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Model-based Design Optimization to Achieve the Performance Goals (16.0 SEER/9.5 HSPF) – Next Generation Low Cost Direct-expansion Heat Pumps Using Refrigerant Mixtures with GWP <150, FY21 2nd Quarter Milestone Report

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03/31/2021

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FY21 2nd Quarter Milestone Report

Model-based Design Optimization to Achieve the Performance Goals (16.0 SEER/9.5 HSPF), Developing Heat Pump by Using Low-GWP Refrigerants and 5 mm Diameter Tubes

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Model-based Design Optimization to Achieve the Performance Goals (16.0 SEER/9.5 HSPF) (Regular Milestone)

Executive Summary

Usage of low-GWP refrigerants can reduce the Green House Gas (GHG) emission of HVAC systems. Our research in previous milestone report has shown that using heat exchangers with 5 mm diameter tubes instead of 9 mm diameter tubes is a promising solution to meet the performance goals of heat pump using low-GWP refrigerants. In addition, the 5mm tube heat exchangers can lead to lower system refrigerant charge and as a result, reduce environmental impact further. However, shifting to small tube diameters requires in-depth heat exchanger design optimization to adapt to the transition to low-GWP refrigerants.

In the 2nd quarter of FY21, we conducted multi-objective optimizations using Particle Swarm Optimization algorithm on a residential 5-ton air source heat pump to investigate the potential system performance improvements and material savings. 4 low-GWP refrigerants, ARM20A, ARM20B, R454A and R454C are investigated in this study. The objectives of the optimization are to minimize the heat pump material cost and to maximize the system performance simultaneously.

As a result, the HXs material cost is reduced by up to 77% according to the copper and aluminum material price in current market. Under heating mode operation, the smart 4-way valve guarantees that the optimal low-GWP systems maintain or outperform the heating performance of the R410A baseline system.

The model-based design optimization yields 18.3-18.9 SEER and 10.6-11.9 HSPF for optimal systems using different low-GWP refrigerants. The initial performance goals (16.0 SEER/9.5 HSPF) are achieved.

Furthermore, up to 50% system refrigerant charge reduction is possible in the optimized low-GWP heat pump system using ARM20B. And 91%-95% predicted life-time direct CO_2 emission reduction is achieved by using the optimal 5mm tube low-GWP heat pumps.

The significant material saving, charge reduction and direct CO_2 emission reduction help in reducing the environmental impacts of heat pump systems. The optimal heat exchangers resulting from this research can fit into the original baseline indoor and outdoor fan-coil unit. This can reduce the retrofitting effort by minimizing the change in manufacturing and installation of the heat pumps and guarantee the compatibility with end-users' house structure. Finally, the new products can be easily accepted by manufacturers and end-users.

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1. Background

Usage of low-GWP refrigerants can reduce the Green House Gas (GHG) emission of HVAC systems, in the 2nd quarter of this research, our goal aims at improving, or at least maintain, the system performance to be the same as the conventional refrigerants. On the other hand, cost-effective solutions of HVAC components are always favorable for the end-users. Our research in previous milestone report has shown that heat exchangers play significant role in refrigeration and air conditioning systems. Using heat exchangers with small tube diameters (less than 5 mm) instead of large tube diameters has been shown to be a promising solution to meet the performance goals of heat pump using low-GWP refrigerants. In addition, the 5mm tube heat exchangers can lead to lower system refrigerant charge and as a result, reduce environmental impact further. However, shifting to small tube diameters requires in-depth heat exchanger design optimization to adapt to the change of refrigerants.

One characteristic of these low-GWP alternative refrigerants are their high slides, i.e., temperature rise from the bubble point to the dew point at one pressure. High slide refrigerants prefer multi-row, counter-flow heat exchanger configuration for a single mode operation. If switching mode, the counter-flow heat exchanger (HX) becomes parallel-flow heat exchanger and induces heat exchanger performance degradation. In this project, a smart four-way valve is developed by our CRADA partner - Emerson Commercial and Residential Solutions (the Helix center). This smart four-way valve can maintain the same refrigerant flow direction inside the heat exchangers between mode switching.

In the 2nd quarter of FY21, we conducted bi-objective optimizations on a 5-ton R410A residential air source heat pump system using Particle Swarm Optimization (PSO) algorithm with a particular focus on the use of 5 mm diameter tubes in the heat exchangers. The objectives are to minimize the heat exchangers' cost and maximize the system performance. The goal of this quarterly milestone is three folded:

1. Optimize multi-row 5-mm tube coils with low-GWP refrigerants and determine the potential material savings and cost reduction
2. Investigate the effect of the smart 4-way valve to maintain optimum heat exchanger configurations in cooling and heating modes
3. Develop multi-stage compressors for different low GWP refrigerants
4. Analyze the annual performance indices of optimal low-GWP heat pumps and access their performance against the initial performance goal

2. System Model

The DOE/ORNL Heat Pump Design Model (HPDM) (Shen and Rice, 2016) is used to model the performance of heat pumps. The HPDM is a public-domain HVAC equipment and system modeling and design tool which supports a free web interface and a desktop version for public use. Some features of the HPDM related to this study are introduced below.

Compressor model: To compare refrigerant performances, it was assumed that the compressor has the same volumetric efficiency (η_{vol}=95% in Equation (1)) and isentropic efficiency ($\eta_{isentropic}$=70% in Equation (2)).

$$m_r = Volume_{displacement} \times Speed_{rotation} \times Density_{suction} \times \eta_{vol} \tag{1}$$

$$Power = m_r \times (h_{\text{discharge,s}} - h_{\text{suction}}) / \eta_{isentropic} \tag{2}$$

where m_r is compressor mass flow rate; Power is compressor power; η_{vol} is compressor volumetric efficiency; $\eta_{isentropic}$ is compressor isentropic efficiency; $h_{suction}$ is compressor suction enthalpy; $h_{discharge,s}$ is the enthalpy obtained at the compressor discharge pressure and the suction entropy; and $Speed_{rotation}$ is the motor rotational speed.

Heat exchanger model: A finite volume (segment-to-segment) tube-fin HX model is used to simulate the performance of the HX with different circuitries. This model has been validated by the experiment data from Abdelaziz et al. (2016). The dehumidification model used in the evaporator simulation is from Braun et al. (1989). Details can be seen in Shen et al. (2018).

Expansion device: Isenthalpic process is assumed in the expansion process.

Fans: The fan power is calculated by fan curve using heat exchanger air pressure as inputs. Since the system is a 5-ton system, the air volume flow rate (VFR) of indoor fan is fixed at 1770 CFM, and the air VFR of the outdoor blower is fixed at 4215 CFM.

Refrigerant Lines: Temperature changes and pressure drops in suction, discharge, and liquid lines are specified using the measured data from the experiments.

Refrigerant Properties: As a progressive workflow from previous project milestone, the simulated systems use four low-GWP refrigerants in addition to R410A. In HPDM, REFPROP 10.0 (Lemmon et al., 2010) is used to simulate the refrigerant properties. Table 1 lists the baseline R410A refrigerant and four low-GWP refrigerants (R454A, R454C, ARM20A, ARM20B) investigated in this study.

Table 1: Investigated Low-GWP Refrigerant and Baseline R410A

Refrigerant	GWP	Safety Class	Glide in Condenser [K]	Glide in Evaporator [K]	Critical Temperature [C]	Composition	Vendor
R410A (baseline)	2088	A1	0.1	0.1	72.8	R32 (0.5)/ R125 (0.5)	Honeywell
R454A	238	A2L	5.4	6.2	78.9	R32 (0.35)/ R1234yf (0.65)	Chemours
R454C	146	A2L	6.0	6.0	82.4	R32 (0.215)/ R1234yf (0.785)	Chemours
ARM20A	139	A2L	6.1	6.9	90.2	R32 (0.18)/ R1234yf (0.7)/ R152a (0.12)	Arkema
ARM20B	251	A2L	5.3	6.0	88.7	R32 (0.35)/ R1234yf (0.55)/ R152a (0.1)	Arkema

To improve the HPDM prediction of 5mm diameter tube heat exchangers, we implemented a set of small diameter tube airside heat transfer coefficient (HTC) and pressure drop (DP) correlations (Sarpotdar et al (2016)) in HPDM. Sarpotdar et al (2016) correlation is developed for 3-5 mm diameter tube slit fin heat exchangers using CFD and we use this correlation in the optimization runs. However, to predict the airside performance of 9 mm diameter tubes heat exchangers in the baseline R410A heat pump, we use the Wang et al. (1998) correlation, since Wang et al. (1998) correlation is developed for conventional large diameter tubes.

3. Optimization Problem Formulation

Equation (3) 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 cooling test A condition (95 °F). The 2nd objective is to minimize the heat exchangers material cost. The heat exchangers include the indoor heat exchanger and outdoor heat exchanger. In Equation (3), 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 2 shows the design space. From Table 2, 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 (3) 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.

$$
\begin{aligned}
&Maximize: \ EER \\
&Minimize: \ HXs\ Cost \\
&Subject\ to: \\
&\qquad Heat\ exchanger\ tube\ diameter = 5\ mm \\
&\qquad 1 \le N_{circuits,evaporator} \le Ntubes\ per\ bank\ of\ evaporator \\
&\qquad 1 \le N_{circuits,condenser} \le Ntubes\ per\ bank\ of\ condenser \\
&\qquad \Delta \mathrm{T}_{\text{superheat, evaporator outlet}} = 10 - \frac{\Delta \mathrm{T}_{glide}}{2}\ [\mathrm{R}] \\
&\qquad 2\ [\mathrm{R}] \le \Delta \mathrm{T}_{\text{subcooling, condenser outlet}} \le 15\ [R] \\
&\qquad \mathrm{Q}_{evaporator} = 16.1\ \mathrm{kW} \\
&\qquad |\ \mathrm{SHR}_{evaporator} - \mathrm{SHR}_{baseline,evaporator}\ | \le 1\% \\
&\qquad \mathrm{Height}_{evaporator} = \mathrm{Height}_{baseline} \\
&\qquad \mathrm{Length}_{evaporator} = \mathrm{Length}_{baseline} \\
&\qquad \mathrm{Height}_{condenser} = \mathrm{Height}_{baseline} \\
&\qquad \mathrm{Length}_{condenser} = \mathrm{Length}_{baseline}
\end{aligned}
\tag{3}
$$

Table 2: Design Space of 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 cross flow 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 used as the representative cost of the system, and the cost is calculated from Equation (4), where *MP* is the material price, ρ is the material density and *V* is the material volume. We assume the tube material is copper with \$8 per kilogram as its price, and the fin material is aluminum, with as \$2 per kilogram as its price. These assumptions are made by referring the raw material price on market during the execution period of this study.

$$C = (MP * \rho * V)_{tube} + (MP * \rho * V)_{fin} \tag{4}$$

4. Cooling Optimized Results

Figure 1 shows the Pareto Fronts for optimization of 5mm diameter tube heat pump system. Each subplot shows the Pareto Front for one refrigerant. In addition to the four optimization runs for the four low-GWP refrigerants (Figure 1(b)-(d)), we also performed an optimization run for an 5mm tube R410A system. Then we sampled one R410A design (highlighted in red circle in Figure 1(a)) which has the highest

EER and plot this R410A system performance on other Pareto Fronts shown as the solid black dot in Figure 1(b)-(d). As can be seen, the optimal R410A system using 5mm tube always have the best performance than the optimal designs using low-GWP refrigerants.

On the Pareto Front for low-GWP refrigerants (Figure 1(b)-(d)), there are several other reference design points. The red triangle, 'R410A-9mmTube-Baseline' shows the performance of the original R410A 5-ton system using 9mm tube heat exchangers. The yellow circle, 'Dropin-9mmTube-Baseline' shows the performance of the baseline system if the low-GWP refrigerant is directly used as drop-in refrigerant. The green diamond, 'Dropin-5mmTube-Baseline' is a design which only replaces 9mm tube with 5mm tube and use low-GWP refrigerant as drop-in. The purple rectangle, 'Dropin-5mmTube-Baseline' is a design which not only replaces 9mm tube with 5mm tube, but also double the number of tubes in both heat exchangers and use low-GWP refrigerant as drop-in.

From Figure 1(b)-(d), comparing the blue circles with red triangle, it can be seen that the optimal systems have significant heat exchanger material cost reduction and significant EER improvement compared with the original R410A 9mm tube system.

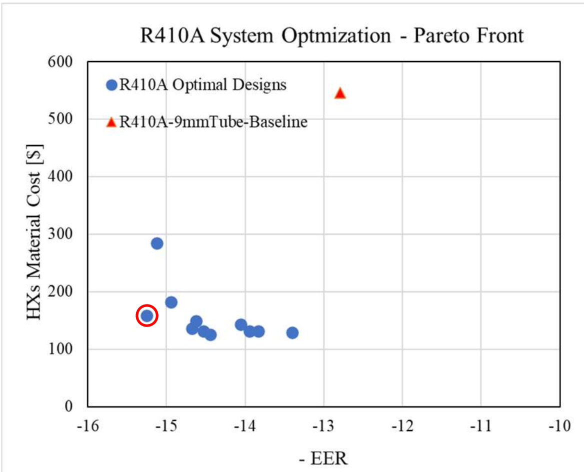
Comparing the red triangle with yellow circle, it can be seen that directly drop-in the low-GWP refrigerants into baseline system will induce EER decrease, while there is no change in HX material cost.

Comparing the yellow circle with green diamond, it can be concluded that, if 5mm tube is directly used to replace 9mm tube in a low-GWP system, the system performance will degrade dramatically due to significant reduction of primary heat transfer area.

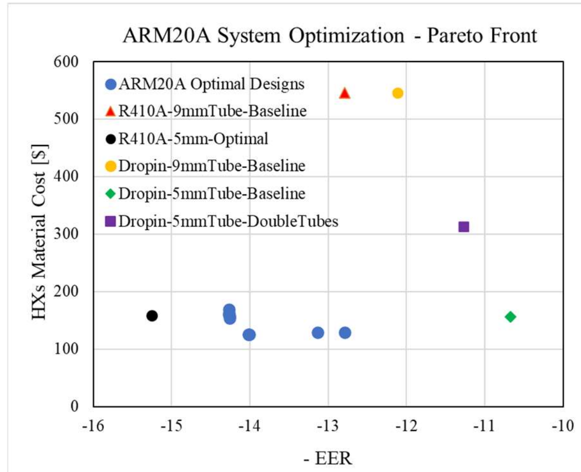
Comparing the purple rectangle with green diamond, it is obvious that, simply doubling the number of tubes of both heat exchangers without component optimization can maintain the system performance to be the same as the baseline, moreover, it makes the HX material cost unnecessarily expensive compared with the cost of optimal designs on the Pareto Fronts.

This analysis emphasizes the necessity to perform multi-objective optimization to improve the performance of 5 mm tube low-GWP system, and meanwhile achieve the cost-effective solutions.

(a)



(b)



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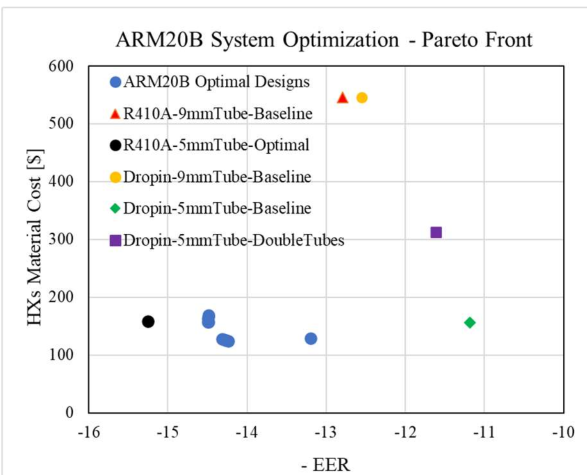
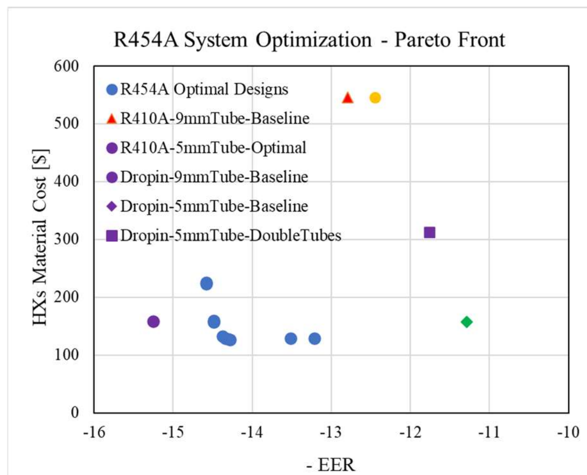


(c)



(d)

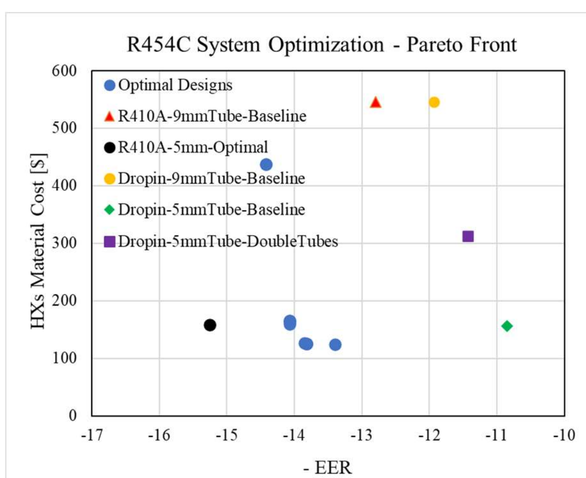


Figure 1: Pareto Fronts for 5 mm Diameter Tube Heat Pump System Optimization using (a) R410A; (b) ARM20A; (c) ARM20B; (c) R454A; (D) R454C

Table 3 shows the relative EER improvement and relative cost saving of sampled optimal designs from each Pareto Fronts in Figure 1 as compared with the R410A baseline system. 'Max-EER Design' is a design located on the left-most of the Pareto Front, and it has the largest EER among all designs. 'Least-Cost Design' is a design located on the right-most of the Pareto Front, and it has the least HX material cost among all designs. 'Medium Design' is a design sampled in the middle of the Pareto Front.

The maximum EER improvements of low-GWP system ranges from 11.5% to 14.0% and the maximum cost reduction ranges from 76.4% to 77.2% depending on choice of the refrigerants. For all the low-GWP refrigerants investigated in this study, there is always an existing optimal design (i.e., Medium Design) which can offer more than 10% EER improvement and 70% heat exchanger cost reduction simultaneously.

Table 3: Improvements of Sampled Optimal Designs for Each Refrigerant

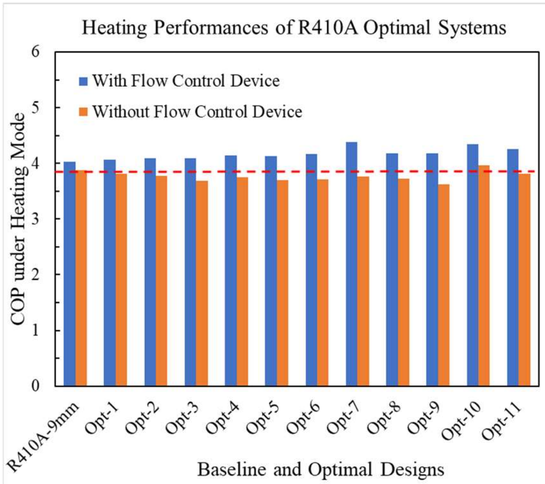
Refrigerant	Max-EER Design		Medium Design		Least-Cost Design	
	EER Improvement	Cost Saving	EER Improvement	Cost Saving	EER Improvement	Cost Saving
R410A	18.2%	47.9%	13.6%	76.0%	4.8%	76.4%
ARM20A	11.5%	70.1%	9.6%	77.0%	0.0%	76.4%
ARM20B	13.3%	69.0%	11.9%	76.7%	11.3%	77.2%
R454A	14.0%	59.0%	13.2%	71.1%	3.34%	76.4%
R454C	12.8%	19.9%	10.0%	70.7%	4.8%	77.2%

5. Heating Performance of Optimal Systems

The heating performance of the optimal systems in Figure 1 is evaluated under AHRI 210/240 47 °F test condition. For each optimal system, we performed two heating simulations, one assuming there is the smart 4-way valve to always maintain the counterflow heat exchanger circuitry pattern between mode switching, one simulation assuming it is a conventional reversible heat pump system without the smart 4-way valve. Figure 2 shows the heating performances of optimal designs for each refrigerant. The red dash line on each subplot represents the heating performance of the baseline R410A 9 mm tube system.

It is evident that only using the smart 4-way valve, the heating performance of the low-GWP optimal heat pumps can tie or outperform the baseline R410A system. Without the smart 4-way valve, the heating performance of the cooling optimized heat pumps are always below that of the baseline. This conclusion is even true for the R410A 5mm tube optimal systems as shown in Figure 2 (a).

(a)



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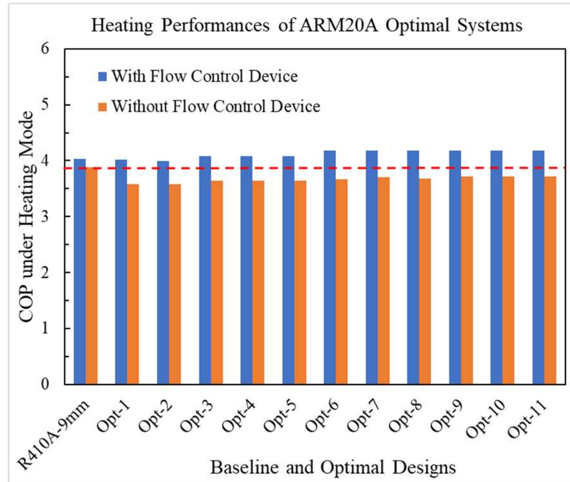

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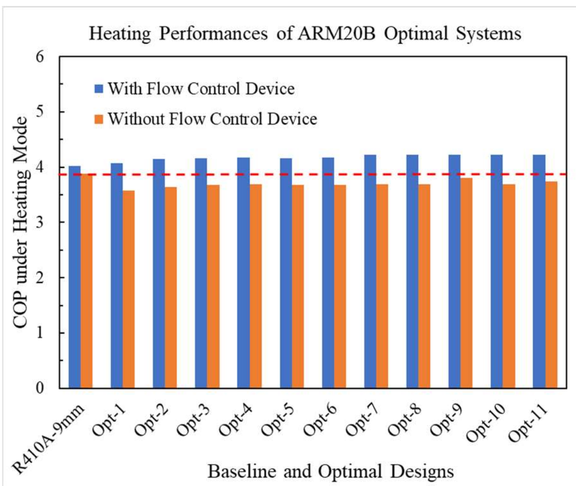

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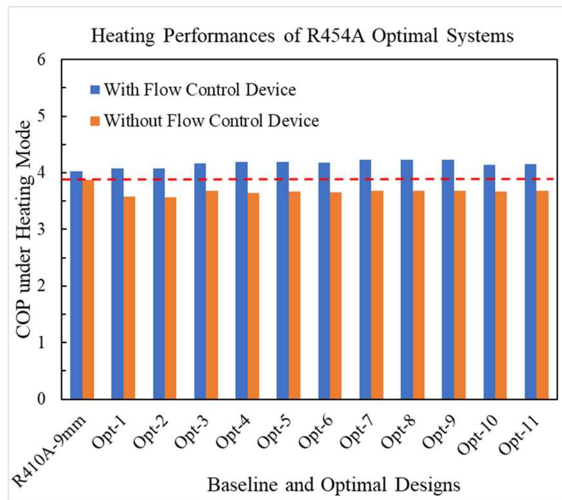

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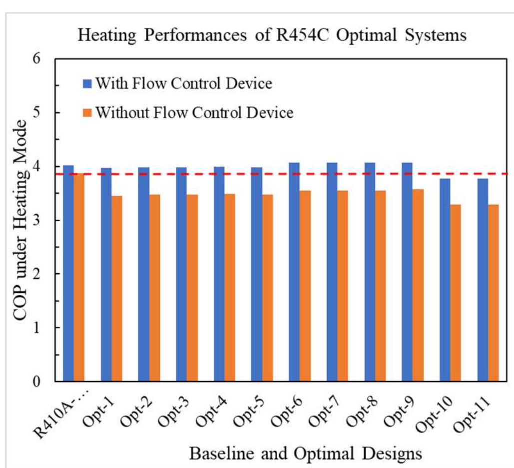


Figure 2: Heating Performances of Optimal Designs in Pareto Fronts (a) R410A; (b) ARM20A; (c) ARM20B; (c) R454A; (D) R454C

6. Seasonal Energy Efficiency Ratio (SEER) and Heating Seasonal Performance Factor (HSPF)

To assess whether the optimal systems can achieve the initial performance goal (16 SEER/9.5 HSPF), annual efficiency analysis is performed on the Medium Design in Figure 1 and Table 3. We use the Medium Design as the reprehensive refrigerant-specific optimal design.

The Seasonal Energy Efficiency Ratio (SEER) and Heating Seasonal Performance Factor (HSPF) analyses are conducted for the optimized systems using two-stage compressors. In the high-stage operation, the compressor displacement volume is solved from the HPDM simulation by matching the 5-ton cooling capacity. In the low-stage operation, the compressor displacement volume and indoor air flow rate is scaled to be 67% of those in high-stage. Under frosting condition, the performance degradation due to frost accumulation is considered in this analysis by applying performance degradation factors, to be specific, 0.91 is used as the degradation factor for heating capacity and 0.985 is used as the degradation factor for total power consumption. It is worthwhile to mention that the analysis assumes that there is no degradation of compressor efficiency when the compressor operates at low-speed.

Figure 3 shows the SEER and HSPF of the R410A baseline and low-GWP optimal systems. From Figure 3, SEER of low-GWP optimal designs is 18.3-18.9 and HSPF of low-GWP optimal designs with smart 4-way valve is 10.6-11.9. Both indices exceed the initial project goal, i.e., 16 SEER/9.5 HSPF.

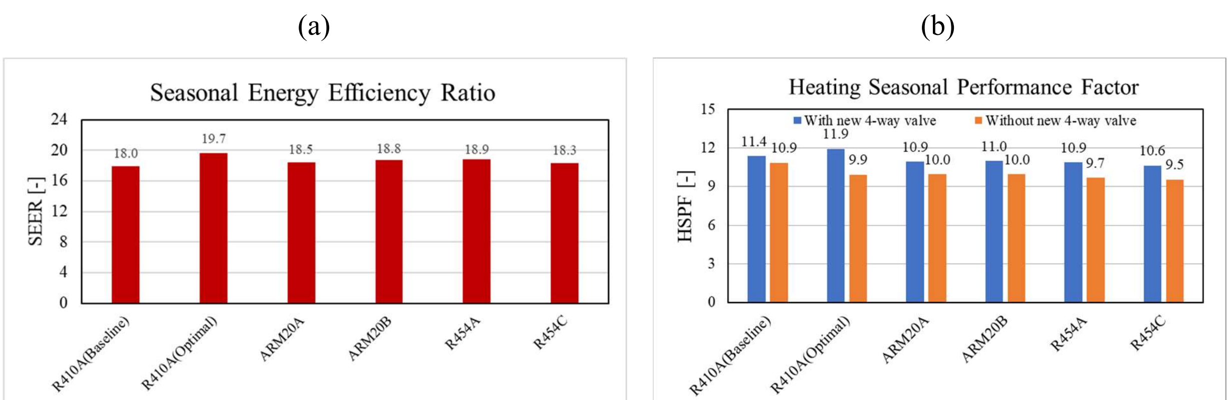
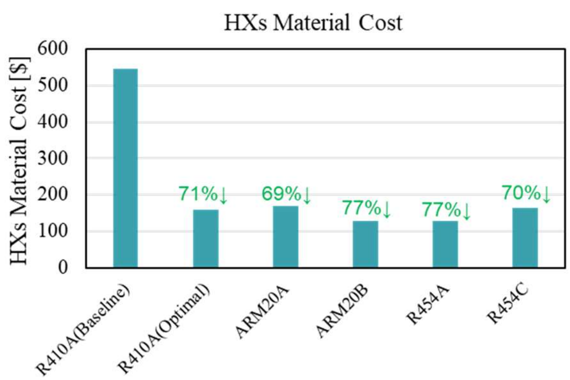


Figure 3: Performance of Sampled Optimal Heat Pump Systems using Different Refrigerants: (a) Seasonal Energy Efficiency Ratio; (b) Heating Seasonal Performance Factor.

7. Cost Saving, Compressor Displacement Volume, Charge Reduction and GHG Emission Reduction

Figure 4 (a) shows the heat exchangers material costs of the baseline and optimal low-GWP systems. As can be seen, heat exchangers material cost is reduced by 69% -77% compared with R410A baseline heat pump. Figure 4(b) shows the compressor displacement volume of the baseline and the optimal low-GWP systems, this information can help us develop the multi-stage compressor in next stage of this project.

(a) (b)



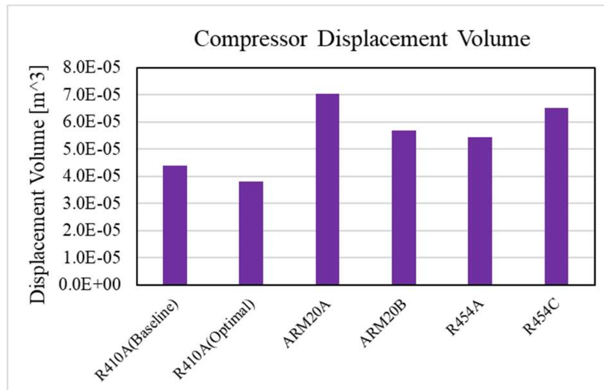
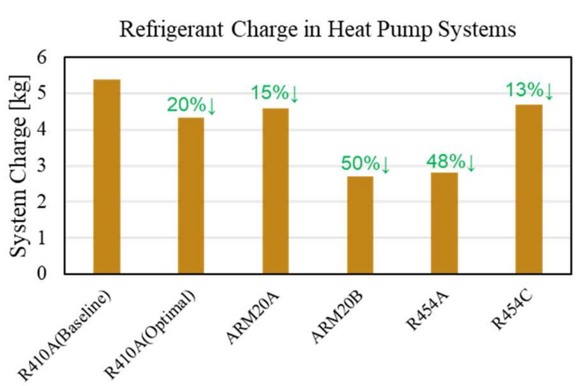


Figure 4: (a) Heat Exchanger Material Cost; (b) Designed Compressor Displacement Volume.

Figure 5 (a) shows the refrigerant charge in the R410A baseline system and optimal low-GWP systems. Depending on the choice of refrigerant, system refrigerate charge is reduced by 13%-50%. As the result, Figure 5 (b) shows direct life-time GHG Emission of different systems. When calculating the direct life-time GHG emission, 4% annual leakage rate, 15% end-of-life leakage rate and 15 years of system life-time are assumed. From Figure 5 (b), direct life-time CO_2 emission is reduced by 91%-95%. This significant direct emission reduction is attributes to the charge reduction as shown in Figure 5 (a) as well as the low-GWP value of these alternatives as summarized in Table 1.

(a) (b)



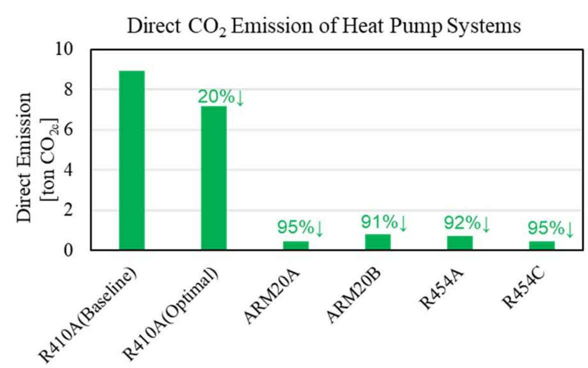


Figure 5: (a) System Refrigerant Charge; (b) Direct GHG Emission.

8. Summary

In this research, we presented a multi-objective optimization of a residential 5-ton air source heat pump system to determine the potential system performance improvements and material savings from using 5 mm diameter tubes in the HXs. The goal is to minimize the system cost and maximize the system EER while using 4 low-GWP refrigerants, ARM20A, ARM20B, R454A and R454C. The optimal performance of the low-GWP systems is achieved by redesigning the heat exchanger geometry and the compressor.

As a result, the HXs material cost can be reduced by 67% to 77% according to the copper and aluminum material price in current market. Under heating mode operation of the heat pump, the smart 4-way valve can guarantee the optimal low-GWP system maintain or outperform the heating performance of the R410A baseline system.

The model-based design optimization yields 18.3-18.9 SEER and 10.6-11.9 HSPF of low-GWP optimal designs with smart 4-way valve. The initial performance project goals (16.0 SEER/9.5 HSPF) are achieved.

Furthermore, up to 50% system refrigerant charge reduction is possible in the optimized low-GWP heat pump system using ARM20B. And 91%-95% life-time direct CO_2 emission reduction is obtained by using 5 mm tube low-GWP optimized heat pumps.

The significant material saving charge reduction and life-time direct CO_2 emission reduction help in reducing the environmental impacts of heat pump. The optimal heat exchangers resulting from this research can fit into the original indoor and outdoor fan-coil unit. This can reduce the retrofitting effort of the HVAC system by minimizing the change in manufacturing and installation processes and guarantee the compatibility with end-users' house structure. As a result, the optimized new products can be easily accepted by manufacturers and end-users.