# Evaluation of Contact Resistance and Fin Effectiveness of Enhanced, Brazed "Dogbone" Fin and Serpentine Tube Heat Exchangers for Air Conditioning and Heat Pump Applications

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## ABSTRACT

The energy consumption due to air conditioning is projected to increase by 1.3 and 4.5 times by 2050 compared to 2010 for members and non-members of the Organization of Economic Coordination and Development (OECD), respectively. Greenhouse Gas (GHG) emissions must be carefully managed to minimize their hazardous contribution to climate change. Air conditioners (AC) and heat pumps (HP) contribute with direct GHG emissions through systemic refrigerant leakage, which consequently degrades the system's performance and therefore increases its indirect GHG emissions as well. To address this issue, the goals of new systems include minimizing leakage risk, using low global warming potential (GWP) refrigerants such as natural refrigerants, and increasing system efficiency. This paper presents part of the efforts in the development of a brazed "dogbone" fin to continuous serpentine tube heat exchanger (HX) for AC/HP applications. The main advantages of such technology are: 1) The potential to reduce 90% or more of the tube return-bends brazed joints compared to conventional tube-fin heat exchangers, thus minimizing leakage vulnerability, enabling the use of natural refrigerants like Propane and Ammonia that carry other concerning risks of flammability and toxicity ; and 2) The brazed fin to the tube joints should yield very low contact resistance. In order to quantify and estimate the impacts of contact resistance and fin effectiveness, a numerical-experimental study is investigated. The novel enhanced brazed "dog-bone" fin is compared to an equivalent pressure expanded fin. The results showed that the expanded fin has consistently greater effective heat transfer but greater contact resistance as well. Furthermore, the brazed fins have up to 20% lower pressure drop. The trade-off comparison shows that the thermal-hydraulic ratio between the two fins is equivalent for air velocities below 2m/s; for greater velocities, however, the expanded fin is favored since it has less fin efficiency penalty. For HVAC&R applications, the air velocities are typically low which makes the proposed technology competitive and a viable option in the pursuit of refrigerant leakage reduction.

*Keywords: tube-fin coil, thermal resistance, experimental, CFD, heat pump*

## 1. INTRODUCTION

Refrigerant leakage in heat pumps (HP) and air conditioners (AC) has great impact on direct and indirect greenhouse gas (GHG) emissions. It is estimated an annual leakage rate on average of 3.5% of the original charge (Eunomia Research & Consulting Ltd and the Centre for Air Conditioning and Refrigeration Research, 2014). Units not regularly serviced will progressively lose even more charge and therefore lose performance and efficiency. The literature has reported findings that 10 to 20% charge loss can result in 13% less cooling capacity and 7.5% loss in COP for air-conditioners (Kim & Braun, 2010), which consequently result in an increase in indirect greenhouse gas GHG emissions. The roadmap of low Global Warming Potential (GWP) refrigerants leads to very low-GWP refrigerants, including natural refrigerants (NR) for reduction of direct GHG emissions. Existing concerns of NR's, in particular Ammonia and Propane, include toxicity and flammability, hence imposing an additional negative impact from leakage.

From an equipment perspective, one way to reduce leakage potential is to reduce the number of vulnerable parts, i.e., the brazed joints. In typical AC's and HP's, the majority of the brazed joints are located in the return bends of tube-fin heat exchangers (HX's). Because the hairpin tubes are inserted into the open round holes on the fins, they need to be brazed to a return bend at the other end to connect to other tubes; thus for every straight length of tube there is one brazed joint associated with it. An alternative concept used in refrigeration is the continuous serpentine tubes and "dog-bone" fin heat exchangers, which can have more than 90% brazed joint reduction since there are no separate return "U" bends. The two main challenges with this are the fin surface area reduction but most importantly, the contact resistance between tube and fin. Unlike conventional tube-fin HX's, where the tubes are expanded towards the fin collars creating a tight fitting, the serpentine tubes are inserted into the fins as a whole. One way to improve the contact resistance is to braze the fins onto the tube surface.

An experimental study on contact resistance between conventional and brazed serpentine tube and "dog-bone" fins showed that the latter exhibited 50% lower airside thermal resistance (ElSherbini, Jacobi, & Hrnjak, 2003). Under frost conditions, the authors of this same study reported that the difference in airside thermal resistance between the two is significantly reduced possibly due to ice filling the gap between tube and fin. Predicting contact resistance is difficult since it is subject to many factors such as tube diameter / thickness, fin collar size, material, type of joint (i.e. press-fit, mechanically expanded, pressure expanded, or brazed), tolerances, and others. A numerical model using finite element analysis (FEA) was presented to predict the contact uniformity of mechanically expanded tubes and were correlated with test data (Tang, Li, Peng, & Du, 2010). In this study, they infer that contact resistance is inversely proportional to tube diameter, a finding reported in other studies as well. An experimental-analytical study applied to various fin types and mechanically expanded tubes showed that for 9.52mm tubes, the contact resistance accounts for 6-19% of the total airside thermal resistance (Jeong, Kim, Youn, & Kim, A study on the correlation between the thermal contact conductance and effective factors in fin–tube heat exchangers with 9.52 mm tube, 2004). This same research group later published an identical study on 7mm tubes and found that the contact resistance accounted for 15-36% of the total airside thermal resistance (Jeong, Kim, & Youn, A study on the thermal contact conductance in fin–tube heat exchangers with 7 mm tube, 2006). More recently, a numerical-experimental study using CFD simulations to determine the contact resistance on an oval tube and flat fin HX's was presented (Taler & Ocłon, 2014).

The present paper presents an analytical-experimental study - similar to those abovementioned - used to determine the differences in contact resistance and overall thermal-hydraulic characteristics between the novel louvered "dog-bone" fin brazed to serpentine tubes, and an equivalent pressure expanded tube-fin HX. Tests were carried out to determine the total airside thermal resistances, while CFD simulations were used to estimate isolated convective thermal resistance; the contact resistance was estimated by the difference of the two. The subjects were all aluminum HX's with a single bank of 7mm diameter tubes and 21FPI.

## 2. MATERIALS & METHODS

### 2.1 Tube-Fin Designs: Conventional Pressure Expanded vs. "Dog-bone" Brazed

A novel "dog-bone" fin design with enhancing louvers was developed with the objective of being competitive to conventional full-collared fins, to enable the use of continuous serpentine tubes with reduced brazed joints. Such fins are brazed to the tubes to reduce or, at best, eliminate the contact resistance (Figure 1). Two additional considerations include the reduced fin effectiveness, since the contact area between tube and fin is reduced, as well as loss of overall heat transfer surface area which is 13% less compared to the pressure expanded HX. To determine its suitability as a replacement for conventional tube-fin HX's, considering its apparent disadvantages, its thermal-hydraulic characteristics are compared to those of an equivalent fin (same enhancement pattern) with full collars and pressure expanded tubes (Figure 2).

### 2.2 Analysis Approach

CFD simulations for airside characterization were carried out to determine effective heat transfer coefficient and pressure drop of these unique fin designs. The simulations consist of a constant tube wall temperature, i.e. disregarding all thermal resistances on the airside except convective resistance (eq. (1)). The reader is referred to (Bacellar, Aute, & Radermacher, 2016) for additional information on the CFD modeling approach for it is not in the scope of this manuscript. From test data using hot water it is possible to calculates the total thermal resistance (eq. (2)), and the total airside thermal resistance is retrieved by subtracting the water side and tube wall terms. The water heat transfer coefficient used in eq. (4) is estimated using Gnielinski correlation (1975). The convective term of the airside resistance is determined using the effective heat transfer coefficient from eq. (1) scaled by the total prototype surface area (eq. (5)). Finally, all metrics of interest are curve fitted as a power function of air frontal velocity (eq. (6)) in order to establish a comprehensive assessment of the performance of each fin in a wide range of airflow conditions.

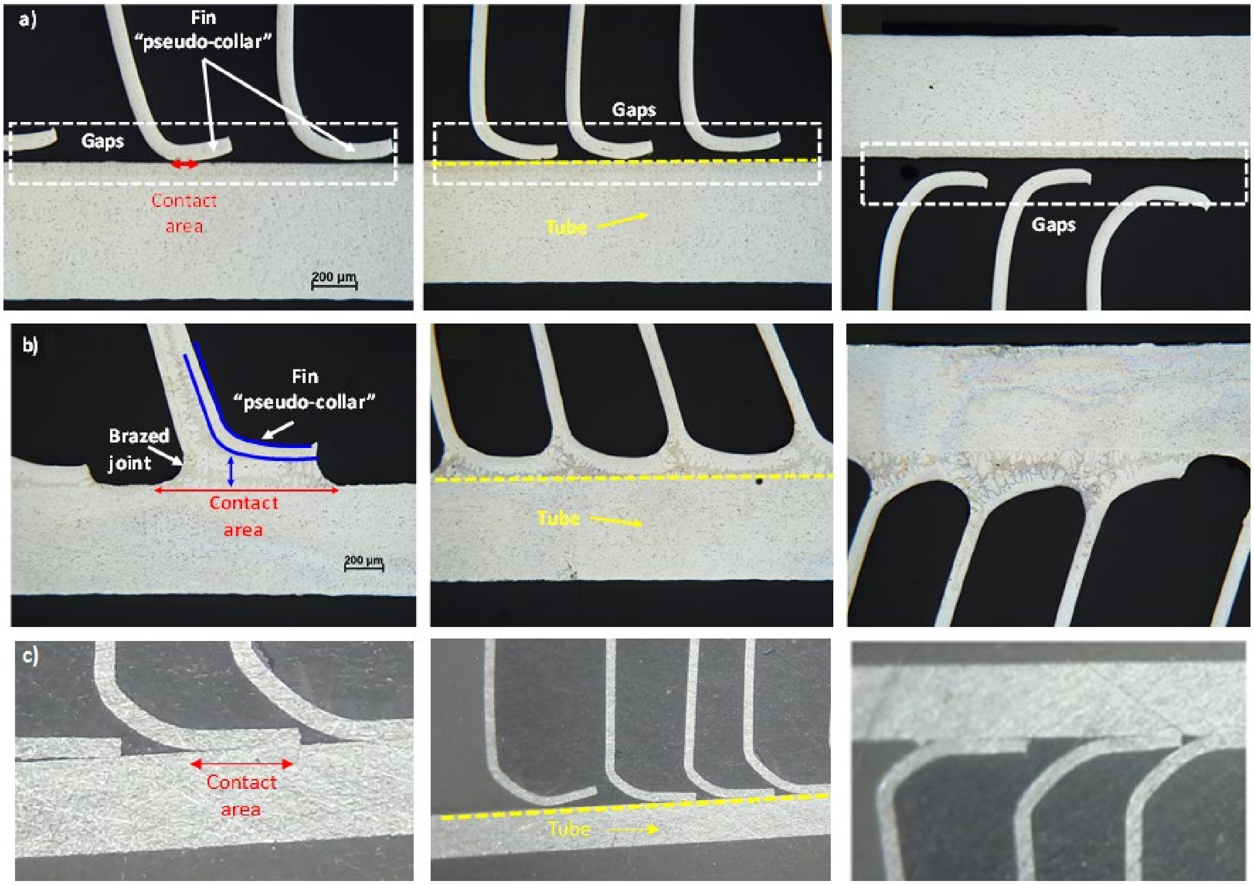



**Figure 1:** Microsection views: a) conventional "dog-bone" fin; b) brazed "dog-bone" fin; c) expanded fin.


**Figure 2:** HX Designs: a) Expanded; b) Brazed; c) Microsection view of the louvers.

$$\left(\eta_o h_{air}\right)_{convective} = \frac{\dot{Q}_{CFD}}{A_{o,CFD}} \cdot \ln\left(\frac{T_w - T_{air,in}}{T_w - T_{air,out}}\right) \Big/ \left(T_{air,out} - T_{air,in}\right) \tag{1}$$

$$R_T = \left(\Delta T_{lm,test} \Big/ \dot{Q}_{test}\right) = R_{air,total} + R_{tube\_wall} + R_{water} \tag{2}$$

$$R_{air,total} = R_T - \left(R_{tube\_wall} + R_{water}\right) = R_{convective} + R_{contact} \tag{3}$$

$$R_{water} = \left(h_{Gnielinski} A\right)_{water}^{-1} ; R_{tube\_wall} = \ln\left(\frac{D_o}{D_i}\right) \Big/ \left(2\pi L k\right) \tag{4}$$

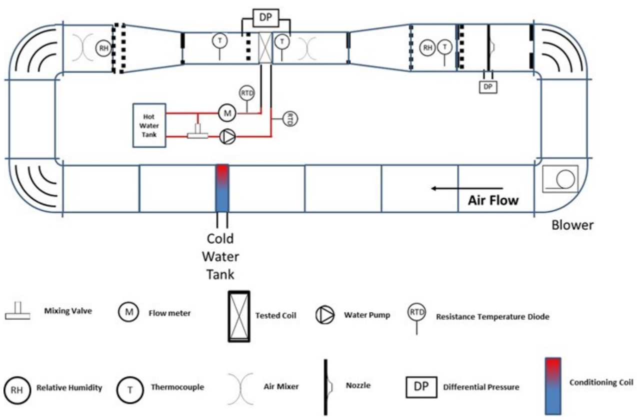
$$R_{convective} = \left[\left(\eta_o \cdot h_{air}\right)_{convective} \cdot A_o\right]^{-1} \tag{5}$$

$$\phi(u) = a \cdot u^b \tag{6}$$

## 2.3 Experimental Setup

The tests were conducted in a temperature and humidity-controlled wind tunnel (Figure 3) while using hot water as the in-tube working fluid. Dry-bulb temperatures were measured using a 9-point and a 25-point thermocouple (TC) grid at inlet and outlet of the coil, respectively. Additional instrumentation included two relative humidity sensors placed near to their respective TC grids. Static pressure difference across the coil and across a calibrated nozzle matrix were measured for pressure drop and airflow rate assessment.

In order to fully characterize the thermal-hydraulic performance of each HX, a 9-point test matrix with a combination of three water flow rates and three air flow rates was tested (Table 1). The flow rates were selected as such to cover a range of interest of air velocities and to ensure fully developed turbulent flow on the water side, while satisfying pump capacity and aiming to maintain high capacitance rate ratio (C*) to avoid temperature pinching.




| Parameter | Type | Range | Systematic Uncertainty |
|---|---|---|---|
| Water flow rate | Coriolis flow meter | 0 to 700 g/s | 0.5% of reading |
| Water inlet & outlet temperature | RTD | -100 to 400°C | ±0.03 K |
| Air inlet temperature | 9-point thermocouple | -200 to 200°C | ±0.17 K |
| Air outlet temperature | 25-point thermocouple | -200 to 200°C | ±0.1 K |
| Coil air-side pressure drop | Differential pressure | 1" WC | 0.25% FS |
| Relative humidity (RH) | RH sensor | 0-95% | ±2% |
| Nozzle air-side pressure drop | Differential pressure | 5" WC | 0.25% FS |

**Figure 3:** Setup Diagram.

**Table 1:** Test Matrix.

| Test | $T_{water}$ | Water Flow Rate | $Re_{water}$ | $T_{air}$ | Air Flow Rate | C* |
|---|---|---|---|---|---|---|
| # | °C | g/s | - | °C | CFM (m/s) | - |
| #1 | 50 | 100 | 14979 | 16 | 500 (2.2) | 0.70 |
| #2 | | 150 | 22469 | | | 0.46 |
| #3 | | 200 | 29958 | | | 0.35 |
| #4 | | 100 | 14979 | | 1000 (4.5) | 0.72 |
| #5 | | 150 | 22469 | | | 0.93 |
| #6 | | 200 | 29958 | | | 0.70 |
| #7 | | 100 | 14979 | | 1500 (7.0) | 0.48 |
| #8 | | 150 | 22469 | | | 0.72 |
| #9 | | 200 | 29958 | | | 0.96 |

# 3. RESULTS

Table 2 and Table 3 summarize the test results for the expanded and brazed HX, respectively. HX tests exhibited very good energy balances; no greater than 2.7% due to the strategic choice of the test matrix points that yield low impact from measurement uncertainties. The airside pressure drop predicted by the CFD simulations exhibited very good agreement with the test data, in particular for lower air velocities as illustrated in Figure 4. These results are strong evidence that the models adequately capture the physical phenomena and are hence reliable for further analyses. As expected, however, the airside effective heat transfer coefficient predictions from CFD simulations are greater than that obtained from test data (Figure 5) since they are not capturing any parasitic resistances which will all be treated herein as contact resistance. In both simulation and tests, the expanded fin exhibited greater thermal performance than the brazed fin. This difference is due to the reduced fin efficiency (and effectiveness) on the latter. The gap between the two is considerably smaller in the actual HX's compared to CFD, suggesting that the brazed fin has much lower contact resistance as compared to the pressure expanded one. On Figure 6, the airside thermal resistance is illustrated for a wide range of velocities and while the expanded fin consistently shows lower total thermal resistance, the contribution of the contact resistance appears to be significantly greater.

**Table 2:** Summary Test Results for Expanded HX.

| Metric | Unit | #1 | #2 | #3 | #4 | #5 | #6 | #7 | #8 | #9 |
|---|---|---|---|---|---|---|---|---|---|---|
| Energy Balance | % | 1.4% | 1.2% | 1.6% | 2.0% | 2.2% | 2.7% | 1.9% | 1.9% | 1.8% |
| Heat Load | W | 3655.8 | 4070.7 | 4355.1 | 4599.3 | 5302.7 | 5800.6 | 5062.7 | 5936.6 | 6458.0 |
| $\Delta T_{water}$ | K | 8.90 | 6.62 | 5.29 | 10.93 | 8.38 | 6.84 | 11.99 | 9.37 | 7.65 |
| $\Delta T_{ml}$ | K | 23.10 | 23.26 | 23.39 | 24.53 | 25.16 | 25.67 | 25.02 | 25.94 | 26.38 |
| UA | W/K | 160.92 | 178.36 | 188.93 | 186.13 | 208.81 | 222.80 | 200.27 | 226.32 | 242.29 |
| $\eta_o * h_{air}$ | W/m².K | 60.39 | 63.56 | 65.35 | 73.45 | 77.57 | 79.91 | 81.38 | 86.19 | 88.82 |
| $R_{air,total} / R_T$ | % | 77.3% | 81.4% | 83.9% | 73.5% | 78.1% | 80.9% | 71.4% | 76.2% | 79.1% |
| $\Delta P_{air}$ | Pa | 43.5 | 43.6 | 43.6 | 131.8 | 132.0 | 131.9 | 260.9 | 261.6 | 261.7 |

**Table 3:** Summary Test Results for Brazed HX.

| Metric | Unit | #1 | #2 | #3 | #4 | #5 | #6 | #7 | #8 | #9 |
|---|---|---|---|---|---|---|---|---|---|---|
| Energy Balance | % | -1.0% | -0.8% | -0.4% | 0.6% | 0.7% | 1.1% | 0.4% | 0.2% | 0.4% |
| Heat Load | W | 3327.8 | 3663.5 | 3866.7 | 4242.5 | 4782.8 | 5141.3 | 4653.5 | 5336.8 | 5770.0 |
| $\Delta T_{water}$ | K | 8.37 | 6.13 | 4.84 | 10.26 | 7.71 | 6.19 | 11.22 | 8.59 | 6.95 |
| $\Delta T_{ml}$ | K | 23.90 | 24.27 | 24.48 | 25.26 | 25.99 | 26.49 | 25.56 | 26.65 | 27.23 |
| UA | W/K | 146.29 | 158.40 | 165.18 | 169.70 | 185.87 | 195.34 | 183.36 | 202.09 | 213.3 |
| $\eta_o * h_{air}$ | W/m².K | 56.05 | 57.99 | 58.94 | 67.69 | 70.31 | 71.71 | 74.96 | 77.99 | 79.69 |
| $R_{air,total} / R_T$ | % | 81.0% | 84.8% | 87.0% | 77.9% | 82.1% | 84.6% | 76.0% | 80.5% | 83.1% |
| $\Delta P_{air}$ | Pa | 33.5 | 33.6 | 33.6 | 104.7 | 104.8 | 104.9 | 212.0 | 212.3 | 212.4 |

Based on the results found in this study, it is estimated that the contribution of the contact resistance to the total airside resistance varies from 20-35% for the expanded coil (Figure 7a), which is of the same order reported by Jeong et. al. (Jeong, Kim, & Youn, A study on the thermal contact conductance in fin–tube heat exchangers with 7 mm tube, 2006) for mechanically expanded 7mm tubes. For the brazed fins, the contact resistance contribution varies from 15-20%, which although is not entirely eliminated, it is a significant improvement. The existing contact resistance in the brazed joints can be due to several joints not fully or uniformly brazed, which means there's opportunity to reduce this resistance even further with tighter tolerances and improved manufacturing techniques. The estimates presented above are merely for reference, since the deviations from CFD predictions are not included, and they typically overpredict heat transfer. Considering, however, the good fit on the pressure drop prediction, it is assumed herein that the CFD heat transfer coefficient is the true convective heat transfer coefficient, and the difference is due to contact resistance only. Another important difference is the pressure drop: the brazed "dog-bone" fin, for having reduced surface area, imposes less resistance, thus resulting in 5-20% lower pressure drop than its expanded counterpart. The thermal-hydraulic ratio – defined as thermal conductance divided by pressure drop – is very similar for both designs. At lower velocities (< 2m/s) the brazed fin has equivalent or greater thermal-hydraulic ratio (Figure 7b). It degrades faster with air velocity because fin efficiency penalty increases accordingly. For HVAC&R applications, the air velocities are typically low (0.5-3m/s) which makes this fin design competitive.

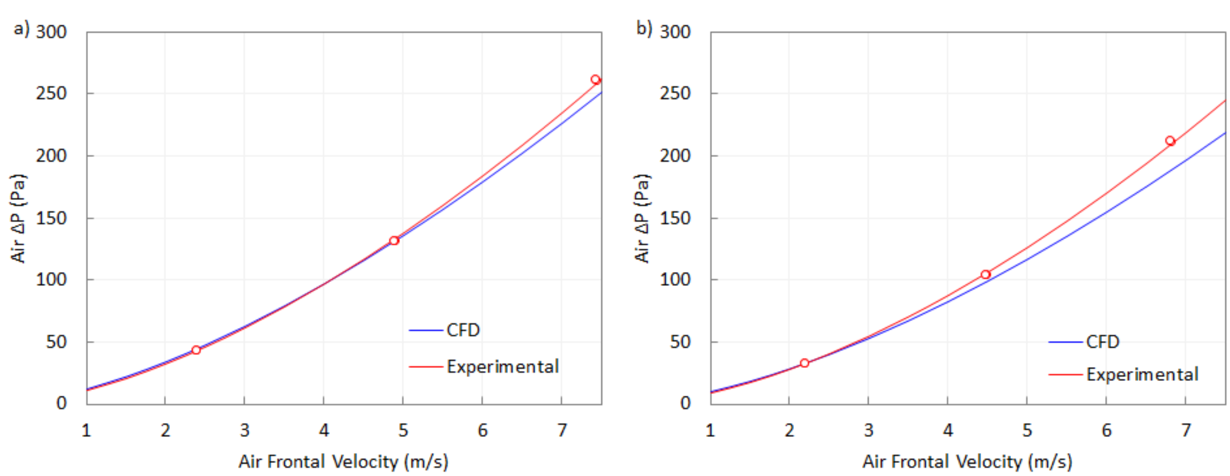



**Figure 4:** Airside Pressure Drop: a) Expanded ; b) Brazed.

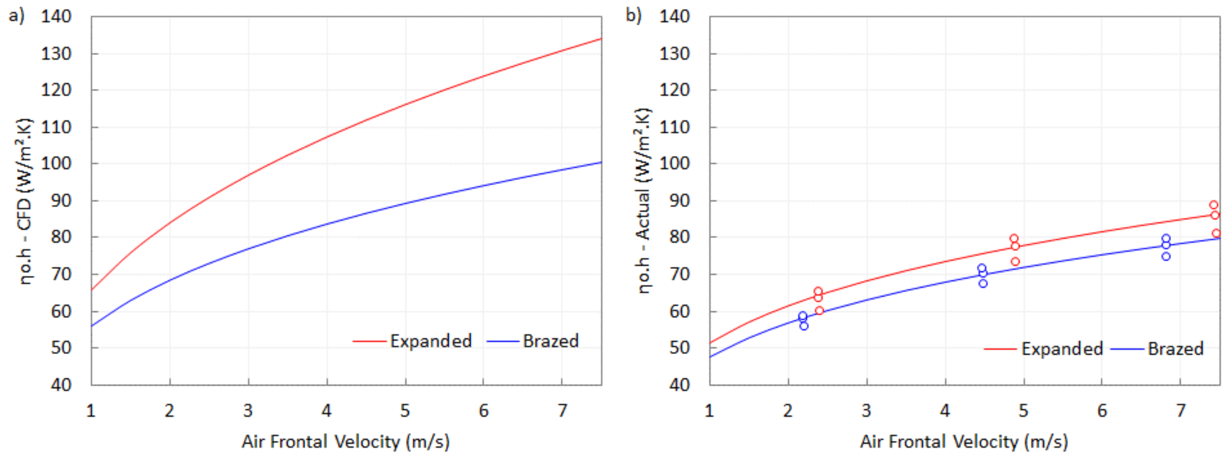



**Figure 5:** Airside Effective Heat Transfer Coefficient: a) Simulation; b) Experimental.

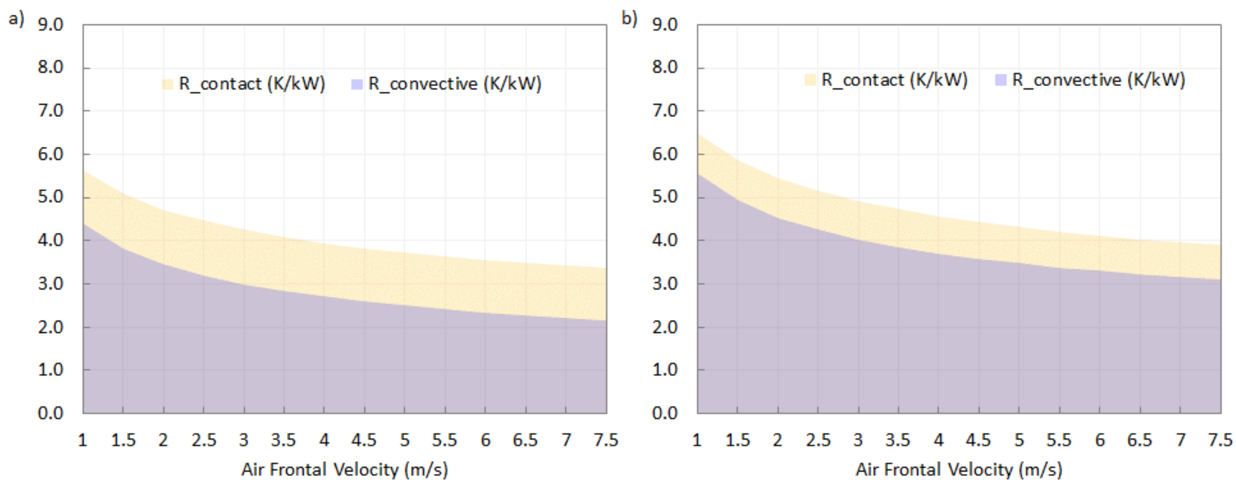



**Figure 6:** Airside Thermal Resistance: a) Expanded; b) Brazed.

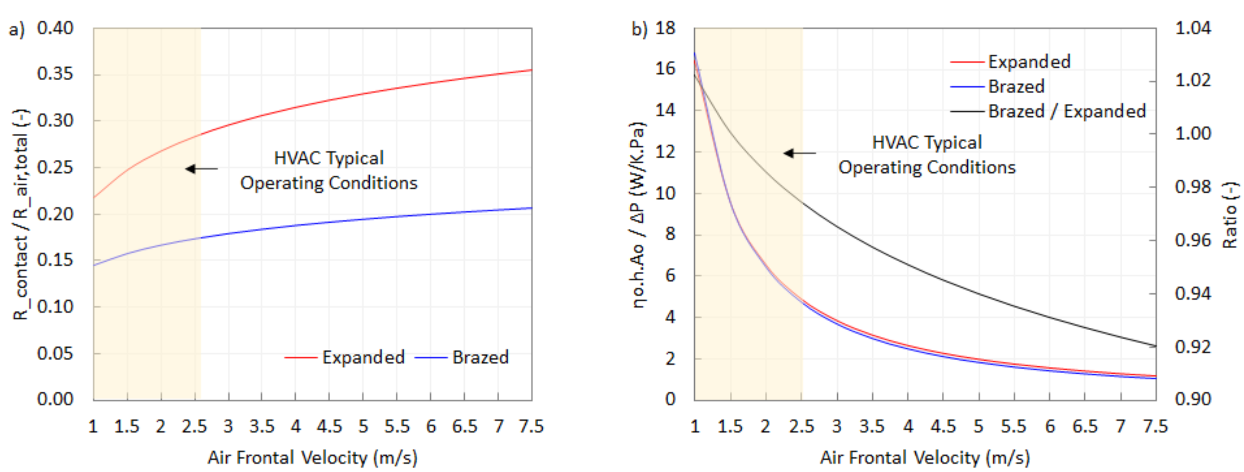



**Figure 7:** Fin Performance: a) Thermal Resistance Ratio; b) Thermal-Hydraulic Ratio.

## 4. CONCLUSIONS

The present work introduced the concept of serpentine tube and enhanced "dog-bone" fin HX's for HP and AC applications, with the goal of reducing refrigerant leakage vulnerability through elimination of brazed joints on tube return bends. The challenges with such a fin concept are contact resistance, reduced surface area and reduced fin efficiency. The first is addressed by brazing the fins to the tubes, thus bridging the typical gaps found in conventional serpentine tube HX's. In order to quantify and estimate the impacts of contact resistance and fin effectiveness, a numerical-experimental study is investigated. The novel enhanced brazed "dog-bone" fin is compared to an equivalent pressure expanded fin. CFD simulations were carried out to provide an estimate of the airside convective heat transfer, isolated from other factors, while the HX tests provided the actual airside characteristics, including contact resistance. The latter is estimated through the difference between the two. The CFD predictions on airside pressure drop exhibited very good agreement with test data thus providing strong evidence that the models captured the physical phenomena properly and are reliable for further analyses. The results showed that the expanded fin has consistently greater effective heat transfer, since the brazed fin has lower efficiency due to the reduced contact area between tube and fin. The difference between numerical simulations and test data indicate that the expanded fin has greater contact resistance. For a wide range of air velocities, the latter contributes to 20-35% of the total thermal resistance, while the brazed fin exhibited contact resistance contributions below 20% throughout. The existing contact resistance in the brazed fins are possibly due to several joints not fully and/or uniformly brazed. Tighter tolerances and improved manufacturing techniques may increase the quality of such joints and reduce the contact resistance even further. The pressure drop is up to 20% lower on the brazed fin due to reduced surface area which gives it an advantage. The trade-off comparison shows that the thermal-hydraulic ratio between the two fins is equivalent for air velocities below 2m/s; for greater velocities, however, the expanded fin is favored since it has less fin efficiency penalty. For HVAC&R applications, the air velocities are typically low, which makes the proposed technology competitive and a viable option in the pursuit of refrigerant leakage reduction.

## NOMENCLATURE

| | | |
|---|---|---|
| A | Surface Area | ($m^2$) |
| C* | Heat Capacitance Ratio | (-) |
| $D_i$ | Tube Inner Diameter | (m) |
| $D_o$ | Tube Outer Diameter | (m) |
| FPI | Fins Per Inch | (1/in) |
| h | Convective Heat Tranfer Coefficient | ($W/m^2.K$) |
| k | Thermal Conductivity | (W/m.K) |
| L | Tube Length | (m) |
| P | Pressure | (Pa) |
| Q | Heat Load | (W) |
| R | Thermal Resistance | (K/kW) |
| Re | Reynolds Number | (-) |
| T | Temperature | (K) |
| u | Air Velocity | (m/s) |
| UA | Global Thermal Conductance | (W/K) |
| $\Delta P$ | Pressure Drop | (Pa) |
| $\Delta T$ | Temperature Difference | (K) |
| $\Delta T_{ml}$ | Log-Min Temperature Difference | (K) |
| $\eta_o$ | Fin Effectiveness | (-) |
| $\eta_o$*h | Effective Heat Transfer Coefficient | ($W/m^2.K$) |
| $\phi$ | Generic Metric | (-) |

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