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REIMAGINING HEAT EXCHANGERS FOR NEXT GENERATION ENVIRONMENTAL SYSTEMS

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ABSTRACT

Air-to-refrigerant heat exchangers (HXs) are essential components in space conditioning, refrigeration, and power systems, and recent efforts have focused on making these devices more compact, reducing refrigerant charge and lowering manufacturing costs. Historically, HX innovation has been limited by available computational resources, design tools, and manufacturing constraints. The best available technologies utilize tube-fin and micro- or macro-channel tubes with fins, which are not necessarily the optimal designs achievable with current technology. In this paper, we highlight the latest advancements in air-to-refrigerant HXs, specifically emphasizing innovations achieved through shape and topology optimization. A multi-scale design optimization approach is introduced, alongside similar methods in literature, which enable highly sophisticated shape-optimized tube designs with more than 50% reduction in size and 25% reduction in refrigerant charge, essential for A3 refrigerant charge limit compliance. The frameworks integrate traditional heat and mass transfer science with state-of-the-art machine learning, genetic algorithms, and adjoint algorithms to create novel designs. While many of these innovative designs may not be manufacturable using conventional methods, they allow us explore the boundaries of what is possible. These novel air-to-refrigerant HXs are key enablers for ultra-low-refrigerant charge heat pump and refrigeration systems.

Keywords: Air-to-refrigerant heat exchanger; Charge reduction; Heat pump; Shape optimization

INTRODUCTION

Air-to-refrigerant heat exchangers (HX) are critical components in space conditioning, refrigeration, and power systems, partially determining the system COP, charge, and footprint. Recently, there has been a significant focus on making these devices more compact, reducing refrigerant charge, and lowering manufacturing costs, which is particularly critical for mobile systems used in consumer and specialized applications.

Historically, HX innovation has been limited by available computational resources, design tools, and manufacturing constraints. Modern HX technologies leverage tube-fin and micro- or macro-channel tubes with enhancements such as louvered fins, internal grooves, and multi-port microchannels. While these HXs are in production today, they are not necessarily the optimal designs achievable with current technology.

In this work, we highlight the latest advancements in air-to-refrigerant HX design, specifically emphasizing innovations achieved through shape and topology optimization. These innovative air-to-refrigerant HXs exhibit substantial reductions in size, weight, and refrigerant charge compared to conventional designs and are poised to be key enablers for the ultra-low-refrigerant charge heat pumps and refrigeration systems of the future.

AIR-TO-REFRIGERANT HX INNOVATION: LATEST DEVELOPMENTS

The continued advancement in air-to-refrigerant HX performance will play a key role in achieving worldwide sustainability goals. In this section, we review some of the most pertinent recent developments in air-to-refrigerant HX design and optimization literature, with a particular focus on HX innovations which leverage shape (i.e., finding the best tube and/or fin shapes) and topology (i.e., distribution / configuration of tubes and/or fins) optimization.

Variable Geometry HXs (VGHXs) are a class of air-to-refrigerant HXs with unconventional geometries which do not conform to typical textbook example HXs, i.e., rectangular HXs with uniform tube and fin sizes, shapes, and pitches throughout the HX core (Fig. 1). Such designs include HXs where multiple tube shapes, fin types, pitches, and general non-uniformity are leveraged. This increasing level of complexity can be utilized to accommodate and account for performance degradations associated with system packaging, e.g., airflow maldistribution [1].

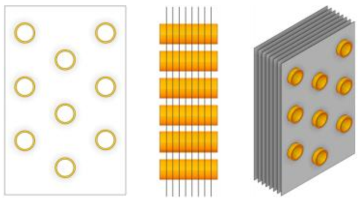


Fig. 1 "Textbook" plain fin and tube HX.

A common example of VGHXs come in the form of variable fin pitch, which is especially common in low temperature applications (Fig. 2) [2], the idea being to achieve an even distribution of frost growth through the entire depth of the HX towards extending system operating time [3-4]. Similarly, variation in fin shape can offer benefits of benefits similar to traditional surface enhancement, e.g., through topology-optimized fin winglets to improve airflow near the fin sheet [5], minimizing material and conforming to package space limitations with accelerating flow evaporators [6], or replacing the traditional A-type HX configuration with a single, continuous C-shape fin for improved condensate drainage [7] (Fig. 3).

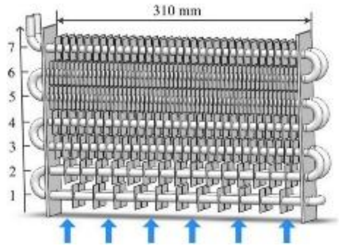


Fig. 2 Staggered fin "frost-free" evaporator [2].

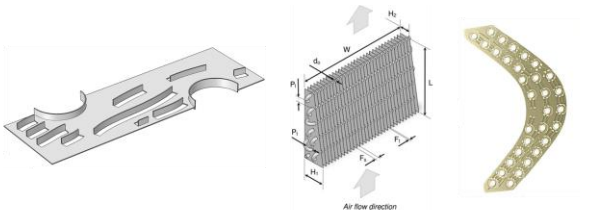


Fig. 3 (Left) Segment of fin-tube HX with topology optimized winglets [5]; (Center) Accelerating flow evaporator [6]; (Right) C-type Fin HX [7].

Shape and topology optimization of the tube surface, i.e., the HX primary area, presents considerable opportunity for improving overall HX performance by eliminating the need for fins, thereby avoiding drawbacks such as increased hydraulic penalties, fouling/frosting potential, and material consumption. Bacellar et al. [8] and Tancabel et al. [9] presented experimentally-validated HX design frameworks based on Approximation-Assisted Optimization (AAO) using Genetic Algorithm with on-the-fly shape and topology change to determine the best tube shapes and layouts for the application of interest. These frameworks [8-9] enabled highly sophisticated shape-optimized tube designs with more than 50% reduction in HX size and 25% reduction in refrigerant charge.

However, these works consider a HX core containing a single non-round tube shape defined using a parametric optimization approach. Advancements in additive manufacturing technology have enabled the use and manufacture of HX designs produced using gradient-based adjoint optimization methods, where sensitivity-based algorithms enable shape optimization for a given metric of interest based on a prescribed seed geometry and iterative mesh and domain morphing. This methodology was adopted by Klein [10], who applied adjoint optimization to an entire HX core and achieved a 45% and 42% reduction in airside pressure drop as compared to all-round and all-non-round tube HX baselines (Fig. 4). While the primary variation in shape was along the direction of airflow, a noticeable change occurred at the top and bottom of the tube bank where end effects are present, suggesting that using different tube shapes throughout the HX core may improve performance in designs with more complex flow fields. Additionally, it should be noted that the all-non-round tube baseline had an internal volume 43.5% smaller than a state-of-the-art microchannel HX [8]. Considering that the average airside HTC of the new designs increased by up to 15%, it is not unreasonable to assume that even greater reductions in internal volume (and refrigerant charge) are possible when multiple tube shapes are employed.

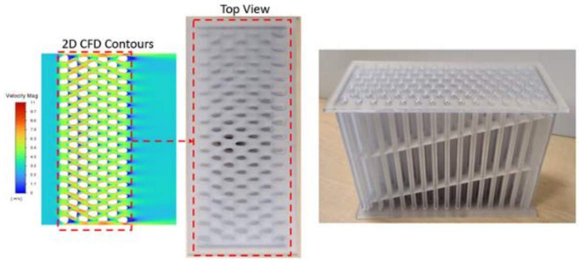


Fig. 4 Adjoint optimized HX [10]: (Left) Flow contours, (Right) 3D-printed prototype top / side views.

The "ultimate HX" is one which considers a fully variable HX core with multiple tube and fin shapes optimally configured to the specific application(s) of interest, e.g., fin-tube HXs [11] and microchannel HXs [12] accounting for variations in fin pitch, tube pitch, and tube dimensions (Fig. 5).

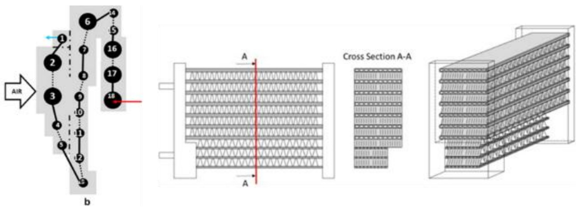


Fig. 5 Fully Variable HXs: (Left) Fin-Tube VGHX [11]; (Right) Microchannel VGHX [12].

One benefit of these highly-flexible HXs is the ability to tailor different sections of the HX to different parts of the transfer process. For example, the subcooled section of a microchannel condenser generally has fewer tubes than the rest of the HX while channeling the same mass flow rate. An increase in pressure drop can be mitigated by adjusting the tube and port dimensions for an increase in cross sectional area and minimal change in mass flux [13]. Furthermore, this region is usually responsible for a small portion of the overall heat load but contains a large fraction of the charge. To this end, Huang et al. [14] designed a VGHX to minimize the sub-cooled section and reduced material volume by >12% and minimal change in capacity. Though not reported, it is reasonable to believe this will reduce the refrigerant charge as well. O'Malley et al [15] designed an air-to-R290 VGHX condenser for an A/C application using non-round tube shapes and multiple tube spacings in a single HX core, achieving more than 20% reduction in HX envelope volume and 64% reduction in refrigerant charge vs. a 5.0 mm fin-tube HX baseline. Further, their VGHX achieved more than 36% additional charge reduction compared to a similar-sized optimal HX which utilized non-round tubes and a single tube spacing in the entire HX core, further demonstrating the potential for VGHXs to outperform their "conventional" counterparts.

VGHX OPPORTUNITIES AND CHALLENGES

VGHXs present significant opportunity to the HVAC&R design community toward achieving sustainability targets through improving heat pump system performance. VGHXs have been shown to outperform their conventional counterparts when a key variable (tube pitch, shape, fin pitch, fin type) is allowed sufficient variation through the HX core. This results in increased thermal-hydraulic performance but can also be used to reduce the impact of frost accumulation and flow maldistribution on both the air and refrigerant side.

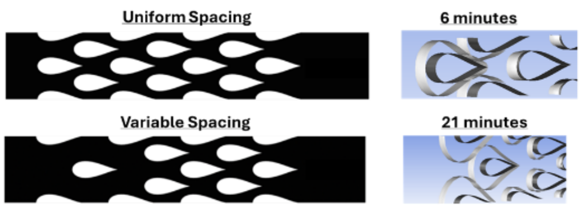
However, there is no free lunch. Because VGHXs have a much larger design space than typical HXs, there is a significant computational cost associated with their development. Additionally, arbitrary changes in key variables are more likely to decrease performance, e.g., manufacturing tolerances can have increased effect on performance for tubes with very small characteristic diameters [16]. This was corroborated by Abdelaziz and Radermacher [17], who showed that variation in tube position can cause a -10% to +53% change in airside heat transfer coefficient.

In order to fully take advantage of VGHXs, an accurate and computationally efficient methodology is needed to rapidly evaluate the variation in tube/fin pitch, type, and shape on overall HX performance.

VARIABLE GEOMETRY HEAT EXCHANGERS FOR IMPROVED FROSTING RESILIENCY

Frost accumulation is a common problem for HXs operating in low temperature high humidity environments such as supermarket refrigeration systems and cold climate heat pumps. For conventional designs blockage from frost (or other matter) occurs between fins, but as HXs become increasingly compact and tube diameters continue to decrease the space between tubes may become obstructed as well [9]. This may be addressed by applying a staggered tube-to-tube spacing throughout the HX core in a similar manner to the staggered fin spacing used in "frost free" evaporators (Fig. 2) [2].

To quantify the impact of variable tube-to-tube spacing on frosting performance of non-round tube VGHX, a case study was conducted using Fluent Icing [18] to simulate the rate of frost accumulation on two different non-round tube VGHXs (Fig. 6). For both HXs, the inlet temperature was set to 1.7˚C, the tube wall temperature was set to -18.3˚C, the inlet velocity was set to 1 m/s, and the inlet RH was to 71%.



	Uniform Spacing	Variable Spacing
Q/V [MW/m³]	2.9	2.4 (17% ↓)
Time to blockage [min]	6	21 (320% ↑)

Fig. 6 Impact of tube spacing on frosting and heat transfer performance of a non-round tube VGHX.

The lighter gray contours on the left of the figure represent the edge of the growing frost layer on the (darker gray) tubes. Once the frost layers from adjacent tubes intersect, we consider the HX blocked, and the simulation is terminated. Compared to the uniform tube spacing, the variable tube spacing design can operate for 3.5x longer under these operating conditions. While this incurs penalty in terms of heat transfer, it is marginal compared to the increase in operating time.

OPTIMIZATION OF A COLD CLIMATE HEAT PUMP OUTDOOR UNIT USING VARIABLE GEOMETRY HEAT EXCHANGERS

A VGHX optimization framework (Fig. 7, top) was applied sequentially to the design of a ~7.3 kW outdoor unit for improved frosting performance in a cold climate heat pump application. The HX optimization framework consists of an AAO involving automated CFD [8-9, 19], Kriging metamodels [20], and optimization with multi-objective genetic algorithm (MOGA) [21] where all tube shapes are represented parametrically using Non-Uniform Rational B-Splines [8-9] (Fig. 7, bottom). Full HX models are built and simulated using an extension of the experimentally-validated air-to-refrigerant HX model from Jiang et al. [22] using empirical correlations for single and two-phase refrigerant flow in small channels and airflow over non-round tube bundles [9].

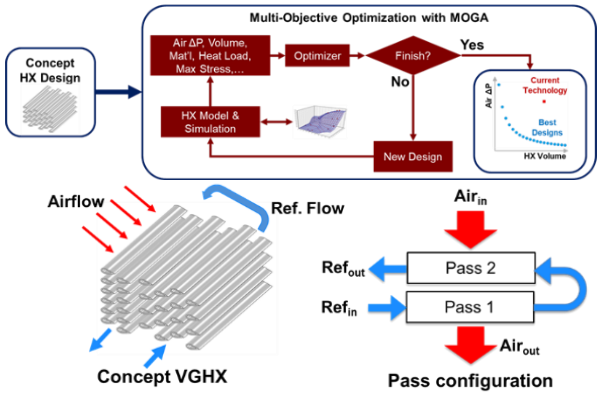


Fig. 7 (Top) VGHX optimization framework (Bottom Left) Concept VGHX; (Bottom Right) Counter-cross pass configuration.

We aim to optimize the outdoor unit (i.e., evaporator) of an R410A cold climate heat pump defined by Tang et al. [23], where the baseline fin-tube HX utilizes 7.2 mm round tubes, targeting minimizing airside pressure drop and HX envelope volume (Eq. (1)) for each pass of a two-pass VGHX with shape-optimized non-round tubes (Fig. b). Here, optimization is carried out for all points along the initial (P1) Pareto front. Additionally, the first pass is designed to meet ~50% of the baseline HX capacity with no more than 50% of the total refrigerant side pressure drop. The second pass is constrained to ensure that the superheat is within 1 K of the baseline HX, thus delivering the same total heat load.

$$\begin{array}{ll} & \min \Delta P_{air}\,;\min V_{env} \quad \text{s.t.} \\ \text{Pass 1:} & \Delta P_{P1,ref} \le 0.5\cdot\Delta P_{BL,ref} \quad \Delta P_{P1,air} \le \Delta P_{BL,air} \\ & 0.5\cdot\dot{Q}_{BL} \le \dot{Q}_{P1} \le 0.55\cdot\dot{Q}_{BL} \quad 0.5 \le H_{P1}/L_{P1} \le 2.0 \\ \hline & \min \Delta P_{air}\,;\min V_{env} \quad \text{s.t.} \\ \text{Pass 2:} & \Delta P_{P2,air} \le \Delta P_{BL,air} \quad \Delta P_{P1,ref} + \Delta P_{P2,ref} \le \Delta P_{BL,ref} \\ & \Delta T_{SH,BL} \le \Delta T_{SH,P2} \le \Delta T_{SH,BL} + 1\,\text{K} \\ & H_{P2}, L_{P2} \text{ within 2\% of } H_{P1}, L_{P1} \end{array} \tag{1}$$

The optimal VGHXs significantly outperform the baseline fin-tube HX across multiple key metrics (Fig. 8), with reductions of up to 87% for envelope volume and 76% for airside pressure drop. Significant improvements in other key metrics (e.g., face area, material volume, internal volume) are also achieved, even though they were not directly considered as objective functions. For a fixed inlet air volumetric flow rate, decreasing the face area increases the inlet air velocity and thus airside heat transfer coefficient and pressure drop. The former enables a reduction in heat transfer area and material volume, while designs with very low hydraulic resistance have correspondingly low airside velocities (and HTCs) which is compensated with more material.

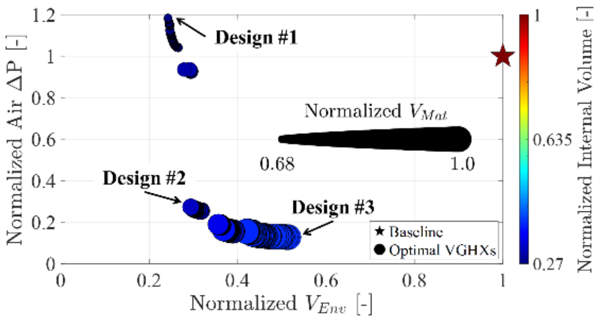


Fig. 8 Optimization results (Normalized w.r.t. baseline fin-tube outdoor unit)

Three designs are selected for cycle-level analysis using an experimentally-validated steady-state simulation tool [24]. As seen in Fig. 9, all three units with the optimal VGHX outdoor unit provide the same capacity and COP as the baseline fin-tube HX while still significant improvements in other key metrics. All HXs have a reduction in material volume, internal volume, and charge. While the majority of HX charge tends to be stored in the condenser, replacing the evaporator with an optimized VGHX design still manages ~11% reduction in overall cycle-level charge.

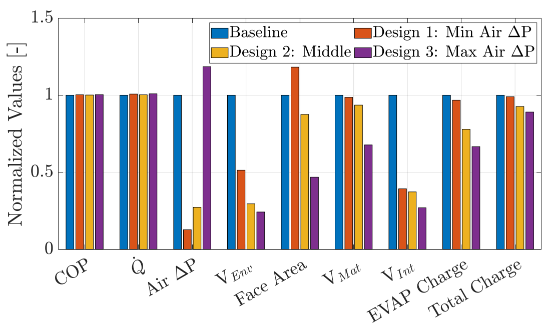


Fig. 9 System performance using optimal VGHX

CONCLUSIONS

Air-to-refrigerant HXs are critical components in all space conditioning, refrigeration, and power systems. While HX innovation has largely been limited by available computational resources, design tools, and manufacturing constraints, recent efforts have begun exploring cutting-edge designs which leverage complex variable geometries including multiple tube and fin shapes / types and pitches towards achieving maximal component and system-level performance in the smallest possible package sizes. We present a multi-scale design optimization approach which can enable highly sophisticated variable geometry HXs with shape-optimized tube designs and non-uniform fin spacings that can enable significant reductions in HX size (>50%), refrigerant charge (>25%), and longer operating times (>3x longer vs. HXs with uniform tube pitch) under frosting conditions. These innovative HXs are poised to be key enablers for the ultra-low-refrigerant charge heat pumps and refrigeration systems of the future.

NOMENCLATURE

H : HX height, m
L : HX length, m
ΔP : pressure drop, Pa
$\dot{Q}$: heat load, W
ΔT_{SH} : evaporator outlet superheat, K
V : HX volume (envelope, internal, material), m^3

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