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Manufacturing & Testing of Air-to-Refrigerant Heat Exchangers Based on 0.8mm Diameter Tubes

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ABSTRACT

Recent research has shown that moving towards small tubes with diameters less than 5 mm brings significant advantages in air-to-refrigerant heat exchangers (HX). These advantages include compactness, high airside heat transfer coefficient, reduced refrigerant charge and reduced weight. Tubes with outer diameter (OD) of 0.8 mm were shown to provide very high heat transfer coefficient resulting in heat exchanger with a high degree of compactness. However, using such small tube diameter comes with manufacturing and field performance challenges. Those include the management of many hundreds of small tubes, developing a method to ensure that a specific gap between the tubes at any given point is kept, maintaining the desired geometrical shape of the complete heat exchanger and sealing the tube to header joints, which is the biggest challenge. Field challenges include areas such as fouling and water drainage. This paper presents the lessons learned from manufacturing and testing of air-to-refrigerant heat exchangers that use such 0.8 mm OD tubes. Three prototypes were manufactured, the first one was made of stainless steel (SSHX), the second one was made of copper (CuHX) and the third one was a larger version of the second prototype (CuHX-10kW). The first and second HX were designed to have the same pressure drop and heat transfer performance. These prototypes were fabricated using different manufacturing techniques and lessons learned were discussed. One of the observations is that the uncertainty in tube spacing due to manufacturing has a significant impact on the performance. This points to the need for robust optimization methods in design of such heat exchangers for which the manufacturing technologies are still being developed. Hydrostatic tests as well as performance tests in a closed loop wind tunnel were conducted. Test results for the three heat exchangers using air and water are discussed. The energy balances were within ±5% and uncertainties in averaged capacity were within ±3% for the three prototypes.

1. INTRODUCTION

Air-to-refrigerant heat exchanger is a key component in heating, ventilation, air conditioning and refrigeration (HVAC&R) systems; thus, a considerable amount of research was performed to improve its efficiency. Thermal analysis reveals that the dominant thermal resistance comes from airside, which is the limiting factor for further improving the overall heat transfer. Air thermal resistance is determined by heat transfer area and air heat transfer coefficient. A traditional solution is an increase in the heat transfer area by adding fins in a tube-and-fin heat exchanger.

Round tube-and-fin heat exchangers are widely used in HVAC&R systems and considerable research was performed to experimentally investigate the heat exchanger's heat transfer and hydraulic characteristics. Rich (1973) investigated 14 coils with tube size of 13.34 mm, with air velocity of 0.95~21 m/s; McQuiston (1978) tested 5 heat exchangers with tube size of 9.96 mm, with air velocity of 0.5~4 m/s; Kayansayan (1993) tested 10 different configurations with tube diameter of 9.52, 12.5 and 16.3 mm with air velocity of 0.5~10 m/s; and Wang tested 15 heat exchangers (1996) with diameter of 10.06 mm, with air velocity of 0.3~6.5 m/s and 22 samples with diameter of 7.3 mm within the same flow range (1997, 2000). However, none of the investigated tube diameters are less than 5 mm. Recent additional studies have shown great potential when tube diameter move below 5 mm (Bacellar, 2014, Paitoonsurikarn *et. al*., 2000, Saji *et. al.*, 2001, Kasagi *et al.*, 2003, Shikazono *et. al.*, 2007)

In the current study, an air-to-refrigerant heat exchanger utilizing tubes with an outer diameter of 0.8 mm was manufactured and experimentally investigated. Three prototypes were manufactured Two with a capacity of 1kW and one with 10kW. The three prototypes were tested as radiators using water and air. The heat transfer performance and hydraulic characteristics were discussed herein.

2. MANUFACTURING

2.1 1kW HX

0.8 mm OD tubes, either in copper or stainless steel, have inherent flexibility that was used in the assembly of the tubes into a heat exchanger. The thin tubes were easily flexed allowing insertion into the round manifolds. On one side, the tubes were passed through a number of spacers located near one of the manifolds, as shown in Figure 1. Enough spacers are used so that at a later stage, when they are spaced out, the tubes will be evenly spaced. To prevent the tubes from extending too deep into the manifolds, a spacer is positioned inside the manifold (not shown). Figure 2 shows the opposite manifold.

Once all tubes were in place, the spacers were moved to their desired locations, about 37mm apart, spacing the myriad of tubes in a nice evenly spaced orderly fashion as shown in Figure 3 below.

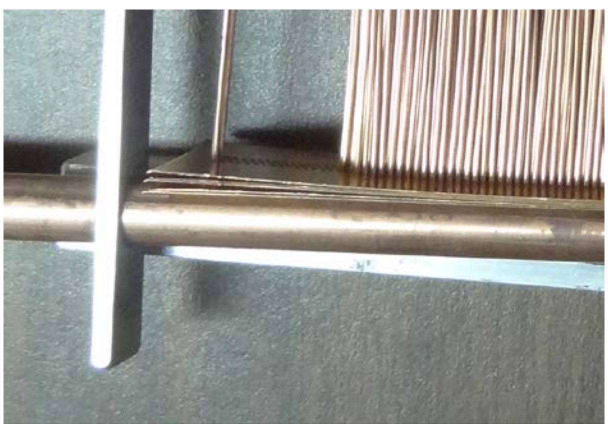

Figure 1: All spacers near one manifold

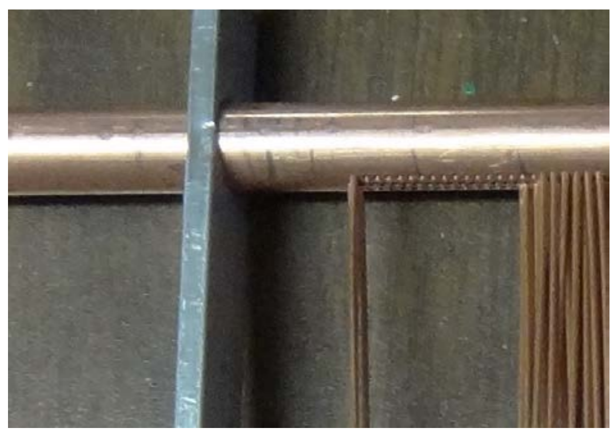

Figure 2: Tubes inserted into opposite manifold

Figure 3: Spacers spaced out

Figure 4: Soldered section between spacer and manifold

One spacer was left 3mm near each manifold. This section was then heated and filled with solder, as shown in Figure 4. The main function of this spacer is to prevent the solder from wicking up between the tubes into the core, due to the short distance between the tubes. Building the stainless steel HX used similar technique except epoxy adhesive replaced the solder. Figure 5 shows the copper spacer near the stainless manifold before epoxy was applied. Figure 6 shows this section under the microscope after the epoxy was applied and cured.

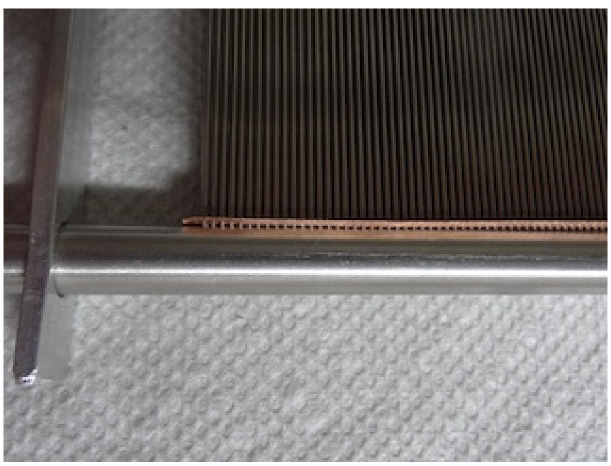

Figure 5: Stainless tubes and manifolds

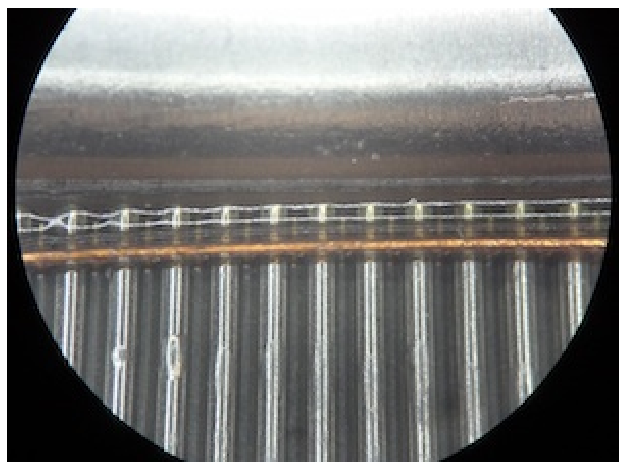

Figure 6: Cured epoxy view under a microscope

The preferred epoxy adhesive was a single-component Loctite 3981 which cures at 100°C in 35 minutes or at 150°C in 16 minutes, providing a cured Tensile Strength, of 62 N/mm^2 (ISO 527-3).

The HX dimensions are 170mm long (including the 9.52mm OD manifolds) x 155mm wide x 10mm deep.

2.2 10kW HX

To produce a 10kW HX, a modular approach was used. Four units were combined to make one heat exchanger, as shown in Figure 7. In addition, the round manifold method was dropped in favor of a new technique eliminating the need for drilling holes in the manifold. One spacer would be used as a header and mated to the square manifold, as shown in Figure 8.

Figure 7: Four modular units with spacers threaded

Figure 8: Outer spacer used as header – soldered to square manifold

The majority of the effort went into threading the tubes through the spacers. Each module consisted of 570 tubes. The total number of tubes was thus 2280. It is envisioned that this process can be automated for production. Each square brass manifolds has 4 slots, one for each module. The outer spacer acting as a header was soldered to the manifold as shown in 9 below.

A few tube leaks were discovered during leak testing. Those leaks were repaired using a soldering iron as shown in Figure 10. Out of 2280 tubes 450mm long, 4 tubes had leaks.

Figure 9: Completed 10kW heat exchanger with fittings

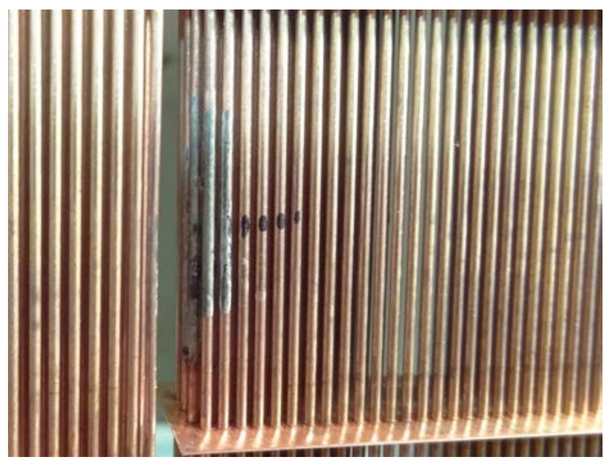

Figure 10: Repaired leak in tube.

The 10kW HX dimensions are 477mm long (including the manifolds) x 550mm wide x 16mm deep.

The gaps between the units were blocked during performance testing. Future heat exchangers can be built without the gaps by using longer spacers or, if outside the spacer manufacturing size limits, spacers can be overlapped.

3. TESTING RESTULTS

The SSHX and CuHX-10kW were tested in one facility and the CuHX was tested in another facility. The experimental setup consists of a closed-loop wind tunnel, a water loop system and a data acquisition system. The closed-loop wind tunnel is designed and constructed based on the ASHRAE standard 41.2 (ASHRAE, 1992) and the test procedure followed the same standard. The radiator dry condition test has water inside the tube and air on the outside. Inlet conditions are shown in Table 1.

Table 1: Inlet conditions for two prototypes

Parameters	SSHX	CuHX
Inlet air temperature [°C]	35	35
Inlet air RH [%]	33	45
Air flow rate [m^3/s]	0.03	-
	0.04	0.04
	0.05	-
	0.06	0.06
	0.07	-
Inlet water temperature [°C]	60	67.5
Water mass flow rate [g/s]	30	15
	50	25
	70	35

3.1 Test results of 1kW prototypes

Test results for the 1kW HXs are summarized in Figure 11, Figure 12 and Figure 13.

The energy balance of heat exchanger capacity is calculated using the definition provided by ASHRAE standard 41.2 (ASHRAE, 1992), which is the percentage ratio of the difference of air capacity and water capacity over the average of air and water capacities and is shown in Figure 11. Energy balance for the SSHX is within ±2.3% and energy balance for the CuHX is within ±2.9%, indicating that the heat loss to the surroundings is negligible. ASHRAE standard 41.2 (ASHRAE, 1992) requires that the energy balance should not exceed ±5.0%.

Heat exchanger capacity is plotted in Figure 12 showing the effects of air velocity and water velocity on heat exchanger capacity. Heat exchanger's capacity (Q) increases non-linearly as air velocity (V_a) increases and water velocity (V_w) increases. The gradient of heat exchanger capacity over air velocity ($\partial Q/\partial V_a$), which is the slope of the trend line, decreases as the air flow rate increases, and increases as water flow increases, for a certain air velocity. This is because when the air flow rate increases or the water flow decreases, the portion of air side thermal resistance decreases, the influence of air velocity on capacity diminishes, and vice versa. The gradient of heat exchanger capacity over water velocity ($\partial Q/\partial V_w$), which could also be seen in Figure 11 by comparing the discrepancy of the trend lines, decreases as water flow rate increases, and increases as airflow rate increases. Similar reasoning could be used to explain this: When water flow rate increases or air flow rate decreases, the portion of water side thermal resistance decreases, reducing the influence of water flow rate on capacity.

Air pressure drop is an important parameter in heat exchanger performance. In Figure 13, the heat exchangers air side pressure drop increases non-linearly with the increase of air velocity and the slope increases as air velocity increases. This is because higher air velocity increases frictional losses. Change in water flow rate changes the air density, resulting in a different air pressure drop, but it can be seen that this change is not significant.

Uncertainties are calculated based on both systematic uncertainty and random uncertainty, and summarized in Table 2.

3.2 Test results of 10kW prototype

The test results for CuHX-10kW are shown in Figure 14, Figure 15 and Figure 16. The Energy balance of CuHX-10kW is within the acceptable range of ±4.4% as shown in Figure 14. The capacity of the heat exchanger is plotted in Figure 15. The variation of heat exchanger capacity (Q), gradient of capacity over air velocity ($\partial Q/\partial V_a$), gradient of capacity over water velocity ($\partial Q/\partial V_w$) with the variation of air velocity (V_a) and water velocity (V_w) are found to be similar to the 1kW SSHX and CuHX. Air pressure drops are shown in Figure 16. It should be noted that the air pressure drop largely depends on airflow rate while the water flow rate has a negligible influence. Uncertainties are shown in Table 2.

Table 2: Uncertainty of all tests

	SSHX		CuHX		CuHX-10kW	
	Air	Water	Air	Water	Air	Water
Temperature	±0.15~0.18K	±0.05~0.08K	±0.5K	±0.5K	±0.10K	±0.05~0.08K
Flow rate	±0.7~1.7%	±0.2~0.3%	±0.089%	±0.2%	±0.9~1.0%	±0.2~0.3%
Pressure drop	±2.1~4.1%	±3.9~12.7%	0.25%	0.05%	±1.9~3.3%	±1.9~3.7%
Capacity	±1.3~1.9%	±1.2~3.7%	±2.24 %	±6.59 %	±1.4~2.0%	±0.7~1.5%

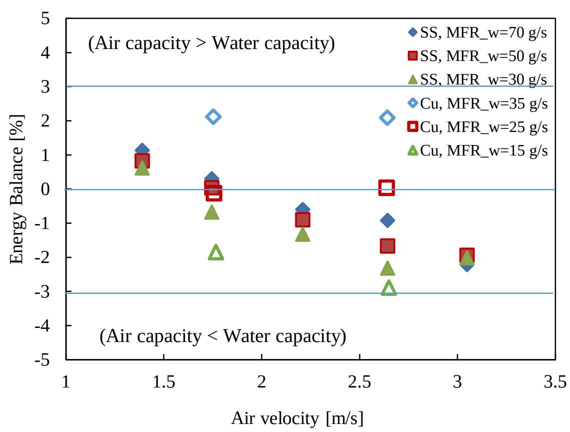

Figure 11: Energy balance between air and water for SSHX and CuHX

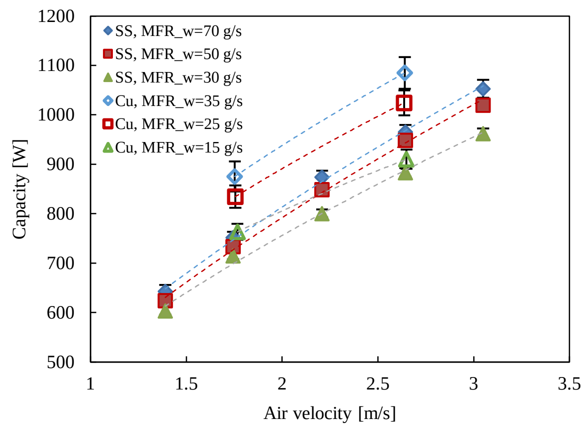

Figure 12: Effects of air velocity on capacity for different water flow rates for SSHX and CuHX

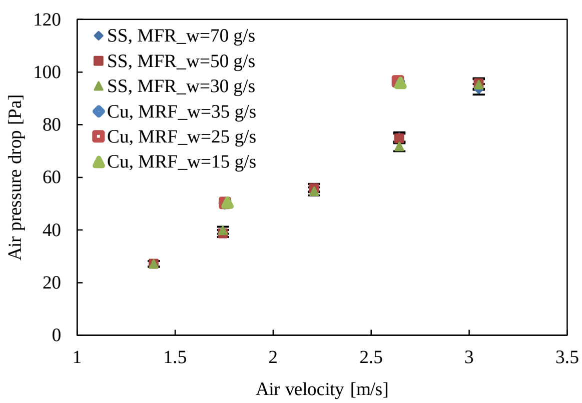

Figure 13: Effects of air velocity on air pressure drop for different water flow rates for SSHX and CuHX

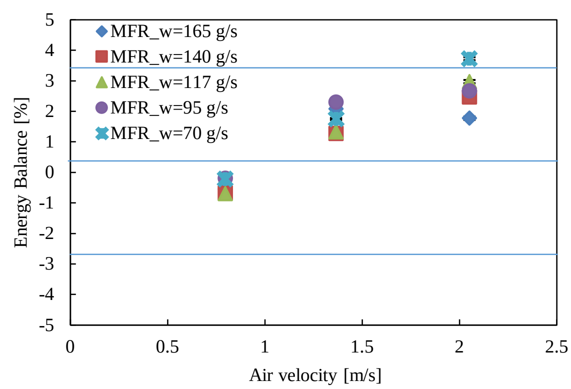

Figure 14: Energy balance between air and water for CuHX-10kW

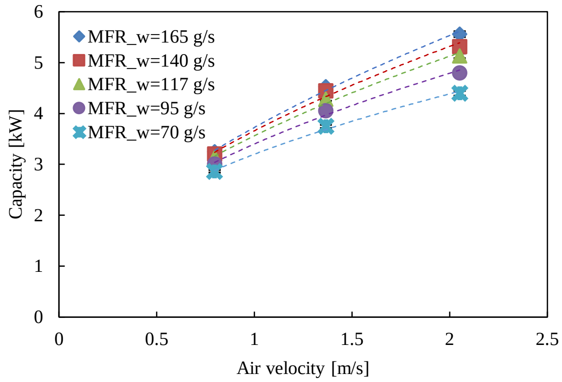

Figure 15: Effects of air velocity on capacity for different water flow rates for CuHX-10kW

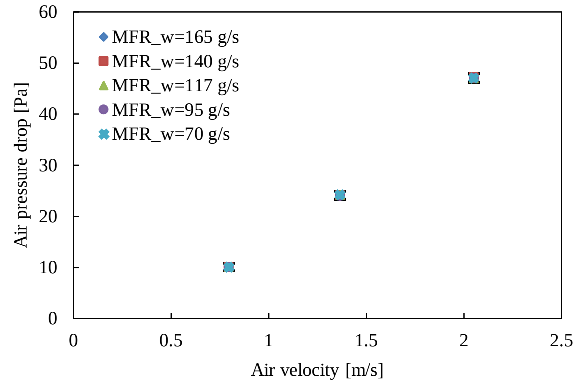

Figure 16: Effects of air velocity on air pressure drop for different water flow rates for CuHX-10kW

3.4 Lessons learned

The test procedures for all three heat exchangers follow the ASHRAE standards. However, due to the unique characteristics of bare tube heat exchangers with very small diameters and different capacities, some considerations are required:

- De-ionized water or filtered water has to be used in order to avoid potential blockages in the heat exchanger tubes by dirt in the fluid.
- Ideally, the test facility should be built in an air-conditioned space in order to ensure constant room temperature. This will minimize heat losses from the wind tunnel. Ductwork and pipes should be insulated to reduce heat losses. Heat leakage correction test should be conducted following ASHARE 41.2 (ASHARE, 1992). Energy balance should be within ±5.0%.
- To ensure capacity and pressure drop uncertainties are within ±5.0%, instruments with proper accuracy should be selected. Temperature and flow rate control should be accurate enough to minimize errors.
- Waterside temperature difference should not be too small, since it will increase uncertainty. The recommended minimum water temperature difference is 2°C.
- The ratio of duct size and the frontal area of the heat exchanger have an impact on the air flow measurement. Ideally, the duct size should be the same as the heat exchanger size; however, a slightly larger duct is acceptable. If the ratio is too high, meaning the duct is oversized, then airflow will be reduced in the upstream and downstream of the duct, resulting in additional heat losses, and the local pressure loss due to the contraction and expansion will not be negligible anymore. The maximum ratio is recommended to be 5.
- The uncertainty in tube spacing due to manufacturing will lead to uneven gaps between tubes, which has a significant impact on the performance. Tube spacers should be properly designed to ensure constant tube spacing.

4. CONCLUSIONS

The current study covers the manufacturing procedure and experimental performance of three bare tube heat exchangers. High capacity with small physical envelope is achieved by reducing the tube diameter to 0.8mm OD.
All three prototypes are experimentally investigated and the heat transfer performance as well as hydraulic characteristics is discussed. The energy balance of all tests are found to be within ±5% and uncertainties in averaged capacity were within ±3%. It is found that the capacity and pressure drop both increases non-linearly as the increase in air velocity. The increase of water flow rate results in the increase of capacity and has a negligible influence on air pressure drop. Due to the small tube size, clean water should be used. Room temperature should be controlled in order to achieve and maintain steady state during tests. Instruments should be properly selected to guarantee the accuracy of results. Water temperature difference larger than 2°C will minimize errors and the ratio of duct size over heat exchanger frontal area is recommended to be no more than 5.

From a manufacturing aspect, it is possible to produce leak-free heat exchangers using the process described herein. The heat exchanger produced has 2280 tubes with 4560 leak-free joints. The use of silver-solder was found to be a forgiving process with the advantage of possible repair, contrary to the use of epoxy. The developed manufacturing process can possibly be automated for production. Tube manufacturer quality control should ensure leak-free tubes. It is difficult to repair tube leaks in a core deeper than 4-rows deep. Use of overlapping spacers when using modular units is preferred in order to avoid gaps between the modules.

NOMENCLATURE

Cu	copper	(–)
HX	heat exchanger	(–)
Q	capacity	(W)
SS	stainless steel	(–)
V_a	air velocity	(m/s)
V_w	water velocity	(m/s)

Subscript

a	air
w	water

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