Optimization of a Residential Air Source Heat Pump using Heat Exchangers with Small Diameter Tubes

Mohamed Beshr
University of Maryland, United States of America, mbeshr@umd.edu

Vikrant Aute
University of Maryland, United States of America, vikrant@umd.edu

Reinhard Radermacher
University of Maryland, United States of America, raderm@umd.edu

Optimization of a Residential Air Source Heat Pump using Heat Exchangers with Small Diameter Tubes

Mohamed BESHR[1], Vikrant AUTE[2*], Reinhard RADERMACHER[3]

[1,2,3]Department of Mechanical Engineering, University of Maryland,
College Park, MD 20742 USA
Phone: [1]301-405-7314, [2]301-405-8726, [3]301-405-5286
Email: [1]mbeshr@umd.edu, [2]vikrant@umd.edu, [3]raderm@umd.edu

* Corresponding Author

ABSTRACT

Heat exchangers play significant role in refrigeration and air conditioning systems. Ongoing research aims to improve, or at least maintain, the system performance while reducing the size, weight and cost of the heat exchangers. This in turn leads to lower system refrigerant charge and reduced environmental impact. 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 aforementioned goals. However, shifting towards small tube diameters requires in-depth analysis and optimization of several heat exchanger design parameters. This paper presents a multi-objective optimization of a residential air source heat pump system using genetic algorithms with a particular focus on the use of small diameter tubes in the heat exchangers. The objectives are to minimize the heat exchangers' cost and maximize the system performance. The goal of this study is to determine the potential material savings and cost reduction when using tube diameters between 3 mm and 5 mm in the heat exchangers. In addition to the tube diameters, multiple fin types, tube spacing and fin densities are also investigated. The optimization is carried out for R-410A, and a lower-GWP alternative, R-32. The system utilizing the improved heat exchanger designs has a cost reduction of 60% in comparison to the baseline system. Also, the improvements in the system's COP and the system charge reduction are around 20% and 35%, respectively.

1. INTRODUCTION

There exists a continuous need to increase the energy efficiency and reduce the negative environmental impacts of residential heating, ventilation, and air conditioning (HVAC) systems. The different methods to reduce the negative environmental effects of the HVAC systems include improving systems' efficiency, using low or zero global warming potential (GWP) refrigerants, and designing the system (components sizing, refrigerant, etc.) while accounting for its environmental impact. The first method requires enhancing the performance of the key components of the system, and matching them properly in the system. Also, the latter method includes optimizing the HVAC system to minimize the material mass, and refrigerant charge.

One of the components system that plays an important role in the design process of the HVAC system is the air-to-refrigerant heat exchanger (HX). Hence, there are many ongoing efforts to design and optimize the HXs in order to maximize their effectiveness while minimizing the system power consumption and refrigerant charge (Webb & Kim, 2005). In order to meet these goals, plate fin HXs with tube diameters smaller than 5 mm started to substitute the 7 mm and larger diameter tubes (Wu, et al., 2012; Pettersen, et al., 19986; Bacellar, et al., 2015; Ding, et al., 2011; Saji, et al., 2001; Kasagi, et al., 2003; Kasagi, et al., 2003; Paitoonsurikarn, et al., 2000; Abdelaziz, et al., 2010; Choi, et al., 2004)(Foli, et al., 2006; Shikazono, et al., 2007; Gholap & Khan, 2007; Sanaye & Hajabdollahi, 2010; Najafi, et al., 2011; Dang, et al., 2011). However, using smaller diameter tubes in a HX without changing the other design parameter leads to a decrease in the heat transfer area, and an increase in the refrigerant pressure drop. Hence, using tube diameters smaller than 5 mm in the HXs requires in-depth analysis and optimization of numerous HX design parameters. On the refrigerant side, the heat transfer and pressure correlations existing in literature are already validated and acceptable for small diameters. However, a small number of correlations exist for the prediction of the air side performance of small diameter HX designs (Bacellar, et al., 2014; Bacellar, et al., 2015). This lack of correlations in the 2-5 mm diameter range together with the research focus on the state-of-the-art finned micro-channel HX (MCHX) encouraged few studies (Wu, et al., 2012; Pettersen, et al., 19986; Bacellar, et al., 2015; Ding, et al., 2011; Bacellar, et al., 2014; Bacellar, et al., 2015)to study the 3-5 mm tube diameter HXs. However, there are no significant studies that investigate the design and optimization of small diameter tube-fin HXs and their effect on HVAC systems' performance and environmental impact. Thus, this paper focuses on the optimization of 3-5 mm tube plate fin HXs as compared to a baseline design of 9.5 mm tube HXs.

The Air Conditioning, Heating, and Refrigeration Institute (AHRI) started an industry wide collaborative effort known as the low GWP Alternative Refrigerants Evaluation Program (AREP) to gather industry resources in the hopes of finding and assessing promising alternative refrigerants (AHRI, 2016)(The Air Conditioning, Heating and Refrigeration Institute (AHRI), n.d.). The assessed refrigerants are either pure R-32, or blends comprised of R-32, R-1234yf, or R-1234ze in an effort to obtain a balance between low GWP, cost, safety, and system efficiency (Pham & Rajendran, 2012). Pure R-32 seems to be one among several promising candidates to replace R-410A in ASHP systems (Wang, et al., 2012). This is due to its comparable performance with the baseline R-410A (Wang, et al., 2012; Alabdulkarem, et al., 2013; Burns, et al., 2013; Lei & Yanting, 2013). Therefore, this paper presents the optimization results for the baseline R-410A and the alternative low GWP refrigerant R-32. The AREP reports focus on drop-in tests. That is, using the baseline R-410A system to determine the performance of R-32 as a replacement refrigerant. This means that the system is not optimized to provide the optimal performance when utilizing R-32. Nevertheless, some of these reports show soft optimization to the R-32 system (Li & By, 2012) performing a further analysis to determine the effect of making slight changes to the system, such as changing the system charge, compressor sizing, and using a suction line HX on the system performance. However, this soft optimization is limited because it focuses on a limited number of parameters, and is performed in a discrete simulation or retesting mode rather than continuous optimization mode. In order to explore more design points in a large design space with mixed-discrete design variables, we use a multi-objective genetic algorithm (MOGA) in the optimization of the HXs.

In conclusion, this paper presents the optimization of a residential ASHP system (Alabdulkarem, et al., 2013) with the objective of minimizing the HX cost (based on minimizing the required HX materials) and maximizing the system coefficient of performance (COP). The optimization is done for both R-410A and its alternative R-32. The geometry parameters of both the condenser and evaporator are used as design variables during optimization. The aim of this paper is to show the potential system performance improvement and cost reduction when using small diameter HXs.

2. SYSTEM SOLVER

In this study, we use a component-model based steady state system simulation tool (Richardson, et al., 2002; Winkler, et al., 2006; Winkler, et al., 2008) to simulate the ASHP system and optimize it. The simulation tool uses three main data structures (components, ports, and junctions) in simulating a vapor compression system. Components are defined as refrigeration system components, and are modeled as black box objects interacting with one another through a series of ports and junctions (Winkler, et al., 2006). This tool has built-in multi-objective optimization routines which make it able to optimize various performance or economic variables. This allows for the changing of any component or system level property. The cycle solving scheme used to solve the cycle is the enthalpy marching solver (Winkler, et al., 2008). In this solver, the input values are the suction pressure and enthalpy, and the discharge pressure. The solver then runs the components in a sequence (compressor, condenser, expansion device, and evaporator with pipes in between the components) where the outlet state of each port goes as the inlet state to the following component. A non-linear equation solver based on the Broyden's method (Broyden, 1965) is used in the current study to solve the residual equations derived from the mass and energy balance until convergence.

3. SYSTEM MODEL

The ASHP system simulated in this study is similar to the system tested experimentally in the AHRI low GWP AREP report by Alabdulkarem et al. (Alabdulkarem, et al., 2013). The schematic of the HXs is shown in Figure 1. The system simulation tool is validated against the experimental data in the report (Alabdulkarem, et al., 2015). The compressor model is a ten-coefficient (AHRI-540-2004 Standard) model. The coefficients are supplied by the manufacturer for R410A. However, we use a power correction factor of 1.12. Also, to adjust the compressor performance for R-32, we use mass flow rate and power correction factors of 1.22 and 0.69, respectively. The HXs component models use a finite volume HX simulation tool (Jiang, et al., 2006) for modeling various geometries and designs of HXs during each iteration in the optimization process. Table 1 shows the correlations used for the air and refrigerant sides in both HXs.

The expansion device's inlet subcooling and suction superheat are the convergence criteria and their values are set to be equal to the experimental values for the corresponding testing conditions and refrigerant. Moreover, the energy consumption of the system during the COP calculation is based on the compressor power and the actual fan power delivered to the air flowing through the HXs. In this study, we use a MOGA (Deb, 2001; Aute & Radermacher, 2014) for the system optimization. Also, we repeat the optimization for both R-410A and R-32, each for ASHRAE (ASHRAE, 1995) cooling tests A B, and C operating conditions. However, we focus on tests A and C operating conditions in this paper because these are the two extreme test points in the ASHRAE cooling test matrix.

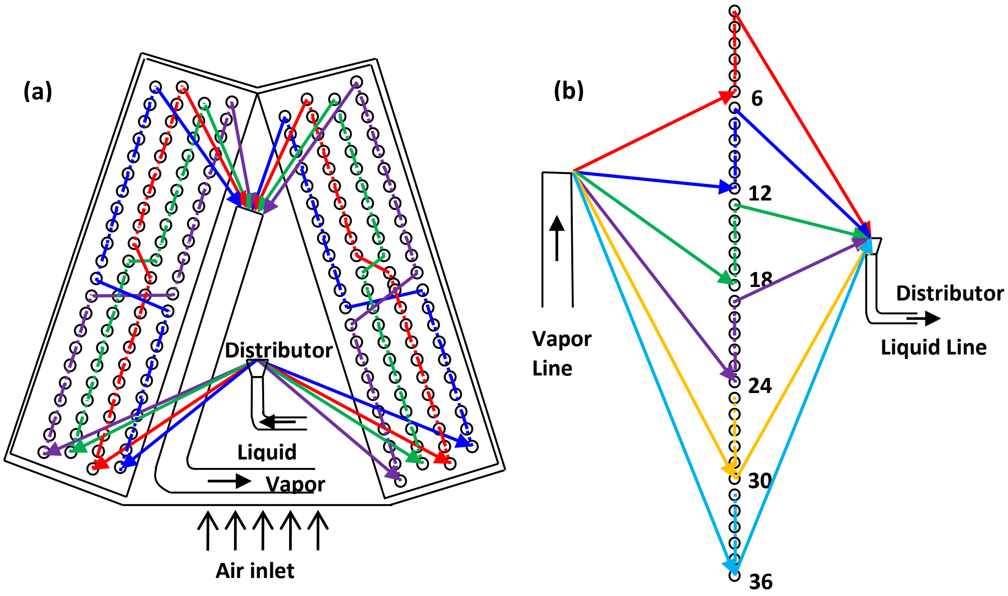

Figure 1: Schematic of baseline HXs: a) evaporator, b) condenser

Table 1: Correlations used in HX Modeling

HX	Type	Phase	Correlation
Evaporator	Heat transfer	Refrigerant - Two Phase	Jung-Radermacher (Jung, et al., 1989)
		Refrigerant - Vapor Phase	Dittus-Boelter (Dittus & Boelter, 1985)
		Air (2-5 mm tube diameter)	Bacellar et al. (Bacellar, et al., 2014)
		Air (9.5 mm tube diameter)	Wang et al. (Wang, et al., 1998)
	Pressure Drop	Refrigerant - Two Phase	Jung-Radermacher (Jung & Radermacher, 1989)
		Refrigerant - Vapor Phase	Blasius (Incropera & DeWitt, 1996)
		Air (2-5 mm tube diameter)	Bacellar et al. (Bacellar, et al., 2014)
		Air (9.5 mm tube diameter)	Wang et al. (Wang, et al., 1998)
	Void Fraction	Two Phase	Koyama et al. (Koyama, et al., 2004)
Condenser	Heat transfer	Refrigerant - Liquid Phase	Dittus-Boelter (Dittus & Boelter, 1985)
		Refrigerant - Two Phase	Cavallini et al. (Cavallini, et al., 2003)
		Refrigerant - Vapor Phase	Dittus-Boelter (Dittus & Boelter, 1985)
		Air (2-5 mm tube diameter)	Bacellar et al. (Bacellar, et al., 2014)
		Air (9.5 mm tube diameter)	Wang et al. (Wang, et al., 1998)
	Pressure Drop	Refrigerant - Liquid Phase	Bhatti-Shah (Bhatti & Shah, 1987)
		Refrigerant - Two Phase	Lockhart-Martinelli (Lockhart & Martinelli, 1949)
		Refrigerant - Vapor Phase	Churchill (Churchill, 1977)
		Air (2-5 mm tube diameter)	Bacellar et al. (Bacellar, et al., 2014)
		Air (9.5 mm tube diameter)	Wang et al. (Wang, et al., 1998)
	Void Fraction	Two Phase	Koyama et al. (Koyama, et al., 2004)

4. OPTIMIZATION PROBLEM

The two objectives in this optimization study are to maximize the COP of the system while minimizing the cost. Equation (1) describes the optimization problems, and Table 2 shows the design space. We include three different optimization studies in this paper. The first one has the fin type as plate fin with and without internal tube enhancement. The second one includes the flat plate and wavy louver fins without internal tube enhancement while the third study has the fin type as wavy louver fin with and without internal tube enhancement. We account for internal tube enhancements in the form of correction factors applied to the refrigerant heat transfer coefficient (HTC) and pressure drop (DP) obtained from the tubes without internal enhancements. We use correction factors of 2.5 and 2.0 for the HTC and DP, respectively. Also, we apply correction factors to the air side HTC and DP to predict the performance of wavy louver fins as compared to the plate fins. The system capacity, HX refrigerant, and the values of the air side pressure drops for the baseline are different from one test to the other. Also, the condenser face area constraint is based on the range of testing and the applicable range of the air velocity in the correlation used, and the baseline air volume flow rate.

$$\begin{aligned}
&Minimize: \ HX \ Cost \\
&Maximize: \ COP \\
&Subject \ to: \\
&\qquad Capacity > Baseline \ [W] \\
&\qquad FA_{evap} < Baseline*1.25 \ [m^2] \ (Baseline = 0.22m^2) \\
&\qquad \Delta P_{ref,evap} < 20000 \ [Pa] \\
&\qquad \Delta P_{air,evap} < Baseline*2 \ [Pa] \\
&\qquad FA_{cond} < 3.6 \ [m^2] \ (Baseline = 1.94m^2) \\
&\qquad \Delta P_{ref,cond} < 40000 \ [Pa] \\
&\qquad \Delta P_{air,cond} < Baseline*2 \ [Pa]
\end{aligned} \tag{1}$$

Table 2: Design space

HX	Design Variable	Unit	Baseline	Range	Variable Type
Condenser	Tube Outer Diameter	mm	9.5	3-5	Discrete
	Fins per inch	in^{-1}	21	20-40	Discrete
	Fin Type	-	Wavy Louver	Flat Plate, and Wavy Louver	Discrete
	Tube Length	m	2.16	1.5-3.5	Continuous
	Vertical Spacing Ratio	-	2.632	1.5-3	Continuous
	Horizontal Spacing Ratio	-	2.632	1.5-3	Continuous
	Number of Tube Banks	-	1	1-2	Discrete
	Circuits per bank	-	6	6-125	Discrete
Evaporator	Tube Outer Diameter	mm	9.5	3-5	Discrete
	Fins per inch	in^{-1}	15	20-40	Discrete
	Fin Type	-	Wavy Louver	Flat Plate, and Wavy Louver	Discrete
	Tube Length	m	0.503	0.15-1	Continuous
	Vertical Spacing Ratio	-	2.105	1.5-3	Continuous
	Horizontal Spacing Ratio	-	2.632	1.5-3	Continuous
	Number of Tube Banks	-	4	2-9	Discrete
	Circuits per bank	-	1	4-125	Discrete

We impose some additional assumptions and design considerations when generating the new HXs for each iteration. First, the total refrigerant flow cross sectional area in the HX is the same as the baseline case. Hence, the total number of tubes for the 3 mm diameter is almost 10 times ($(9.5/3)^2$) the number of tubes of the baseline 9.5 mm tube diameter. This assumption is used to calculate the total number of tubes in the new design. The total number of tubes is then divided by the number of tube banks, which is a design variable, to determine the number of tubes per row. Also, the HX generated for each iteration has counter flow configuration, as opposed to the cross flow baseline design. This is because the counter flow configuration is the most efficient in terms of heat transfer rate per unit surface area. Furthermore, the HX material cost is used as the representative cost, and is calculated from Eq. (2) below. Finally, in Eq. 2, we assume the tube material cost per unit mass to be 1.5 times the fin material cost per unit mass.

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

5. RESULTS AND DISCUSSION

Figures 2 shows the Pareto sets for R-410A for case study 1 (i.e. plate fins with and without internal tube enhancements). The shading of the symbols on this figure indicates the relative evaporator face areas between the designs while the size of the symbols represents the relative condenser face area. However, Figures 3 and 4 show the Pareto sets for both refrigerants for the two other case studies (i.e. plate fins and wavy louver fins (case study 2), and wavy louver fins with and without internal tube enhancements (case study 3)). The shading of the symbols on this figure indicates the relative evaporator face areas between the designs while the size of the symbols represents the relative system charge. The cost reduction is 60% for systems designed and optimized for tests A and C operating conditions. Also, there is potential COP improvement of 20%. For the optimal designs, the evaporator tube diameter is either 3, 4, or 5 mm, while the condenser tube diameter is either 3 or 4 mm. The 5 mm tube diameter condenser designs show lower costs and a better performance than the baseline design. Nevertheless, the 3 and 4 mm condenser tubes have lower cost and better performance than that of the 5 mm tubes. That is because for the same material volume, the 3 and 4 mm tubes show lower air and refrigerant sides thermal resistance. Thus, the 5 mm tubes do not show in the condenser Pareto set designs.

For the small diameter tubes, the required number of tubes to maintain the total refrigerant flow cross sectional area in the HX is much larger (5.6 times for 4 mm tubes, and 10 for 3 mm tubes) than the number of tubes in baseline design. Nevertheless, for many of the Pareto designs, the evaporator and condenser face areas are smaller than the baseline face areas. This is because there exists other design parameters which affect the face area. These parameters include the tube spacing, number of tube banks, and tube length.

Increasing the number of tube banks helps to accommodate the increase in the number of tubes without increasing the face area. However, this causes an increase in the air side pressure drop, which is one of the constraints. Thus, increasing the number of tube banks beyond a certain limit would cause the design to be infeasible due to the high pressure drop. Thus, for all evaporator Pareto designs, the number of tube banks is in the range of 2 to 7 although the design space ranges from 2 to 9.

Another factors that affects the face area compared to that of the baseline include the fin type and the internal tube enhancements. Internal tube enhancements helps to increase the HTC and DP on the refrigerant side while wavy louver fins have the same effect on the air side. Thus, without tube enhancements or when using plate fins, the condenser Pareto designs with a smaller face area than the baseline has 2 tube banks as compared to the 1 tube bank design. However, some condenser Pareto designs with a smaller face area than the baseline still have 1 tube bank similar to the baseline as these designs have either internal tube enhancements or wavy louver fins or both.

Reducing the vertical tube spacing helps reduce the face area because it allows more tubes to fit in the same HX height. It also helps in air flow acceleration and mixing, which improves the heat transfer performance. However, as the tubes get closer, the air side pressure drop increases, which can surpass the constraint level, especially as the number of tube banks increases. Therefore, when using plate fins without internal tube enhancements, only for some Pareto evaporator designs with 4 tube banks, the tubes' vertical spacing ratio was 1.5 (which is the minimum value in the design range). However, for wavy louver fins and/or internal tube enhancements, low spacing ratios exist helping in maintaining small face areas.

For finned tubes, as the fins per inch increases, effects similar to the effects of reducing the tube vertical spacing are seen. Therefore, having higher fins per inch for the small tube diameters helps improve the fin effectiveness by up to 25% of the baseline effectiveness. Furthermore, shorter HX tubes cause a decrease in the HX face area. In order to obtain the same heat load as the baseline designs, shorter tubes are only possible if the heat transfer in the small diameter tubes is improved. This is dependent on the air and refrigerant side thermal resistances and the tube circuitry. In general, using wavy louver fins and internal tube enhancements help maintain the same performance as the optimum designs with plate fins and no enhancements but at smaller face area and much lower HXs' cost.

For most of the Pareto designs, especially with plate fins and no internal enhancements, the number of refrigerant tube inlets is equal to the maximum number allowed in the design problem (i.e. the maximum number of circuits is half the number of tubes per tube bank). Also, this high number of refrigerant inlets and the use of shorter tubes lead to a significantly lower refrigerant pressure drop. Nevertheless, one of the main causes of refrigerant flow mal-distribution in HXs is the mechanical design of headers and feeder tubes (Muller & Chiou, 1988). In this study, we do not account for air or refrigerant flow mal-distribution. Thus, from a practical point of view, the aforementioned high number of refrigerant feeder tubes is expected to cause or increase the refrigerant mal-distribution, which in turn might negatively affect the performance of the HXs. Furthermore, one manufacturing aspect of the optimized designs that needs to be considered is the need for two different sizes of U-bends in each HX. This is because of the different values for vertical and horizontal tube spacing in each HX.

Figures 3 and 4 show the refrigerant charge of the optimum designs compared to the baseline system. The charge of the baseline system can be reduced by 35% by shifting towards the optimized low cost small tube diameter HX designs for both refrigerants. It is worth noting that the system charge calculation is based on the internal volume of the HXs and the void fraction correlation (Koyama, et al., 2004). Also, the accuracy of the charge calculation is dependent on the void fraction correlation.

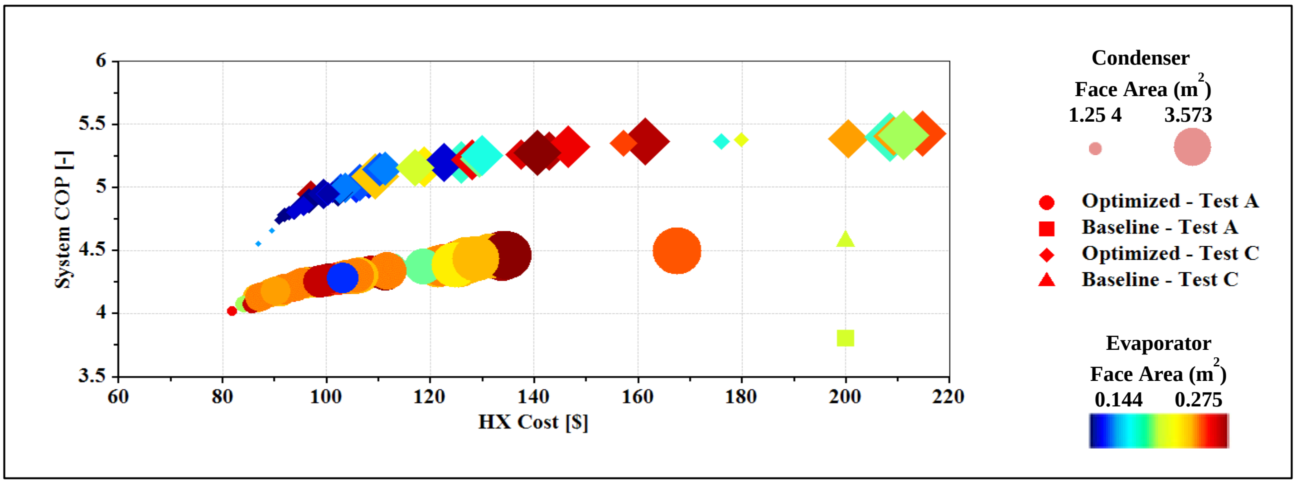

Figure 2: R-410A Pareto sets for case study 1

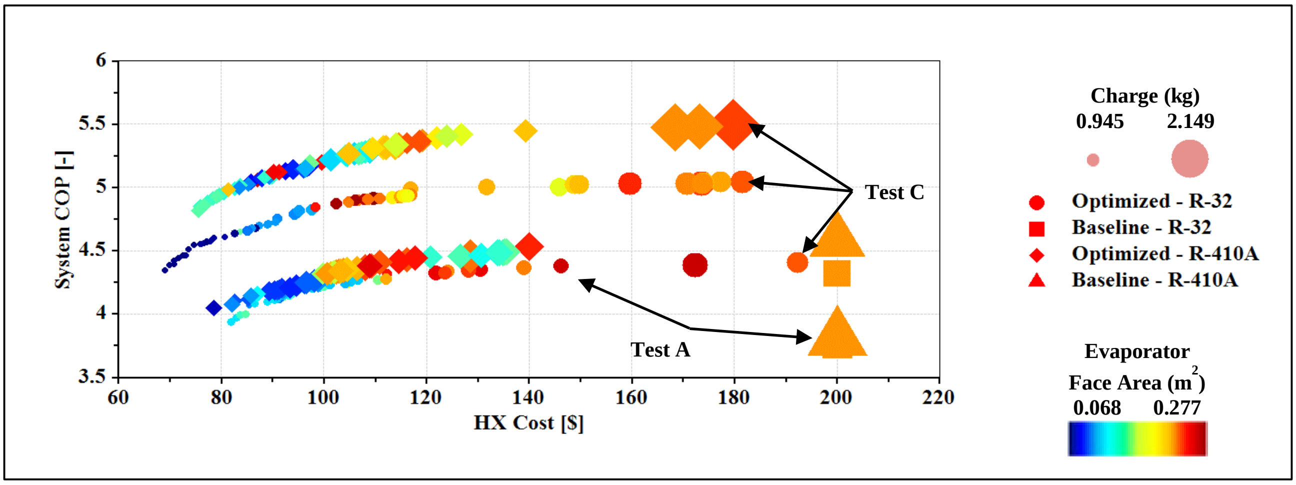

Figure 3: Pareto sets for case study 2

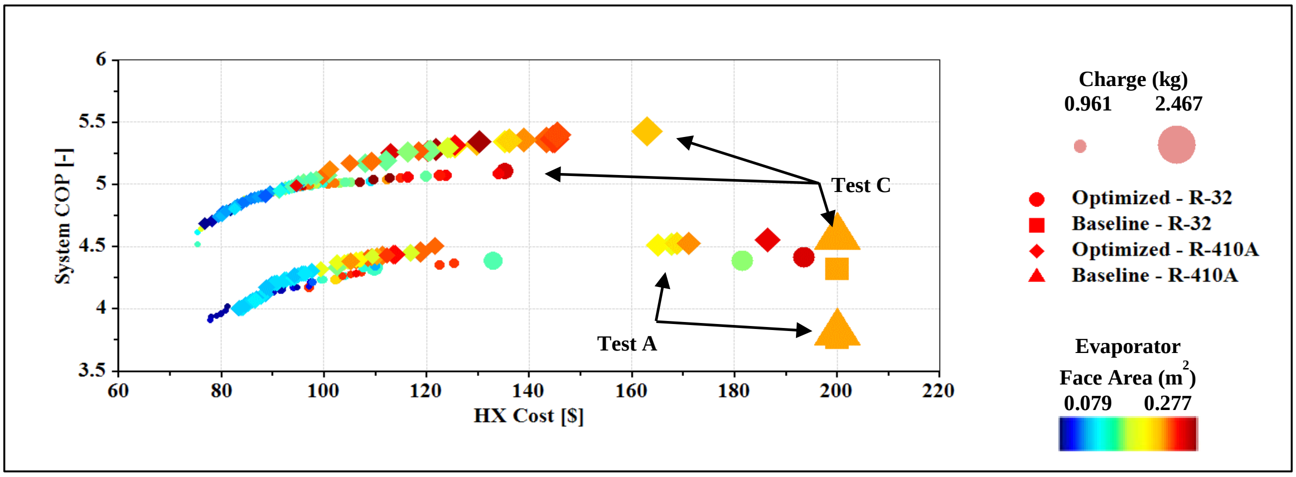

Figure 4: Pareto sets for case study 3

Figure 3 and 4 show that the charge for the R-32 systems, whether baseline or optimized, is much lower than the R-410A systems. This is because the density of R-32 is much smaller than R-410A. This is another benefit to the R-32 system, on top of its lower GWP, which makes it promising as a replacement for R-410A. Also, the improvements for each refrigerant for the systems designed and optimized for tests A and C operating conditions are comparable. Nevertheless, when running an individual Pareto design that shows higher cost reduction at one operating conditions (e.g. test C) at the other operating conditions (e.g. test A), it might not meet the capacity constraint or COP at the latter conditions (test A). Thus, it is important when designing and optimizing an ASHP system to make sure that it satisfies all the constraints at different testing and operating conditions. Also, it is clear from this study, especially for the third optimization study, that the performance of the optimized R-32 system can exceed that of the optimized R-410A system especially in the low cost region. This provides an additional motivation to focus on R-32 as a replacement to R-410A in the ASHP systems. Moreover, although HX circuiting optimization potentially helps to improve the HX performance and causes further material reduction, we focused in this study on simple refrigerant circuiting rather than optimized or complex circuits. This is for two reasons. The first is that, given the current optimized designs, the potential additional improvements are expected to be small. The second reason is that with the large number of HX tubes and feeder tubes, further circuiting is expected to impose higher flow maldistribution and HX manufacturing difficulties.

6. CONCLUSION

This paper presents the multi-objective optimization of a baseline R-410A ASHP system to determine the potential system performance improvements and material savings when using small diameter tubes in the HXs. The goals are to minimize the system cost and maximize the system COP. The study compares the optimum designs when using a conventional refrigerant R-410A and an alternative lower GWP refrigerant, R-32. The HXs cost is the representative cost of the system cost, and the HXs geometry, circuity, fin type, and internal tube enhancements are the design variables. The optimal designs have evaporator tube diameters of either 3, 4, or 5 mm, and a condenser tube diameters of either 3 or 4 mm. The HXs cost can be reduced by 60% while the COP improvement for the same HXs cost is 20%. Furthermore, a charge reduction of 35% is possible in the optimized ASHP system for both refrigerants. These material and charge reduction help to reduce the environmental impacts of the vapor compression systems while maintaining the same system performance. Also, it helps to design and manufacture systems with higher seasonal energy efficiency ratio without the need to increase the size or material used in the baseline HXs.

NOMENCLATURE

Symbol	Quantity	Units
C	HX Cost	$
FA	Face Area	m^2
MC	Material Cost per unit mass	$ kg^{-1}
ρ	Density	kg m^{-3}
P	Pressure	Pa
V	Volume	m^3

Subscripts

air	Air
cond	Condenser
evap	Evaporator
fin	Fin
ref	Refrigerant
tube	Tube

Abbreviations

AREP	Alternative Refrigerants Evaluation Program
ASHP	Air Source Heat Pump
COP	Coefficient of Performance
GWP	Global Warming Potential
HVAC	Heating, Ventilation, and Air Conditioning
HX	Heat Exchanger
MCHX	Micro-channel Heat Exchanger
MOGA	Multi-Objective Genetic Algorithm

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