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ABSTRACT

The ultralow Global Warming Potential (GWP) of propane (R290) and isobutane (R600a) refrigerants make them highly attractive for refrigerator and freezer applications, although their flammability necessitates strict use conditions with respect to refrigerant charge. Copper tubes with smaller diameters are widely used to reduce refrigerant charge. The process of downsizing copper-tube diameters involves detailed simulations and prototype construction as well as testing and validation. A proprietary heat exchanger design and simulation software tool (Jiang *et al.*, 2006) was used to evaluate the performance of and optimize the design of domestic refrigerator condenser coils made with 5-mm outer-diameter copper tubes. Optimization was accomplished through the use of reduced order models, meta-models and a multi-objective genetic algorithm (MOGA). Reducing refrigerant charge was the primary objective. Secondary objectives included the reduction of the total footprint and the total tube-and-fin material mass. The baseline design used 6.35-mm O.D. copper tubes with a minimum wall thickness of 0.41 mm, *i.e.*, quarter-inch tubes with 0.016-inch wall thickness. The new designs use wavy-herringbone fins with reduced fin thicknesses as compared to the baseline design. Other variables included the horizontal and vertical spacing of the tubes; number of tubes per bank; fin density; wavy fin pattern depth; tube length; and tube circuitry. For an R600a residential application, reduced internal volume was considered to be more important than the airside pressure drop. A Pareto chart is presented of optimized values from the design space. Compared to the baseline design, the best 5-mm design reduced the internal tube volume by 41 percent, along with a 57 percent reduction in coil footprint. Additionally, test data to validate the performance of prototype coils is presented with emphasis on the design, construction and manufacture of the heat exchanger coils.

1. INTRODUCTION

The tightening of environmental regulations on fluorinated gases and loosening of safety regulations on flammable gases is encouraging a global transition from high-GWP HFCs to ultralow GWP hydrocarbons in refrigerators and freezers. The use of these flammable low-GWP gases and the concurrent development of heat exchangers made with smaller-diameter copper tubes is a fascinating story. The U.S. Environmental Protection Agency (EPA) Significant New Alternatives Policy (SNAP) program lists various hydrocarbons as “acceptable, subject to use conditions” as a substitute for CFCs and HFCs. The SNAP program generates lists of acceptable and unacceptable substitutes for each of the major industrial use sectors as well as specific product categories. Once it was established that these flammable refrigerants were acceptable substitutes for high-GWP refrigerants, the EPA proceeded to change the status of environmentally undesirable F-Gases from acceptable to unacceptable (*e.g.*, EPA, 2016).

For certain commercial refrigeration equipment, one of the "use conditions" limits hydrocarbon refrigerant charge to 150 grams (5.3 ounces). For domestic refrigerators and freezers, the charge limit is 57 g although proposals remain in play to raise the charge limit for hydrocarbons in domestic refrigerators to 150 g. The European Union and other regions have long used a 150 g limit for hydrocarbons in domestic refrigerators. Reduced refrigerant charge is essential when propane or isobutane are used as refrigerants because according to ANSI/ASHRAE 34-2016, Designation and Safety Classification of Refrigerants, these gases have a safety classification of A3, indicating that they have low toxicity but are highly flammable.

2. RECENT RESEARCH ON SMALLER DIAMETER COPPER TUBES

When regulators signaled that hydrocarbons would be allowed in light commercial applications, the technology of MicroGroove^{TM} smaller-diameter inner-grooved copper tubes was already available for use in manufacturing, having been developed nearly a decade ago for use with residential air conditioners and traditional refrigerants (Tetzloff *et al.*, 2016).

Current research on smaller diameter copper tubes has focused on the use of low-GWP refrigerants. In particular, laboratory experiments on smaller diameter copper tubes were conducted at three universities: the University of Padova (Mancin *et al.*, 2016, Longo *et al.*, 2016); Tokyo University of Marine Science and Technology (Inoue *et al.* 2016); and Kyushu University, Fukuoka, Japan (Nakamura *et al.*, 2016). Such research provides predictive correlations that can be used in simulations of the performance of coil designs that use smaller diameter copper tubes with a variety of inside-the-tube enhancements (*i.e.*, microfins) and low and ultra-low GWP refrigerants.

Previously, Kondo and Hrnjak (2012) carefully examined the condensation behavior of R410a and R744 in 6.1 mm copper tube with and without inner fins. They found that a very thin film of condensate forms even in the desuperheating zone and that this film affects heat transfer in that zone. Kaji *et al.* (2012) compared three types of copper tubes, including smooth and inner grooved tubes, by viewing R744 flow inside the tubes through glass. It was found that inner grooving can be effective in removing oil away from the inner surface of the tubes and thus enhancing performance.

Smaller-diameter copper tubes have also been compared to microchannel tubes. The International Copper Association, Inc. (ICA) sponsored a research project to allow for meaningful comparisons of heat exchangers made from these disparate tube technologies (Hipchen *et al.*, 2012). The method of comparison is simple: A search was made for a state-of-the-art, best-in-class brazed aluminum microchannel heat exchanger. The performance specifications were then identified and set as a target for a round-tube-plate-fin (RTPF) heat-exchanger with smaller diameter copper tubes. The design space considered candidate RTPF designs that met this performance specification. In another study, Cremaschi *et al.* (2012) compared frosting and drainage of coils made with smaller diameter (5 mm and 7 mm) and conventional diameter (9.5 mm) copper tubes with microchannel tubes. The copper tubes showed excellent water drainage and good performance in frosting operating conditions as compared to the microchannel tubes.

Wu and Ding from Shanghai Jiao Tong University (SJTU) were coauthors with Zheng, Gao and Song from the Shanghai Office of the International Copper Association (ICA) of a research report describing software programs for optimizing coils with smaller diameter copper tubes (Wu *et al.*, 2012). The software was developed by ICA in cooperation with a consortium of OEMs representing the majority share of production of room air conditioners globally. The paper describes a step-by-step procedure for optimizing heat exchanger design and illustrated the principles with case studies.

Shabtay *et al.* (2014) reviewed the subject of RTPF heat exchangers for alternative refrigerants. They concluded that new copper-based technologies for heat exchangers are available to enable a smooth transition to alternative refrigerants in residential and commercial air-conditioning systems and commercial refrigeration systems; and that the total heat-transfer performance is improved using small-diameter inner-grooved tubes with additive benefits from both internal surface enhancement and diameter reduction.

The core benefits of smaller diameter copper tubes with inside-the-tube enhancements include energy efficiency, less material usage and reduced refrigerant charge. Tube-diameter reduction and inner enhancements result in more effective heat transfer and consequently smaller, lighter coils. The smaller internal volume of the coils and higher heat transfer coefficients mean that less refrigerant is necessary to charge the coil. The need for less refrigerant results in other design advantages including a further reduction in overall system weight. For systems using flammable refrigerants, the smaller refrigerant volume allows for the desired cooling capacities to be reached without exceeding the charge limits imposed by the regulatory authorities.

3. SIMULATIONS OF ISOBUTANE CONDENSERS

A manufacturer of high-end, built-in residential refrigerators, freezers and wine coolers pioneered the dual-refrigeration concept which provides separate and independent sealed systems for the refrigerator and freezer. In this design, the same condenser has two refrigerant circuits, one for each of two independent vapor compression cycles. One circuit is for the refrigerator compartment; and the other is for the freezer compartment. An example is depicted in Figure 1.

The 57-gram limit will necessitate new designs as F-Gases are phased out by the year 2021. Existing appliances typically use 7.9 mm (five-sixteenths inch) or 6.35 mm (one-quarter inch) copper tubes. As smaller-diameter MicroGroove tubes were proposed to meet the 57-gram "use conditions" on isobutane, a proprietary heat exchanger design and simulation software tool (Jiang *et al.*, 2006) was used to identify a feasible 5 mm replacement coil design.

The baseline condenser coil uses 6.35 mm (quarter-inch) O.D. copper tubing with flat plate fins and a low fin density. As described above, the condenser has two refrigerant circuits, with each circuit serving an independent vapor compression cycles for the refrigerator and freezer compartments. A baseline condenser model was developed and validated against experimental data prior to conducting the optimization analysis.

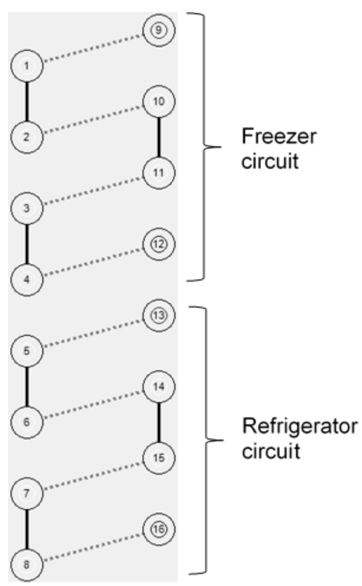


Figure 1: Condenser dual circuit design

Prior to evaluating potential small diameter replacements, a study was conducted to first evaluate the effect of refrigerant circuitry on the existing coil performance. Three operation modes were evaluated: refrigerator circuit running, freezer circuit running, and both circuits running. It was found that for a parallel tube circuitry, heat dissipation and rejection increased for both refrigerator and freezer circuits during all operation modes. Based on this review, this improved circuitry was selected for use in the optimization study.

The proposed designs differed from the baseline design in circuitry configuration. All of the optimized designs used the same parallel circuitry, which differed from the "series" circuitry of the baseline. A circuitry evaluation that was conducted prior to the optimization study suggested that the parallel circuitry design would perform better overall (in all operation modes) than the series design.

4. OPTIMIZATION STUDY

An optimization study was conducted to identify condenser designs that can provide equal performance to the baseline while reducing refrigerant charge. In this study a multi-objective genetic algorithm (MOGA) was used to solve the optimization problem. The focus of this study was on new 5mm-tube heat exchanger designs to replace existing (6.35mm diameter tube) condensers. Exploratory studies were also conducted on even smaller diameter tubes but such preliminary results are not reported here.

Genetic algorithms are a type of evolutionary algorithm. Evolutionary optimization differs from classical optimization in that a population of possible solutions is evaluated in each iteration instead of just one, allowing for the utilization of parallel processing. The study begins with the creation of a population of random designs. These designs are evaluated and ranked based on their performance, and a new population of designs is created from the "children" of the best designs. This process continues for a set number of iterations, ultimately providing designs that are significantly improved over the baseline (Deb, 2001; Aute & Radermacher, 2014).

Multi-objective optimization refers to optimization where multiple objective functions are minimized or maximized simultaneously. MOGA is therefore the combination of the two methods. The primary objective of the proposed optimization study was to design a condenser coil that would equal the performance of the baseline coil while lowering the refrigerant charge. Secondary objectives were to reduce the total footprint of the coil and the total tube-and-fin material mass.

This optimization study had two objective functions: 1) to minimize total tube internal volume, normalized (internal volume / baseline internal volume); and 2) to minimize airside pressure drop, normalized (airside pressure drop / baseline airside pressure drop). These objective functions are reflected in the chart of Pareto designs.

Optimization problems often are subjected to constraints which any feasible solution must satisfy. The following constraints were imposed on the optimization study:

- Heat rejection greater than or equal to the heat rejection of the baseline design;
- Sub-cooling equal to or greater than sub-cooling of the baseline coil;
- Saturation temperature drop kept within one degree of the baseline; and,
- Airside pressure drop within acceptable range for the existing condenser fan.

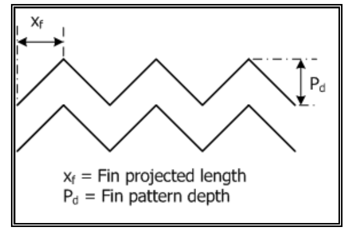
The design variables of the optimization study included the following: heat exchanger length, fin density, horizontal tube spacing, vertical tube spacing, and fin geometry. The optimization solver can handle both discrete and continuous variables. In the optimization study for 5mm-tubes, all variables were continuous. Fin geometry parameters were considered to be continuous.

Table 1 shows the design space of the variables in the optimization study. The fin geometry ranges were those that were supported by the air-side heat transfer coefficient and pressure drop correlations for wavy fins in small diameter tube-fin heat exchangers (Bacellar et al., 2016).

The optimization study produced 5 mm tube designs with significantly less internal tube volume than the baseline coil. The best 5 mm design reduced the internal tube volume by as much as 41 percent as compared to the baseline, along with a potential 57 percent reduction in coil footprint, while maintaining a near equal airside pressure drop. The results for the best 5 mm designs are plotted in Figure 2.

The new designs use wavy-herringbone fins with a reduced fin thickness as compared to the baseline. Other variables included the horizontal and vertical spacing of the tubes; number of tubes per bank; fin density; wavy fin pattern depth; and tube length.

Table 1: Optimization Design Space

Variable	**Type**	**Lower Limit**	**Upper Limit**	
P_l/D_o (horizontal spacing / outer tube diameter)	Continuous	1.25	4	
P_t/D_o (horizontal spacing / outer tube diameter)	Continuous	1.25	4	
Height fraction (height/baseline height)	Continuous	0.5	1.0	
Fin density [FPI]	Continuous	7	40	
P_d/X_f (fin pattern depth / fin projected length)	Continuous	14	16	
Tube Length [m]	Continuous	0.305	0.432	

As can be seen in Figure 2, all of the optimized 5-mm designs that are featured increase the airside pressure drop as compared to the baseline. Multiple factors contributed to an increase in airside pressure drop for the smaller diameter tube coils. In particular, the optimized designs feature wavy fins, which inherently contribute to higher airside pressure drop than the flat fins used by the baseline. The face area was also reduced for all coils to maintain the tube spacing ratio; for a fixed airflow volume, a reduced face area increases the air velocity, which in turn increases the airside pressure drop. These pressure drop increases were deemed reasonable, considering the current system fan selection. The power consumption of the condenser fan is typically a few watts, which is quite low compared to the compressor power even considering any proposed increase in fan power. Pressure drops less than or equal to twice the baseline value were acceptable for candidate designs.

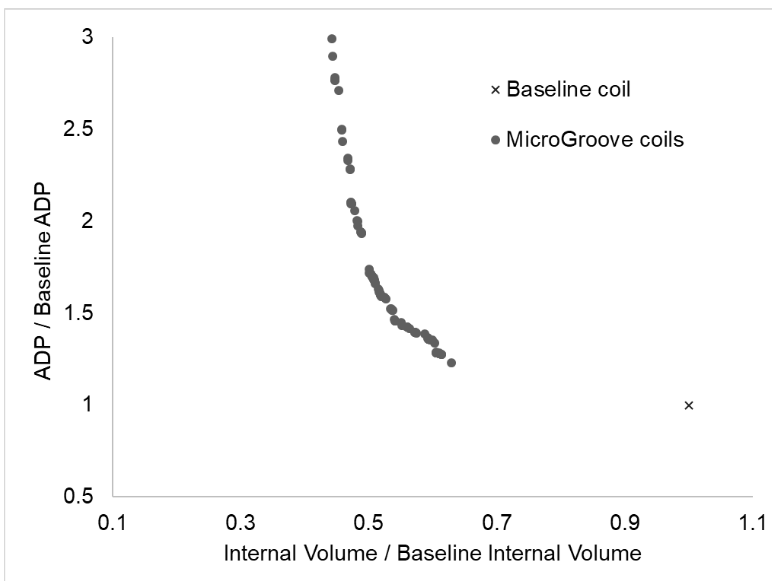


Figure 2: Pareto curve of MicroGroove designs, plotting normalized internal volume versus normalized airside pressure drop (ADP)

In summary, several new condenser designs were identified with significant potential to reduce internal volume while maintaining performance, and thereby reducing total system charge. The reduced footprint of the coils allows for a smaller enclosure.

5. PROTOTYPE CONSTRUCTION AND TESTING

As previously mentioned, the existing condenser is a fin-and-tube heat exchanger with parallel configuration (*i.e.*, freezer circuit at the top, and refrigerator circuit at the bottom). The proposed design differs from the baseline in terms of the circuitry configuration and tube diameter.

Figure 3 compares the tube circuitry of baseline heat exchanger with the proposed new design. Table 2 gives the tube and fin spacing for both designs. Note that the number of tubes and the overall size of the heat exchanger is the same for the baseline design and the new design. In terms of total material savings, the prototype of the proposed design was 21 percent lighter than the baseline.

A hot water calorimeter was used to assess the airside heat transfer performance of the proposed heat exchanger design and compare it to the baseline design. Temperature measurements were made to determine the Logarithmic Mean Temperature Difference (LMTD) between the airflow across the heat exchanger and the hot water flowing through the tubes.

Table 2: Heat exchanger configurations

	Baseline	**Proposed Design**
Tube Diameter (mm)	6.5	5
Tubes per Bank	8	8
Tube Banks	2	2
Horizontal Spacing (mm)	22.75	22.75
Vertical Spacing (mm)	26	26
Tube Length (in.)	17	17
Fin Type	Flat	Flat
Fin Density (fins per inch)	7	7
Fin Thickness	0.19	0.19

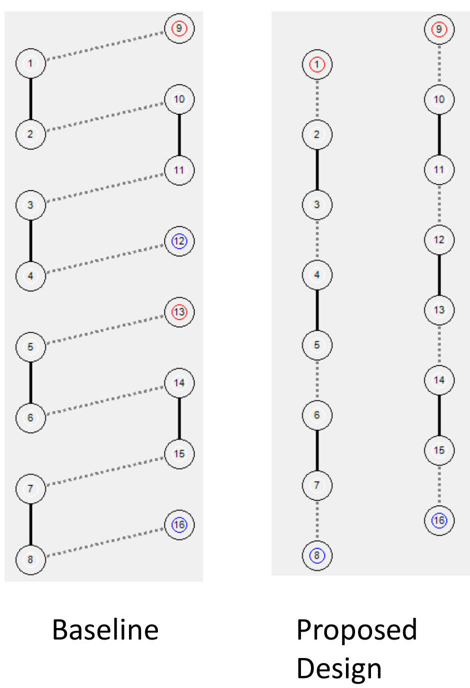


Figure 3: Condenser circuitry for baseline and proposed design

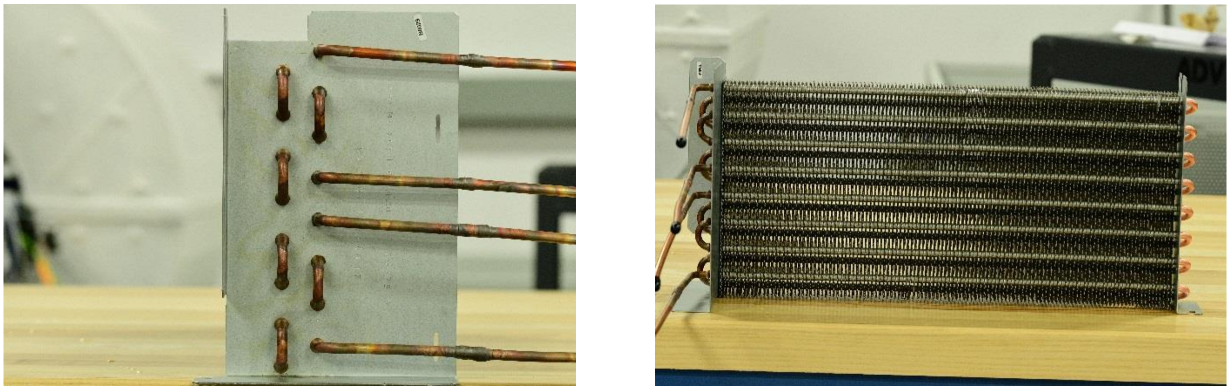


Figure 4: Baseline design

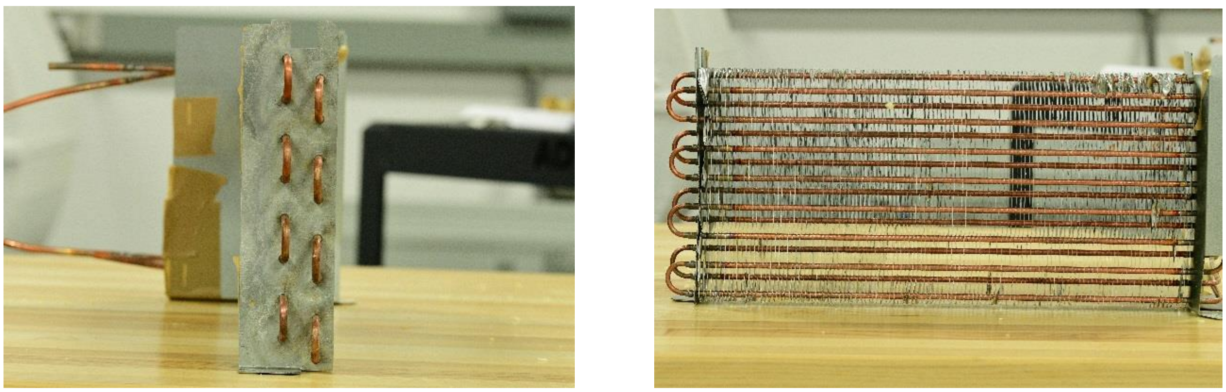


Figure 5: Prototype of proposed design

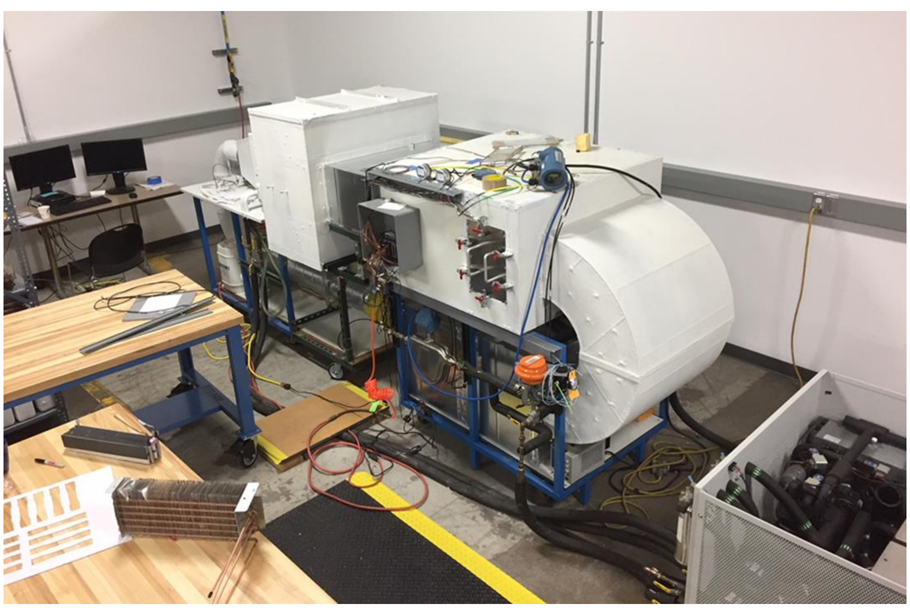


Figure 6: Hot water calorimeter

The heat loss by the water and heat absorbed by the air were also measured. The average value of heat loss (Q) and the LMTD were used to calculate the overall conductance (UA) of the coil. See Equation 1 and Equation 2, where ΔT_A and ΔT_B are the temperature differences between the air and water at the ends, A and B, respectively. The airside pressure drop was also measured.

$$Q = UA \times LMTD \tag{1}$$

$$LMTD = \frac{\Delta T_A - \Delta T_b}{ln\left(\frac{\Delta T_A}{\Delta T_B}\right)} \tag{2}$$

Figure 6 shows the hot water calorimeter that was used to measure the heat transfer and airside pressure drops for both heat exchangers, including the proposed design and the baseline. Due to calorimeter limitations, just one circuit of each heat exchanger was evaluated. The heat transfer value used in the UA calculation is the average between the heat lost by the water and the heat gained by the air as it flows across the heat exchanger.

The metric to evaluate the airside heat transfer performance was the overall heat conductance (UA) as a function of the airside pressure drop. Figure 7 shows that UA is slightly higher (2.33 percent higher) for the proposed design compared to the baseline for an airside pressure drop of 0.033 inches of H_2O, which was the airside pressure drop of the original baseline design. Hence, for a given overall conductance the proposed design would require slightly less fan power, or it would provide more heat transfer for a given fan power. This result differs from the simulation result because the smaller face area and wavy fins of the simulation led to a higher pressure drop. The prototype of the proposed design has flat fins although the ultimate design would be for wavy fins.

Comparing smooth wavy fins with flat fins, the built-in correlations of the coil design software showed an increase of 19.2 percent for the heat transfer as the airside pressure drop increased by 38.4 percent. In other words, heat transfer could be increased at the expense of an increase of airside pressure drop. Also, the prototype of the proposed design was lacking a fin collar; the proposed design would have an additional improvement in overall conductance when fabricated in a manner similar to the current baseline heat exchanger.

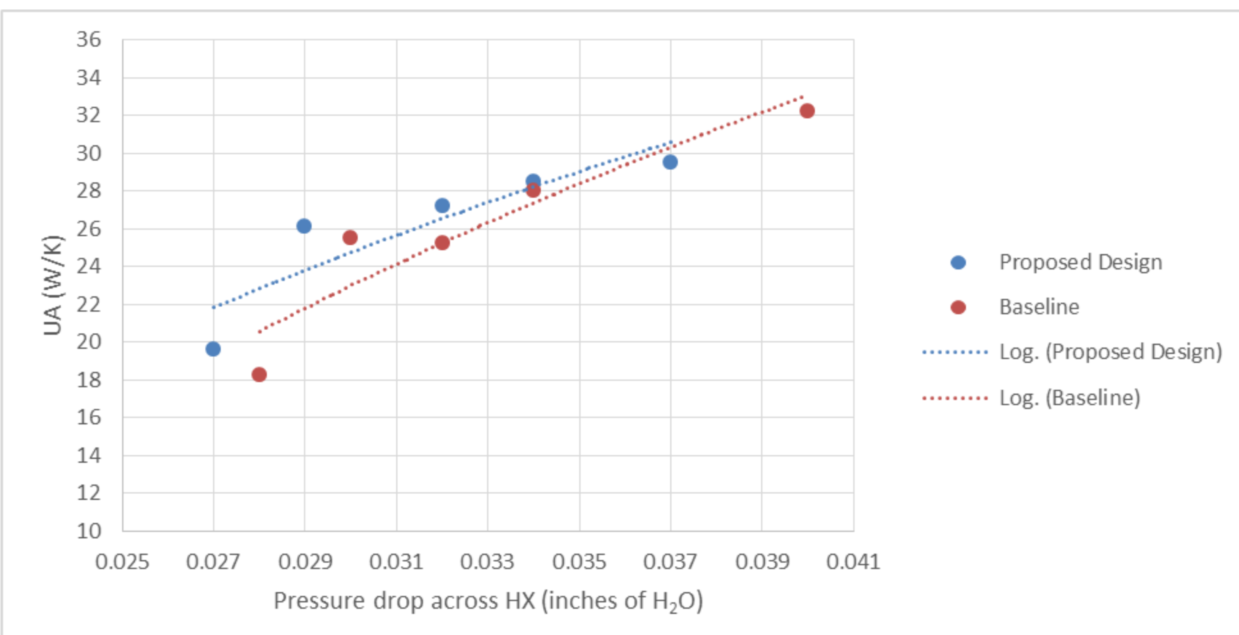


Figure 7: Overall conductance versus pressure drop

6. PRODUCTION CONSIDERATIONS

The supply chain for the manufacture of high efficiency MicroGroove coils is well established. OEMs today have many options available as they strive to make the transition from conventional to smaller-diameter copper tubes. It should be mentioned that a new pressure expansion method is now available to expand copper tubes into aluminum fins. Testing shows that the new method has significant improvements over traditional mechanical expansion processes. Pressure expanded coils have a more consistent expanded diameter than mechanically expanded coils, better tube-fin contact, and less damage of internal tube enhancements for pressure expanded tubes and coils (Tetzloff *et al.*, 2016).

7. SUMMARY AND CONCLUSIONS

The phasedown of HFCs by such regulations as the F-Gas regulations of the European Union, the SNAP process of the EPA and the Kigali Amendment to the Montreal Protocol contributed to the interest in low-GWP hydrocarbons such as propane and isobutane. Yet it is the attractive physical properties of propane and isobutane that have led to their adoption, once the regulatory hurdles in favor of hydrocarbons and against HFCs were in place.

The excellent thermodynamic properties of hydrocarbons such as R290 and R600a and the fact that they are readily available and affordable are important factors. Refrigeration systems that use ecofriendly hydrocarbons as refrigerants generally also deliver high-efficiency and high-performance.

Although these hydrocarbons are classified as A3 flammable refrigerants, it is safe to use them when proper protocols are followed. In the case of isobutane, the charge limit remains at 57 grams in the U.S. although ongoing legislation is aimed at increasing this charge limit. This low-charge limit motivated evaluation of optimized heat exchangers that could deliver the needed cooling capacity using smaller diameter copper tubes.

This study demonstrated the simulation and design of new heat exchangers that can maintain the heat transfer performance of the baseline and allow for lower refrigerant charge in a smaller, lighter envelope. The proposed design has 41 percent less internal volume than the baseline. A significant reduction in refrigerant charge is achieved when this heat exchanger is used as a condenser in the refrigerator.

The proposed design to replace the baseline has a slight improvement in heat transfer performance at a given airside pressure drop, e.g., an improvement of 2.33 percent at 0.033 inches H2O of airside pressure drop. An even greater increase of heat transfer performance is expected once the proposed design is production-tooled with fins that include collars. Besides equal or better performance and reduced refrigerant charge, the prototype was 21 percent lighter than the baseline.

NOMENCLATURE

ADP	Airside pressure drop	(inches of H_2O)
LMTD	Logarithmic mean temperature difference	(K)
Q	Average value of heat loss	(W)
UA	Overall conductance	(W/K)
ΔT_A	Temperature Difference at A	(K)
ΔT_B	Temperature Difference at B	(K)

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