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Critical Analysis of Replacements for R410A in Heat Pump Applications

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ABSTRACT

Using low-GWP refrigerants can reduce the Green House Gas (GHG) emission of HVAC systems. Research has shown that using heat exchangers with small diameter tubes is a promising solution to meet the performance goals of heat pump using low-GWP refrigerants due to reduced refrigerant charge, reduced flammable impact and environmental impact. However, application of small diameter tube requires in-depth component design optimization to make the new system adapt to low-GWP refrigerants.

In this paper, multi-objective optimizations using Particle Swarm Optimization (PSO) algorithm on a R-410A residential 5-ton air source heat pump is performed for improved system performance and reduced material cost. Five R-410A alternatives, i.e., R-32, R-454A, R-454B, R-454C and R-455A are investigated. R-455A and R-454C have GWP lower than 150. As a result of optimization, 12.4%-19.1% Energy Efficiency Ratio (EER) improvement and up to 71% HXs material cost saving is achieved. Life Cycle Climate Performance (LCCP) analysis shows that optimized systems reduce total CO2 emission by 13%-33% depending on the choice of refrigerant and climate zone.

The optimal heat exchangers resulting from this research can fit into the original R-410A fan-coil units. The proposed heat pump design method establishes a production and installation path to produce cost-effective low-GWP heat pumps easily accepted by end users.

Key words: Low GWP, Small diameter tube, Heat exchanger, Optimization, LCCP

1. INTRODUCTION

Residential Air conditioning and water heating contributes substantially to the modern life in the US. Environmental concerns are the primarily driving force for refrigerant changes during the past 40 years. The phased-out schedule of Ozone-depleting refrigerants established by the Montreal Protocol (1987) affected chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC), which were primarily replaced by hydrofluorocarbons (HFCs).

The 2016 Kigali Amendment responds to the concerns about climate change by reducing the direct effect related to the use of refrigerants. Most of the new low-GWP alternatives are flammable and classified as 2L (lower flammability), 2 (flammable) and 3 (higher flammability) by refrigerants designation and classification standard ASHRAE 34. The emergent use of mildly flammable class 2L refrigerants has driven intense research on flammability. Investigations have been performed on fundamental flammability characteristics, full-scale experiments, and risk assessments. The results are being used in updating installation standards such as ASHRAE Standard 15 and equipment standards such as UL 60335-2-40. The combined non-ODP and low-GWP requirements further reduced the options for some of the mainstream applications such as RAC HPs and HPWHs. When exploring new refrigerants for systems with air-to-refrigerant heat exchangers, which are the most common in these applications, the industry preference is to identify a single-component fluid with a similar performance to the refrigerant to be replaced. If a single fluid is not available, an azeotropic or near-azeotropic blend with no temperature or small glide are also desirable options. Unfortunately, most of the alternatives are non-azeotropic blends with large temperature glide where optimizing heat exchangers can be challenging.

Over the last several years, multiple fluids have been proposed as low-GWP substitutes for heat pumps. The relative merits of alternative refrigerants depend on a combination of several factors discussed including performance, safety, and material compatibility. The importance of such factors fluctuates between countries due to economic development, regional regulations, and types of equipment. The fluids included in this study should comply with national regulations, regional regulations and the phase-down schedule mandated by the Kigali Amendment. For non-article 5 countries, this planning strategy corresponds to the reduction steps imposed by the Kigali Amendment, where medium-GWP refrigerants (GWP<750) will satisfy the 2024-2025 phase-down, and even lower-GWP refrigerants (estimated at GWP<150) will be required to comply with the additional reduction impending in 2029.

2. METHODOLOGY

2.1 System Model and Selection of Refrigerants

The DOE/ORNL Heat Pump Design Model (HPDM) is used to model the performance of heat pumps. HPDM is a public-domain HVAC equipment and system modelling and design tool, which supports a free web interface and a desktop version for public use. A finite volume (segment-to-segment) tube-fin HX model is used to simulate the performance of the heat exchanger with different circuitries. This model has been validated by the experiment data Abdelaziz *et al.* (2016). The dehumidification model used in the evaporator simulation is Braun *et al.* (1989). More details of HPDM can be found in Shen *et al.* (2018). In HPDM, REFPROP 10.0 Lemmon *et al.* (2010) is used to simulate the refrigerant properties.

Regarding refrigerants, there is continuous introduction of new low-GWP fluids. Near term options have GWP<750 and include R-32 and R-454B. Long term options would likely require GWP<150, which require the use of fluids that do not match the incumbent fluid (R-410A). Hence, component and system optimization are required. Among these candidates, R-454C and R-455A require significant changes of heat exchanger design to address their high saturation temperature glide that could result in lower efficiency.

Table 1 depicts the characteristics of R-410A and its low-GWP alternatives for a typical residential air source heat pump. R-32 and R-454B have acceptable performance but present the flammability challenge. R-454C and R-455A have GWP less than 150 are the long-term candidates. The temperature glides are evaluated at saturation pressure corresponding to 8 °C dew-point temperature.

Table 1: Characteristics of Refrigerants Investigated in This Research

Refrigerant	GWP	Safety Class	Composition and Mass Fraction	Glide [C]	Critical Temperature [C]
R-410A	2088	A1	R-32/R-125: 50%/50%	0.1	72.8
R-32	675	A2L	R-32: 100%	0	78.1
R-454B	466	A2L	R-32/R-1234yf: 68.9% /31.3%	1.3	77
R-454A	238	A2L	R-32/R-1234yf: 35%/65%	6.2	78.9
R-454C	146	A2L	R-32/R-1234yf: 21.5% /78.5%	6.0	82.4
R-455A	139	A2L	R-32/R-1234yf/CO_2: 21.5%/75.5%/3%	6.9	90.2

To improve the model prediction of 5 mm tube heat exchangers, a set of small diameter air side heat transfer coefficient and pressure drop correlation, i.e., Sarpotdar *et al.* (2016), is implemented in HPDM. Sarpotdar *et al.* (2016) correlation is developed for 3-5 mm diameter tube slit fin heat exchangers using CFD. It is worthwhile to mention that, to predict the airside performance of 9 mm tube heat exchangers in the baseline R410A heat pump, Wang *et al.* (1999) correlation is adopted.

2.2 Baseline Reversible Heat Pump System

To compare the refrigerants in an existing reversible heat pump system, a commercial 5-ton R-410A residential two-speed heat pump is modelled. Figure 1 shows the schematic of the baseline heat pump operating under cooling mode and heating mode. The refrigerant direction inside the heat exchangers is reversed after mode switching.

(a) (b)

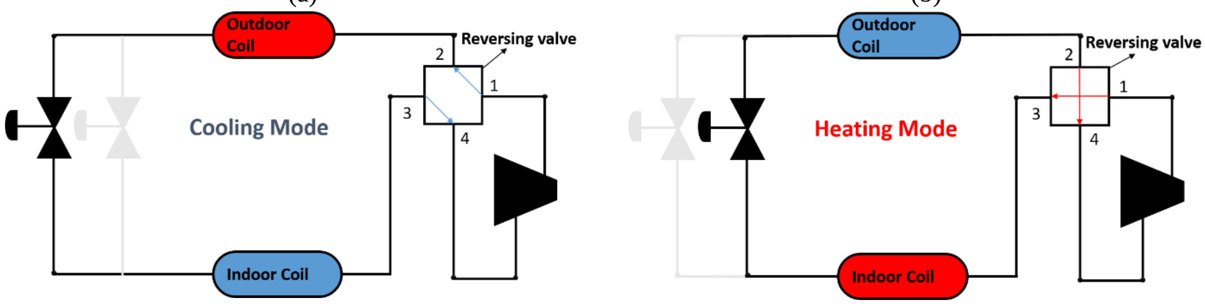


Figure 1: 5-ton R410A Baseline Heat Pump System: (a) Cooling Mode Operation; (b) Heating Mode Operation.

Table 2 lists the structural parameters of the baseline heat exchangers as well as the air volume flow rate and fan power for the indoor and outdoor fans.

Table 2: Parameters of Indoor and Outdoor Units of Baseline 5-ton Two-stage Heat Pump

Parameters (heating mode)	**Indoor HX**	**Outdoor HX**
Face area, ft^2	3.6	33.7
Total Tube Number	72	96
Number of Rows	3 (cross mixed flow)	2 (cross mixed flow)
Number of Circuits	8	8
Fin Type	Slit	Slit
Fin Density, fins/ft	168	276
Tube Outside Diameter [mm]	9.52	9.52
Tube Horizontal Spacing [mm]	25.4	22.0
Tube Vertical Spacing [mm]	25.4	25.4
	Indoor Blower	**Outdoor Fan**
Flow Rate [CFM]	1770	4215
Power [W]	478	181

The circuitry of the baseline R410A indoor and outdoor heat exchangers (HXs) are shown in Figure 2 (a) and Figure 2 (b), respectively. The indoor HX has 72 tubes and 3 tube rows and is divided into 8 mixed flow circuits. The outdoor HX has 96 tubes and 2 tube rows and is also divided into 8 mixed flow circuits. Different colors represent different circuits.

(a) (b)

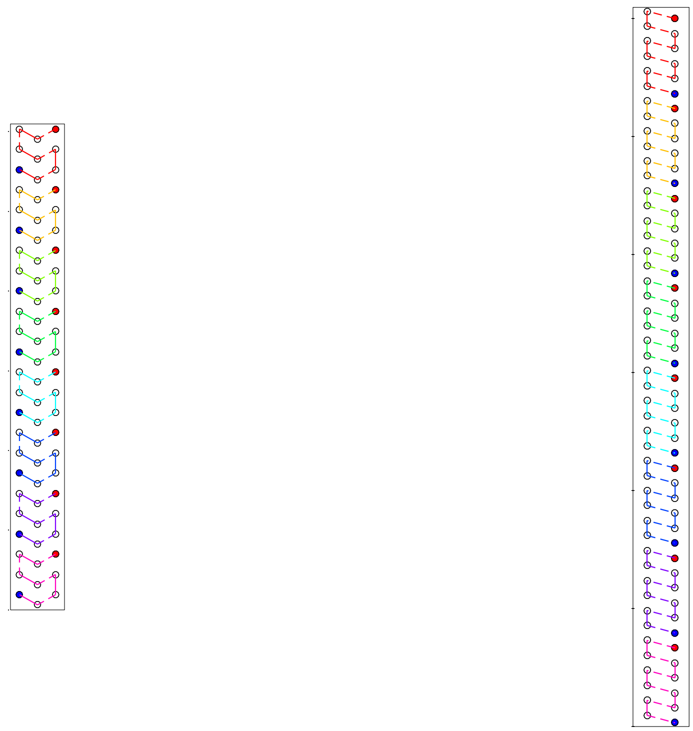


Figure 2: Baseline Tube-fin Heat Exchanger Circuitries: (a) Indoor HX; (b) Outdoor HX.

2.3 Optimization Problem Formulation

Shen *et al.* (2012) developed an optimization framework that integrates HPDM with GenOpt (Wetter (2001)), a public domain optimization package. In this research, the Particle Swarm Optimization (PSO) algorithm implemented in GenOpt is used to optimize the heat pumps. Regarding PSO setting, the optimization runs use 100 as population size and 200 as number of generations.

Equation (1) shows the bi-objective optimization problem formulation. The 1st objective in this optimization study is to maximize the Energy Efficient Ratio (EER) of the heat pump under AHRI Standard 210/240 AHRI (2008) cooling test A condition (95 °F). The 2^{nd} objective is to minimize the heat exchangers material cost. The heat exchangers include the indoor heat exchanger and outdoor heat exchanger. In Equation (1), the number of circuits in indoor and outdoor heat exchangers are two design variables, which varies between 1 and the total number of tubes in each bank of the indoor and outdoor heat exchangers. Table 3 shows the design space. As can be seen, the number of tubes in each bank of the heat exchangers is also a design variable. It means that the number of circuits has a self-adaptive upper limit, instead of a fixed upper limit.

In terms of constraints on operating conditions, the evaporator outlet superheat degree is specified based on the temperature glide of different refrigerants as recommended by refrigerant OEM. The condenser outlet subcooling degree is automatically adjusted, but it is constrained between 2 R to 15 R. the cooling capacity of evaporator is fixed to be the same as that of the original 5-ton R410A heat pump. The compressor displacement volume is automatically altered in HPDM to meet the target evaporator cooling capacity.

The last four constraints in Equation (1) guarantee that the optimal indoor and outdoor heat exchangers have the same frontal shapes as the baseline heat exchangers, i.e., the optimal heat exchangers can fit into the original indoor and outdoor fan-coil unit perfectly. Using those geometry constraints, we want to ease the retrofit effort of upgrade the old R410A heat pump to the new low-GWP system by minimizing the change in manufacturing and installation processes and guarantee that the optimal systems have the best compatibility with end-users' house structure. As a result, the new products can be easily accepted by manufacturers and end-users.

$$
\begin{aligned}
&\textit{Maximize}: \ EER \\
&\textit{Minimize}: \ HXs\ Cost \\
&\textit{Subject to}: \\
&\quad Heat\ exchanger\ tube\ diameter = 5\ mm \\
&\quad 1 \le N_{circuits,evaporator} \le Ntubes\ per\ bank\ of\ evaporator \\
&\quad 1 \le N_{circuits,condenser} \le Ntubes\ per\ bank\ of\ condenser \\
&\quad \Delta T_{\text{superheat, evaporator outlet}} = 10 - \frac{\Delta T_{glide}}{2}\ [\mathrm{R}] \\
&\quad 2\ [\mathrm{R}] \le \Delta T_{\text{subcooling, condenser outlet}} \le 15\ [R] \\
&\quad \mathrm{Q}_{evaporator} = 16.1\ \mathrm{kW} \\
&\quad |\ \mathrm{SHR}_{evaporator} - \mathrm{SHR}_{baseline,evaporator}\ | \le 1\% \\
&\quad \mathrm{Height}_{evaporator} = \mathrm{Height}_{baseline} \\
&\quad \mathrm{Length}_{evaporator} = \mathrm{Length}_{baseline} \\
&\quad \mathrm{Height}_{condenser} = \mathrm{Height}_{baseline} \\
&\quad \mathrm{Length}_{condenser} = \mathrm{Length}_{baseline}
\end{aligned}
\tag{1}
$$

Table 3: Design Space of Heat Exchanger Optimization

HX	Design Variable	Unit	Baseline	Range	Variable Type
Outdoor HX	Vertical Spacing Ratio (Pt/OD)	-	2.67	1.5-3	Continuous
	Number of Tube Banks	-	2	2-6	Discrete
	Number of Tubes Per Bank	-	48	48-144	Discrete
	Number of Circuits	-	8	1 - NTubes Per Bank	Discrete
Indoor HX	Vertical Spacing Ratio (Pt/OD)	-	2.67	1.5-3	Continuous
	Number of Tube Banks	-	3	3-9	Discrete
	Number of Tubes Per Bank	-	24	24-72	Discrete
	Number of Circuits	-	8	1-NTubes Per Bank	Discrete

For all the optimization runs, the heat exchanger circuitry pattern is fixed as counter flow configuration, as opposed to the crossflow circuitry pattern in the baseline system. This is because the counter flow configuration has the most efficient heat transfer which shows significant advantage for high-glide zeotropic mixtures. The HX material cost is calculated from Equation (2), where *MP* is the material price, ρ is the material density and *V* is the material volume. We assume the tube material is copper and the fin material is aluminum. The copper price is assumed as 4 times of the aluminum price per unit mass. These assumptions are made by referring the raw material price per unit mass on market during the execution period of this study.

$$C = (MP*\rho*V)_{tube} + (MP*\rho*V)_{fin} \tag{2}$$

3. RESULTS

3.1 Optimization Results

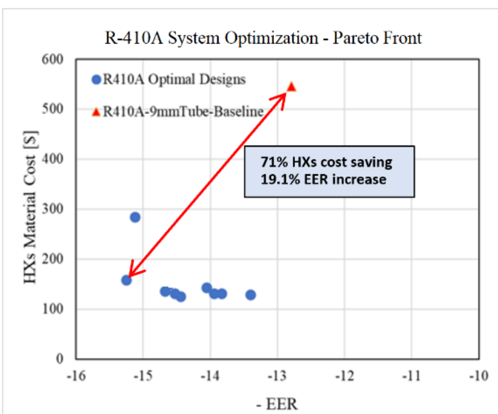
Figure 3 shows the results for R410A and all alternative refrigerants. The horizontal axis depicts the Energy Efficiency Ratio (EER) while the vertical one shows the material cost of the heat exchangers. The optimal R410A 5 mm tube design shows good performance, as expected. This performance point is also plotted on other Pareto fronts as a solid black hexagon.

R32 results (Figure 3b) show good performance, as expected because of its good thermal properties. On the other plots (Figure 3c-3f), reference points were set. The red triangle represents the baseline R410A system using 9 mm tube HXs. The yellow hollow circle represents a drop-in simulation using the baseline system. The green diamond symbol represents a drop-in simulation replacing the 9 mm tubes with 5 mm tubes. The purple rectangle symbol represents a design in which the 9 mm tubes are replaced with 5 mm tubes with the number of tubes being doubled.

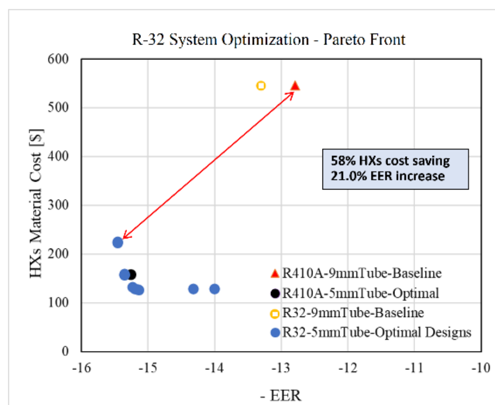
The drop-in comparison (red triangle with yellow hollow circle) shows decreased efficiency for all alternative refrigerants. Replacing the 9 mm tubes with the 5 mm tubes (yellow circle with green diamond) shows an even greater decrease, as expected by the reduced heat transfer area. Furthermore, doubling the number of tubes (purple rectangle with green diamond) shows that increasing HX area without optimization also fails to deliver a satisfactory solution. This analysis demonstrates the excellent sensitivity of the heat pump design model and emphasizes the need to perform optimization of the 5 mm tube system for all low-GWP alternatives.

Finally, the optimized systems (blue circle with red triangle) show significant HX cost savings and efficiency improvement compared with the baseline R410A 9 mm tube system. The maximum efficiency improvements for low-GWP systems range from 11.7% to 14.1%, and the optimized HX design can save material costs by at least 62% depending on the choice of refrigerants.

(a)

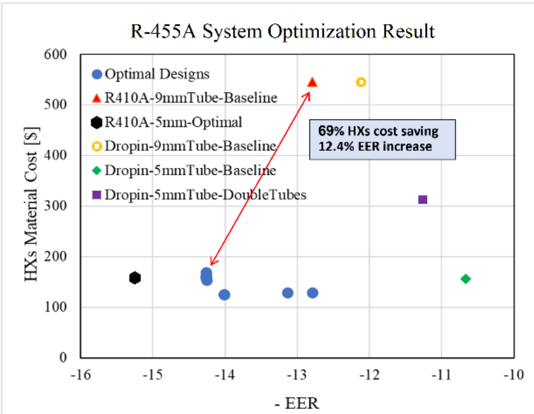


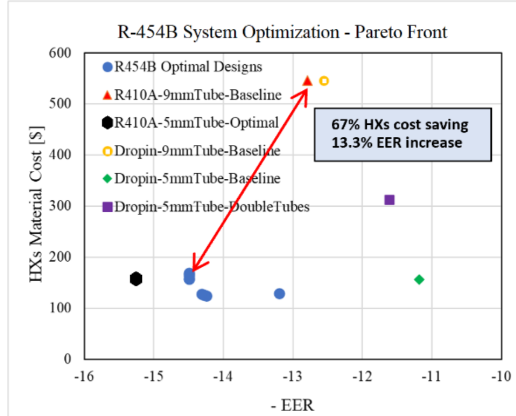
(b)



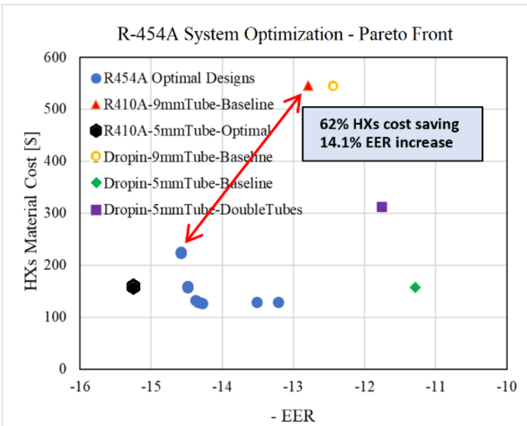
(c)

(d)





(e) (f)



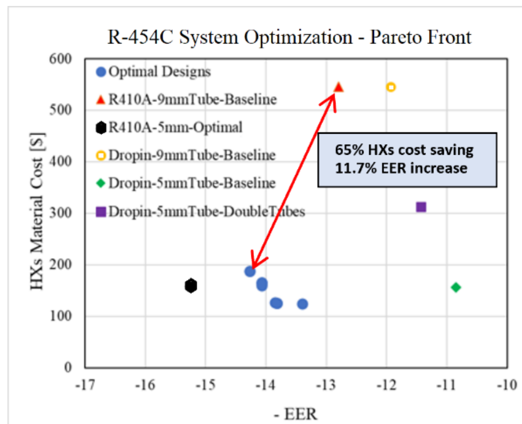
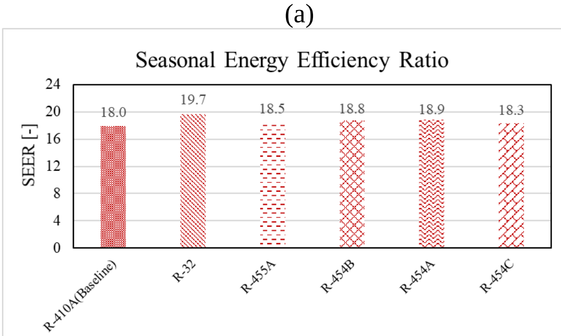


Figure 3: Pareto Fronts for 5 mm diameter tube heat pump system optimization using (a) R-410A, (b) R-32, (c) R-455A, (d) R-454B, (e) R-454A, and (f) R-454C.

3.2 Performance of Optimal Heat Pump Designs

The seasonal energy efficiency ratio (SEER) and heating seasonal performance factor (HSPF) were calculated for the optimized systems, according to AHRI 210/240 test standards AHRI (2008). In all cases, the volumetric displacement was adjusted to match the baseline cooling capacity. The performance degradation owing to frost accumulation was considered by applying performance degradation factors (0.91 for heating capacity and 0.985 for power consumption at the 35˚F dry bulb/33˚F wet bulb ambient condition). Figure 4 shows performance for the R-410A baseline system and low-GWP optimized systems with SEER over 16 and HSPF over 9.5.

(a) (b)



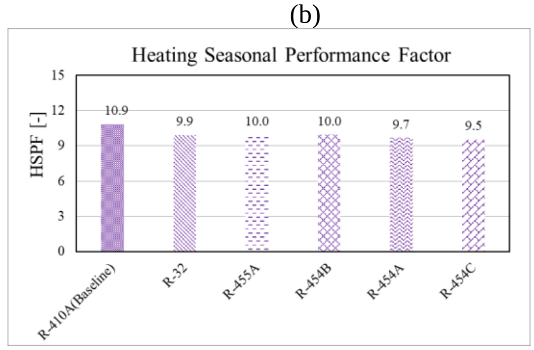


Figure 4: Performance of sampled optimized heat pump systems using different refrigerants: (a) SEER and (b) HSPF.

Figure 5(a) show the optimized systems charges with reductions ranging from 13% to 50%, likely because of the use of optimized 5 mm tube HXs. However, compressor displacements are larger than in the baseline, indicating the need for further development.

(a) (b)

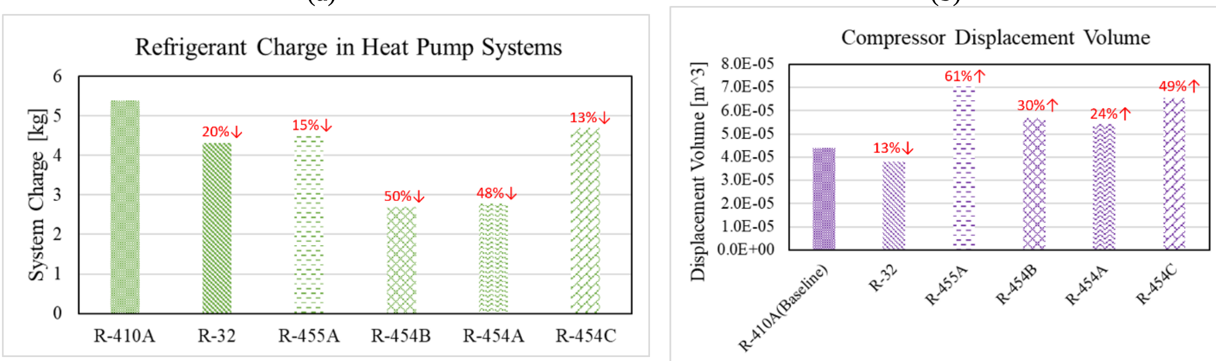


Figure 5: (a) System refrigerant charge and (b) designed compressor displacement volume.

3.3 Life cycle climate performance analysis

Life cycle climate performance (LCCP) evaluation Troch *et al.* (2016) was performed to analyze the direct and indirect greenhouse gas emissions of the system. To evaluate the annual energy consumption, each system was evaluated two cooling conditions and three heating conditions according to AHRI 210/240 test standards AHRI (2008). Other values used for evaluating the LCCP are shown in Table 4. The cut-off outdoor temperature and the temperature at which the heat pump starts are also shown.

Table 4: Input values for baseline system LCCP calculation

Factor	Value
Refrigerant	R-410A or its alternatives
Refrigerant charge (kg)	From Figure 3 (a)
Unit weight (kg)	190
Annual refrigerant leakage (%)	4
EOL leakage (%)	15
Lifetime (years)	15
Cut-off temperature (°C)	−17.8
Temperature at which the heat pump starts (°C)	−12.2

Figure 6 shows LCCP results for five cities representing all climate zones in the United States. Relative to the R-410A benchmark system, the optimized systems using low-GWP refrigerants reduce total lifetime greenhouse gas emissions by 13% to 33% depending on the specific climate zones.

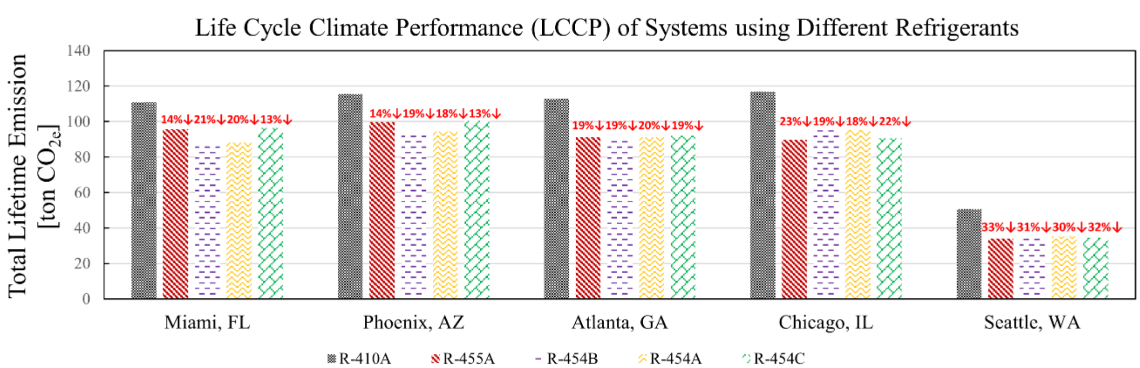


Figure 6: Total greenhouse gas emissions of the baseline system and low-GWP optimized systems.

4. CONCLUSION

This study presents heat exchanger and system development technologies to support the transition to refrigerants with GWP lower than 150. High efficiency levels in cooling (SEER over 16.0) and heating modes (HSPF over 9.5) were achieved by a model-based design optimization approach for low-GWP refrigerants using 5 mm tube heat exchangers. The potential to reduce the overall lifetime emissions of CO2 by 13% to 33% was also shown.

The optimal 5 mm tube heat exchangers obtained from this research can fit into the original R-410A system frame, which helps to minimize changes in manufacturing and installation, thus reducing impacts on manufacturers and end users. The proposed approach establishes a production and installation path to produce cost-effective low-GWP reversible heat pumps.

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