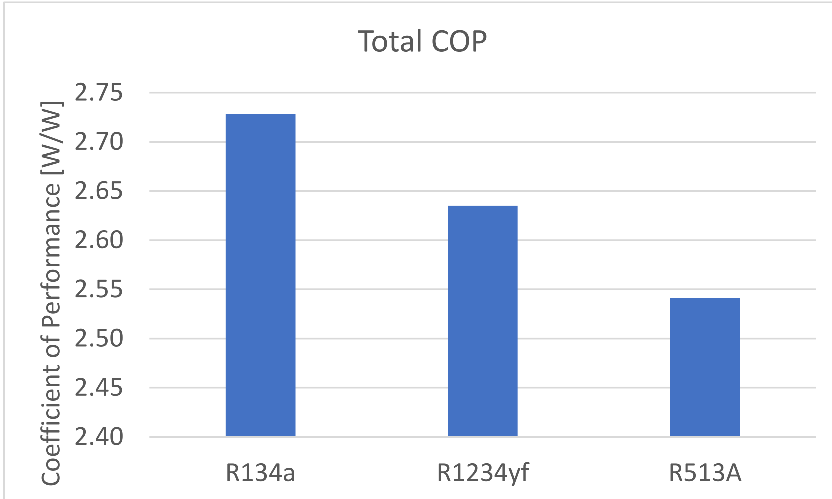

Figure 7. Predicted total COPs of three refrigerants. The COP for each refrigerant is the sum of LT capacity and MT capacity divided by the total power consumption (excluding the evaporator fan power) for that refrigerant.

3. LIFE CYCLE CLIMATE PERFORMANCE ANALYSIS

A life cycle climate performance (LCCP) analysis using the bin methodology was conducted to assess the environmental impact of a 35,000 ft^2 supermarket in Atlanta, Georgia. The study made specific assumptions regarding the refrigeration systems' leak rates, positing a 25% leak rate for the central DX system and a leak rate of less than 10% for the scroll distributed system. Furthermore, the analysis considered the efficiencies of existing R-134a compressors to accurately estimate GHG emissions. The findings revealed a significant environmental benefit, showing that overall GHG emissions could be reduced by 53% when compared with GHG emissions from the traditional central DX system (Figure 8). This reduction highlights the potential for more sustainable refrigeration practices in the supermarket industry, emphasizing the importance of selecting efficient systems with lower leak rates.

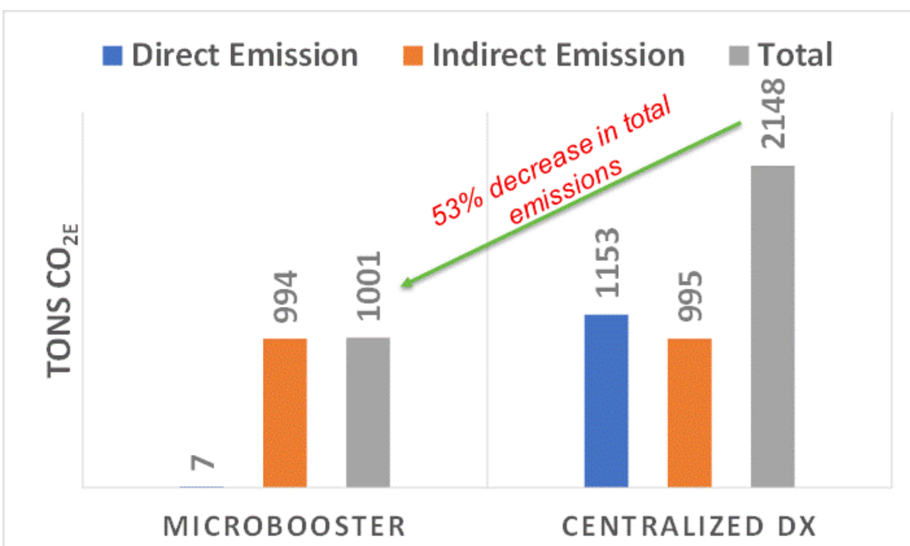

Figure 8. LCCP analysis for a central refrigeration system and a scroll booster system with the same cooling capacity.

4. PERFORMANCE TESTING

4.1 LAB TESTING SETUP AND TESTING PROCEDURE

4.1.1 Equipment for the Scroll-Booster System

The evaluation is conducted in an environmental chamber at ORNL. Two refrigerants are being studied in the test: R-134a and R-1234yf. The experimental setup is composed of an LT display case augmented with a false load to reflect a thermal capacity of 3 tons, approximately 28.5% of the thermal load. In parallel, an MT display case is similarly equipped with a false load, simulating a thermal capacity of 7.5 tons, or 71.5% of the total thermal load. A general description of these display cases are shown in Figure 9 (LT) and Figure 10 (MT).

Figure 9. Low temperature display case specifications and description.

Figure 10. Medium temperature display case specifications and description.

These two test cases are operated by a refrigeration rack system powered by Copeland Scroll compressors, which are highly regarded for their efficiency and operational stability in variable refrigeration scenarios. For both the MT and LT loops, a variable frequency drive is equipped on one compressor. This configuration accommodates load variations and enhances the overall efficiency of the system. The condensing process within the system is facilitated by an air-cooled microchannel condenser, selected for its superior heat transfer and reduced spatial requirements. The compressor rack is shown in Figure 11.

Figure 11. Compressor rack.

4.1.2 Equipment for the Standard DX System (R-404A Benchmark)

The efficiency obtained for this system will undergo a comparative analysis against benchmark data from a system previously studied at ORNL, which used R-404A and R-448A as refrigerants (Figure 12). That standard DX system, which was tested with R-404A, will be used as benchmark; it is well described in Fricke et al. (2017).

Figure 12. R-404A refrigeration system previously studied by ORNL. (Source: Fricke et al., 2017.)

4.1.3 Instrumentation

To accurately monitor and analyze the refrigerant dynamics within the system, six Coriolis flow meters are installed within the refrigerant liquid lines. These precision instruments are integral to measuring the mass flow rate of the refrigerants with high accuracy.

Furthermore, the experimental design incorporates a variety of sensors to continuously record system parameters such as pressure and temperature at strategic points throughout the system, thereby enabling a thorough thermodynamic analysis and calculation of an energy balance for each heat exchanger component. Power transducers are also installed to independently measure the electrical power consumption of the compressors, condenser fans, and display cases, thus allowing for an in-depth examination of the system's energy consumption patterns.

The experimental methodology is designed to minimize uncertainty, with an estimated experimental uncertainty of ±5%. This level of precision ensures that laboratory testing yields reliable data, which are critical for the extrapolation of the system's performance under real-world conditions and for the advancement of refrigeration technology in terms of energy efficiency and environmental impact.

4.1.4 Operating Conditions

The system shown in Figure 8 will be tested using two refrigerants: R-134a and R-1234yf. The operating conditions will be as follows: outdoor (75°F, 90°F, and 105°F) and indoor (70°F and 60% relative humidity). The system will be allowed to cycle ON and OFF, and energy consumption will be integrated over a period of at least 6 h.

5. SUMMARY

The purpose of the cooperative research and development agreement established between ORNL and Copeland was to investigate the potential of an innovative scroll booster refrigeration system developed for supermarkets. This system aims to significantly reduce GHG emissions and lower energy consumption relative to conventional supermarket refrigeration systems.

The research conducted comparative system modeling to evaluate the scroll booster system's performance employing R-134a and R-1234yf refrigerants against a standard centralized system using R-404A. Subsequently, an LCCP analysis was conducted to contrast the direct and indirect CO_2 emissions of these configurations. A laboratory evaluation followed to verify the performance of the scroll booster system.

The findings from the modeling and subsequent laboratory testing indicate that the scroll booster system using R-134a and R-1234yf achieves an energy efficiency 5% superior to that of the R-404A centralized system. R-1234yf and R-134a demonstrated equivalent efficiency and cooling capacity to each other when used in the scroll booster system, whereas the use of R-1234yf demonstrated the notable advantage of reducing direct CO_2 emissions by more than 99%.

In summary, the scroll booster system has fulfilled the anticipated performance and CO_2 emission benchmarks, marking it as a progressive and viable technology for the supermarket refrigeration sector.

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