Design and Numerical Parametric Study of Fractal Heat Exchanger

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ABSTRACT

Air-to-refrigerant heat exchangers are a main component in air-conditioning and heat pump systems and are therefore a topic of major research focus. Such heat exchangers, mainly made of fin-and-tube and microchannels, use fins to augment the heat transfer area of the air-side. Recently it has been shown that finless designs using ≤ 1 mm hydraulic diameter bare tubes can deliver better air-side heat transfer performance than conventional heat exchangers. In current study, a novel air-to-refrigerant heat exchanger consisting of branching bare tubes with two different tube diameters is designed using fractal theory. The tubes in the first level split into two or more tubes at certain angles. The secondary tubes then merge back to the first level tube again. Principles of fractal design are applied to improve both air-side heat transfer area and heat transfer coefficient. The heat transfer and pressure drop characteristics are numerically analyzed using a 3-D model and compared with the bare tube heat exchangers. The novel finless heat exchanger based on fractals is found to have intrinsic advantage of increasing heat transfer performance and is a promising area for future investigations.

1. INTRODUCTION

Improving the performance of air-conditioning and heat pump systems is a main research topic due to the increasing demand of energy saving. Lots of research has been done to enhance heat transfer performance of air-to-refrigerant heat exchangers due to its crucial role in heating, ventilation, air-conditioning and refrigeration (HVAC&R) systems. The key issue to further increase the efficiency of air-to-refrigerant heat exchangers is to reduce thermal resistance on the air side; thus, various designs have been investigated in past decades to solve this problem.

A typical method to enhance air-side heat transfer is to augment air-side heat transfer area by employing extended heat transfer surfaces, such as fins. Fin-and-tube heat exchangers with a diameter larger than 5 mm have been widely investigated (Wang et al., 2000, Singh et al., 2009, 2011). Many fin configurations have been proposed and experimentally investigated in literature in the past few years, such as slit fin (Wang et al., 1999, Du and Wang, 2000), fins with longitudinal vortex-generator (Kwak et al., 2002, Torri et al., 2002, Zhang et al, 2008), and crimped spiral fins (Tang et al., 2009). Despite all these researches, the air-side heat transfer coefficient measured at lower Reynolds number (Re) is still relatively small. Joardar and Jacobi (2008) reported the air-side heat transfer coefficient was only around 50 $W \cdot m^{-2} \cdot K^{-1}$ when Re was approximately 900 as the test results of a fin (with vortex generator)-and-tube heat exchanger with outer diameter of 10.67 mm.

Recently, small diameter (<5 mm) finless heat exchangers have been proposed and investigated. Bacellar (2014) numerically investigated bare tube heat exchanger (BTHX) and plain fin-and-tube heat exchanger with a diameter of 2 - 5 mm and developed correlations based on simulation results of CFD. Paitoonsurikarn et al. (2000) numerically found that bare tube heat exchanger could achieve a much larger air-side heat transfer coefficient as 300 $W \cdot m^{-2} \cdot K^{-1}$ with the air velocity range of 1 ~ 6 $m \cdot s^{-1}$ by reducing tube diameter to 0.3 ~ 0.5 mm. Thus, finless designs using 1 mm and smaller hydraulic diameter bare tubes are capable of exceeding the air-side heat transfer performance of conventional heat exchangers.

In heat sink design for electronic devices, fractal channels have been found to have intrinsic advantage of minimized flow resistance and strong heat transfer capability. (Chen and Cheng, 2002, Wang et al., 2010, Yu et al., 2012). However, no research has been conducted to liquid-to-gas heat exchangers in the past.

In current study, a novel fractal heat exchanger consisting of branching bare tubes was proposed and numerically studied using CFD simulation tools. A parametric study was conducted and thermal and hydraulic performances were analyzed and compared with bare tube heat exchangers. Parameters studied include tube diameter, bifurcation angle, length ratio and air velocity. All CFD models were carried out using Gambit® 2.4.6 and ANSYS Fluent® 14.5.

2. FRACTAL HEAT EXCHANGER DESIGN

This novel heat exchanger consists of two levels of tubes, the main tubes and the branch tubes. Main tubes are all vertical tubes in Figure 1 of which outer diameters are symbolled as D_1 while branch tubes are symbolled as D_2. Longitudinal tube pitch (P_l) is defined as the center distance of two adjacent main tubes. The transversal tube pitch (P_t) is then defined as the center distance of two adjacent layers, as shown in Figure 2. Bifurcation angle (θ) is the angle between brunching tube and the center line. Centerlines of all tubes will generate a honey comb structure consists of multiple hexagons in this example, indicated by dotted line. Thus this novel heat exchanger is named as honeycomb heat exchanger (HCHX). Figure 3 shows the heat exchanger shape with header and the flow directions of two fluids on both sides. Tubes can be either staggered or in-line in the air flow direction.

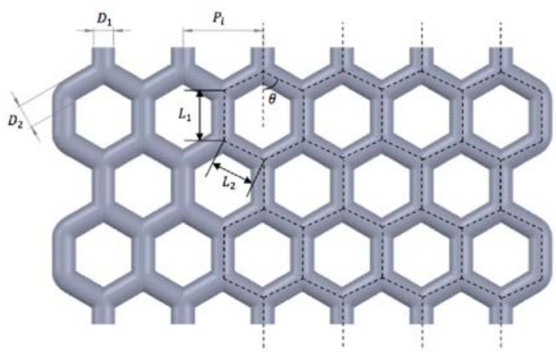

Figure 1: HCHX-tube structure

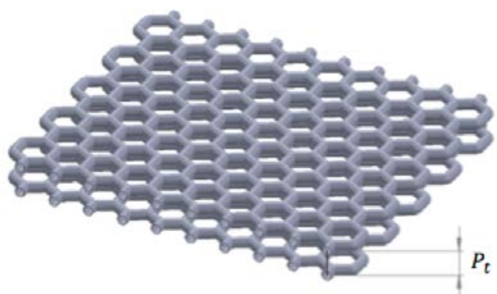

Figure 2: HCHX-staggered pattern

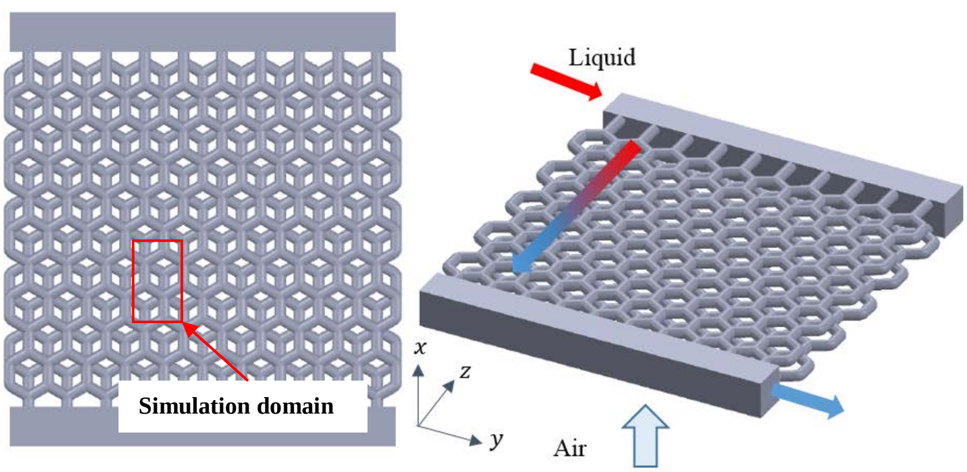

Figure 3: HCHX schematic (staggered) and simulation domain

3. PARAMETRIC STUDY

3.1 Parameters

The parameters studied are summarized in Table 1. Diameter ratio (DR) is defined as the ratio of D_1 over D_2 and length ratio (LR) is defined as the ratio of length of main tube (L_1) over length of branch tube (L_2). DR was fixed to be 0.7 to maintain the mass flux constant inside tubes in current study. To compare the heat exchanger's performance, the main tube's diameter (D_1) is kept the same as the diameter of bare tube heat exchanger (D) in comparison.

Table 1: Parameters for BTHX and HCHX

Types	Parameters	Units	BTHX	HCHX
Constants	P_l	[mm]	1.5 D	$1.5D_1$
	P_t	[mm]	1.5 D	$1.5D_1$
	DR	-	-	0.7
Variables	D or D_1	[mm]	1, 2, 3	1, 2, 3
	V_a	[m/s]	0.5, 2, 3.5	0.5, 2, 3.5
	θ	[deg]	-	30, 45, 60
	LR	-	-	1.414, 1.732

3.2 CFD Modeling

The computational domain of BTHX is shown in Figure 4 as well as the air flow direction. It is a two dimensional cross section of two row staggered tubes of the heat exchanger. End effects are neglected. Boundary conditions are defined as constant and homogeneous velocity distribution at inlet, constant pressure at outlet (0.0 Pa gauge), symmetry flow at top and bottom of computational domain, and tubes as walls. A triangular mesh element is set for the models and a refined boundary layer mesh at tube walls is modeled in order to capture the momentum and thermal boundary layers development with higher accuracy. The computational domain of HCHX is shown in Figure 3 and Figure 5. It is a three dimensional cross section of two rows of staggered tubes of the heat exchanger. End effects are neglected. Boundary conditions are defined as constant and homogeneous velocity distribution at inlet, constant pressure at outlet (0.0 Pa gauge), symmetry flow at front and back (xz-plane) of computational domain, periodic boundaries on the top and bottom (xy-plane) of the computational domain and tubes as walls. A hybrid tetrahedral mesh element is set for the models and a refined boundary layer mesh at tube walls is modeled in order to capture the momentum and thermal boundary layers development with higher accuracy. The air inlet temperature is fixed at 300 K while the tube wall temperature is 350 K. The turbulent k-ε realizable model is used with enhanced wall functions enabled in every simulation. A second order upwind space discretization is set to ensure better accuracy. Convergence criteria is defined as 1.0e-5 for continuity and velocities, 1.0e-6 for energy, and 1e-3 for turbulent kinetic energy (k) and eddy viscosity (ε). Ideal-gas model is used for density, and all the other properties are assumed to be constant.

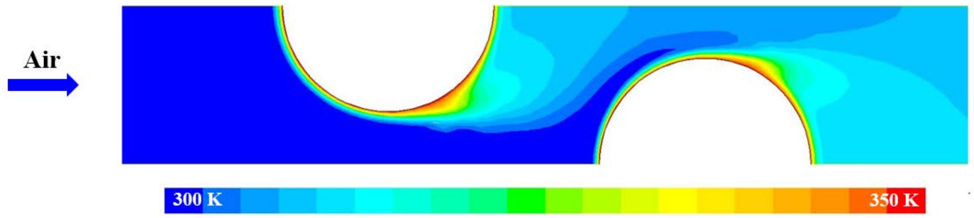

Figure 4: BTHX computational domain

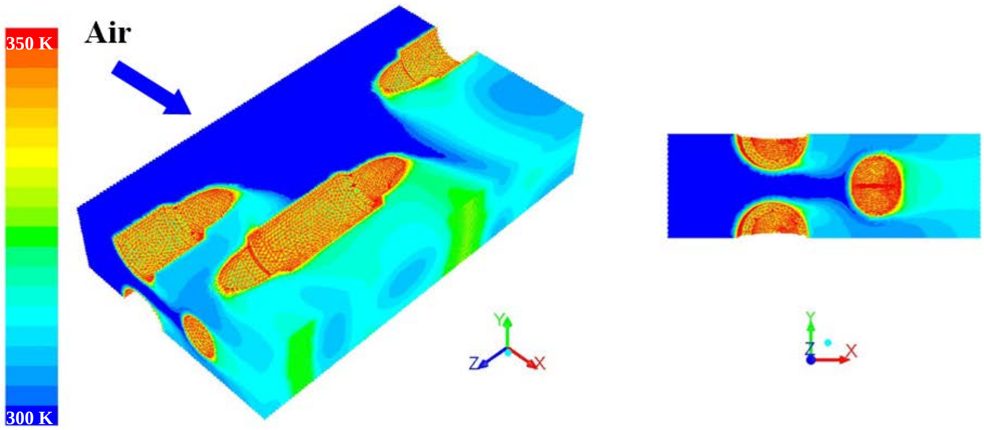

Figure 5: HCHX computational domain (left) and bottom view (right)

3.3 Results and discussion

3.3.1 Heat transfer area

Air-side heat transfer of HCHX is intrinsically larger than that of BTHX with same diameter. The comparison of cases studied is summarized in Figure 6. As bifurcation angle increases or length ratio decreases, the bifurcation number increases in a certain heat exchanger length, resulting in a larger heat transfer area improvement. Diameter has no effect on heat transfer area improvement as long as the diameter ratio is fixed.

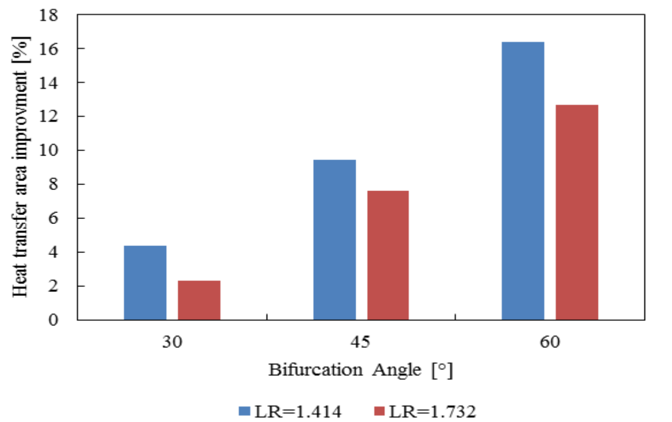

Figure 6: Air-side heat transfer area improvement compared with bare tube heat exchanger

3.3.2 Heat transfer coefficient

Heat transfer coefficient of HCHX under different conditions are plotted in Figure 7 and Figure 8 while the heat transfer coefficient percentage improvement compared with BTHX are shown in Figure 9 and Figure 10. Different colors of dots represent different diameters while different shapes represent different bifurcation angles. Here we discuss about the influence of air velocity, diameter, bifurcation angle and length ratio on heat transfer coefficient.

As air velocity increases, the air-side heat transfer coefficient increases non-linearly for all cases while the increasing rate decreases. As diameter decreases or bifurcation angle increases, the air-side heat transfer coefficient increases, indicating that smaller diameter with larger bifurcation angle is preferred for higher heat transfer coefficient for HCHX. The influence of length ratio is minor which can be concluded by comparing Figure 7 and Figure 8. Among these three factors, diameter has a major influence because that the highest four data sets are HCHX with diameter of 1 mm. Bifurcation angle also has a large impact when diameter goes up for that with bifurcation, the HCHX with diameter of 3 mm can surpass the heat transfer coefficient of BTHX with diameter of 2 mm, indicating that the bifurcation structure has an advantage of air heat transfer coefficient. For all the cases of HCHX with an air velocity no less than 2 m/s, the air-side heat transfer coefficient is higher than that of the corresponding BTHX. At 0.5 m/s, BTHX only surpass the performance of HCHX when the diameter is 1 mm.

The percentage improvement is shown in Figure 9 and Figure 10. The percentage improvement increases as air velocity increases, diameter increases and bifurcation angle increases while length ratio has a minor effect. Therefore, HCHX with larger diameter with larger bifurcation angle is preferred for higher heat transfer coefficient improvement compared with BTHX.

3.3.3 Pressure drop

The air-side pressure drop is plotted in Figure 11 and Figure 12 while the pressure drop percentage improvement is plotted in Figure 13 and Figure 14. Pressure drop increases as air velocity increases, diameter decreases and bifurcation angle increases. Bifurcation angle has a major impact on pressure drop, which can be concluded by observing that the geometries with bifurcation angle 60° have the highest pressure drop, followed by those with 45° bifurcation angle and those with 30° bifurcation angle. This is because as bifurcation angle increases, the ratio of tube volume over computational domain volume increases, thus pressure drop increases. The decrease of length ratio will cause the air pressure drop to increase because that at same length there are more branching tubes. Thus for lower pressure drop, larger diameter and smaller bifurcation angle are preferred.

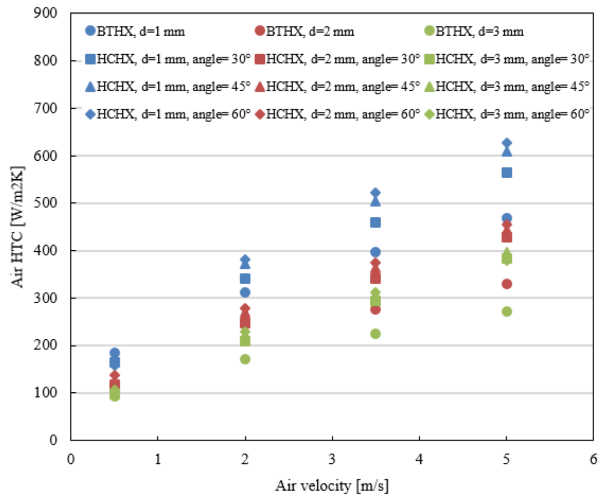

Figure 7: Heat transfer coefficient comparison (LR=1.414)

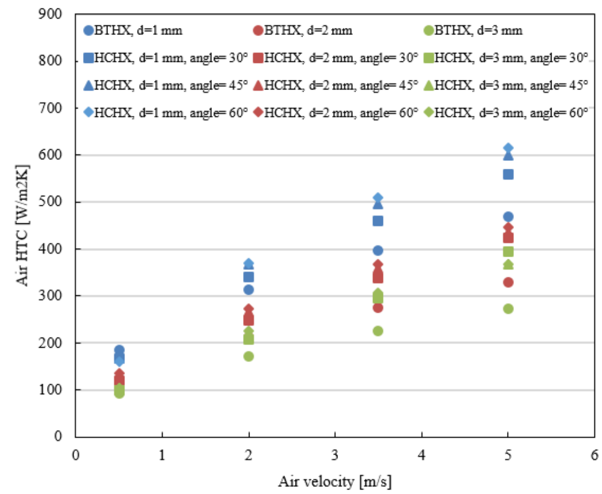

Figure 8: Heat transfer coefficient comparison (LR=1.732)

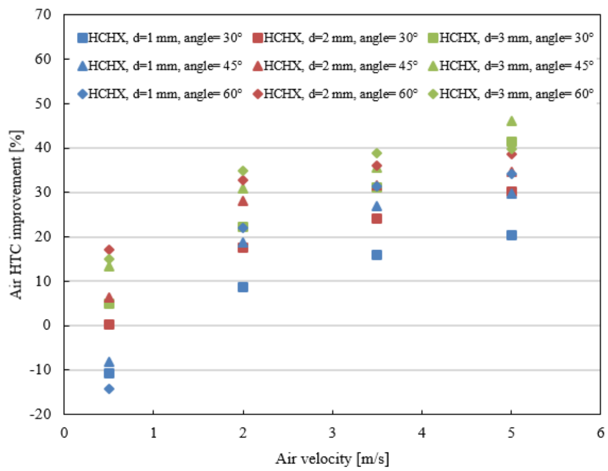

Figure 9: Heat transfer coefficient improvement compared with BTHX (LR=1.414)

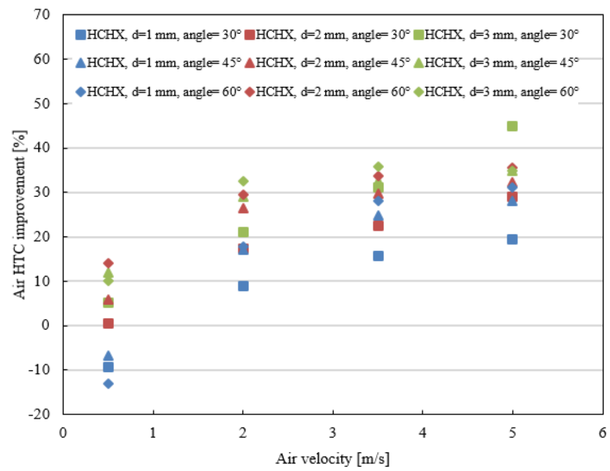

Figure 10: Heat transfer coefficient improvement compared with BTHX (LR=1.732)

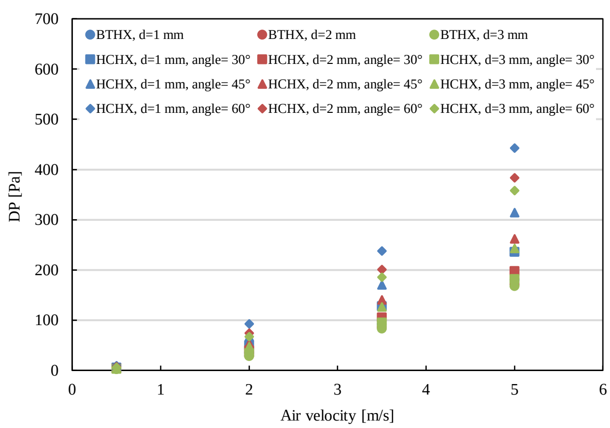

Figure 11: Heat exchanger air-side pressure drop comparison (LR=1.414)

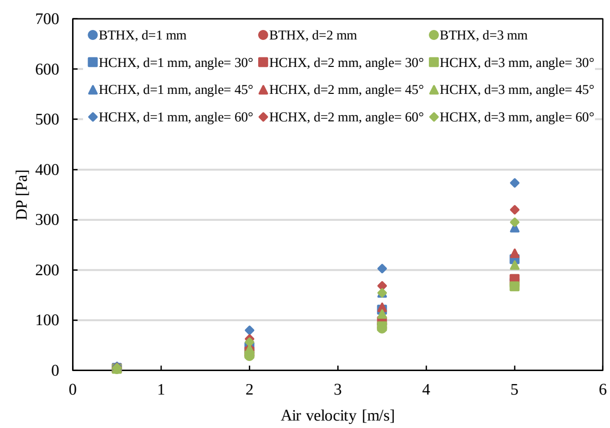

Figure 12: Heat exchanger air-side pressure drop comparison (LR=1.732)

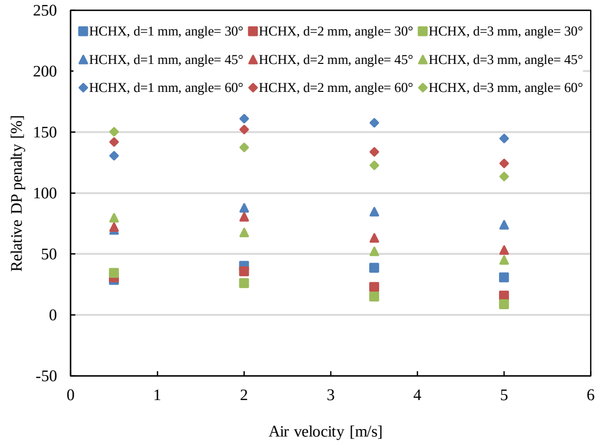

Figure 13: Heat exchanger air-side pressure drop improvement compared with BTHX (LR=1.414)

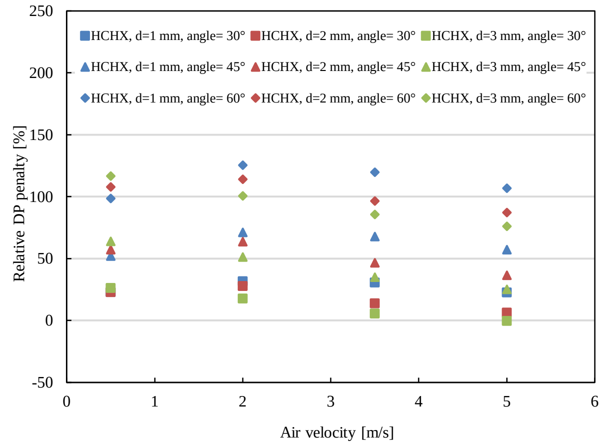

Figure 14: Heat exchanger air-side pressure drop improvement compared with BTHX (LR=1.732)

Compared to the pressure drop of BTHX, there are air-side pressure drop penalty for most cases while in some cases, the HCHX yields a slightly lower pressure drop. Relative pressure drop penalty increases as bifurcation angle increases or diameter decreases for most cases. As for HCHX with a bifurcation angle of 60° at 0.5 m/s, the relative pressure drop penalty increases as diameter decreases, bifurcation angle increases, or length ratio decreases. As the air velocity increases, the relative pressure drop penalty slightly increases first and then decreases for some cases but decreases for most cases. This is due to the different increasing rate of BTHX and HCHX as air velocity increases within current air velocity range. Thus HCHX designs with smaller diameter and larger bifurcation angle are preferred for a lower relative pressure drop penalty.

3.3.4 hA value

To find out the improvement in heat transfer of the geometry itself, a comparison of hA, which is the product of heat transfer coefficient and heat transfer area, is necessary. hA values are plotted in Figure 15 through Figure 20. For each graph, it is obvious that hA values of HCHX are higher than that of the corresponding BTHX. hA value increases when air velocity increases and diameter increases. In current hA calculation, A is the heat transfer area of the calculation domain, meaning the heat transfer area decreases as bifurcation angle increases or length ratio decreases. Thus as bifurcation angle increases or length ratio decreases, hA decreases as shown in Figure 15 through Figure 20. Thus larger diameter is preferable for achieving a larger hA value.

The percentage improvement of hA value is plotted in Figure 21 and Figure 22. The percentage improvement of hA value increases as air velocity increases, diameter increases, bifurcation angle increases or length ratio decreases. Thus larger diameter and larger bifurcation angle with lower length ratio are preferred to result in a higher percentage improvement of hA.

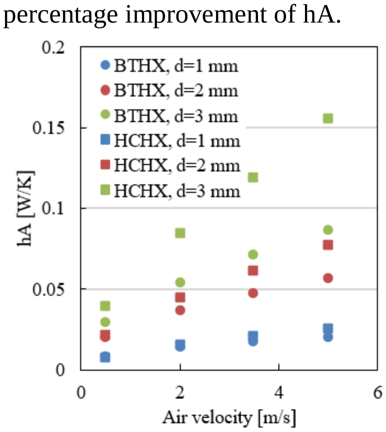

Figure 15: hA value (LR=1.414, angle=30°)

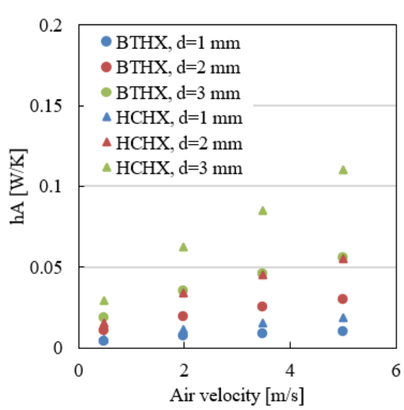

Figure 16: hA value (LR=1.414, angle=45°)

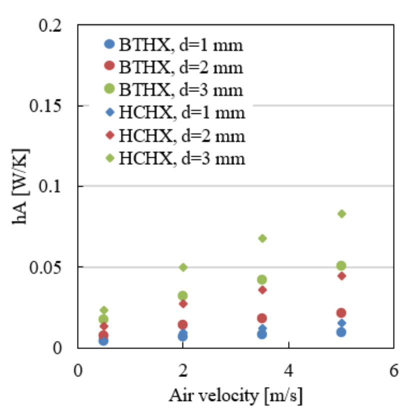

Figure 17: hA value (LR=1.414, angle=60°)

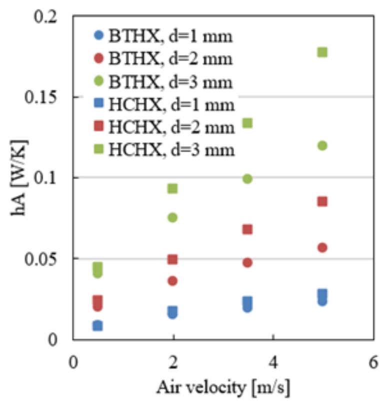

Figure 18: hA value (LR=1.732, angle=30°)

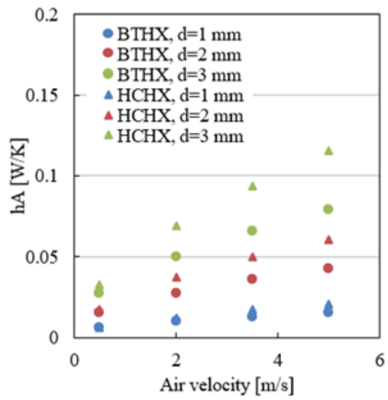

Figure 19: hA value (LR=1.732, angle=45°)

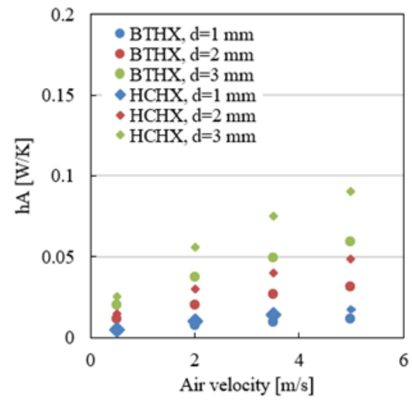

Figure 20: hA value (LR=1.732, angle=60°)

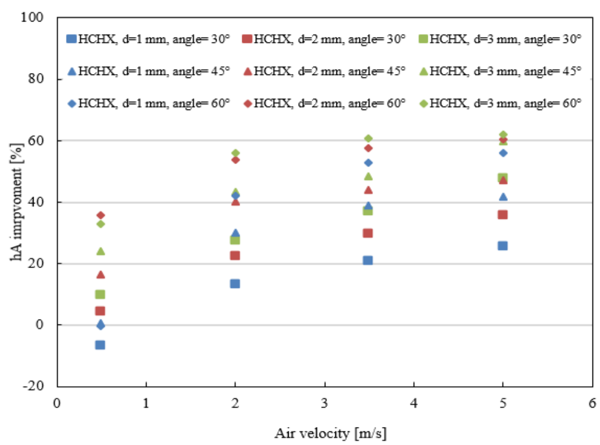

Figure 21: hA improvement compared with BTHX (LR=1.414)

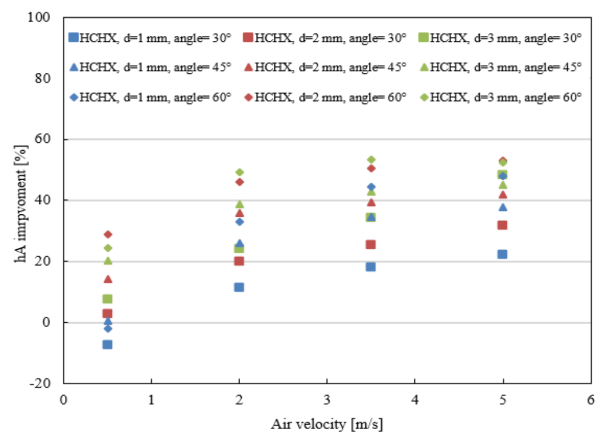

Figure 22: hA improvement compared with BTHX (LR=1.732)

4. CONCLUSIONS

In the current study, a novel finless heat exchanger design was proposed using fractal theory and was numerically investigated. A parametric study was conducted and thermal hydraulic performances are analyzed. Parameters studied include air velocity (1~3 m/s), tube diameter (1, 2, 3 mm), bifurcation angle (30, 45, 60°) and length ratio (1.414, 1.732). The results are compared with that of BTHX for the same diameters. The simulation demonstrates that larger bifurcation angle and larger air velocity are desired while length ratio has a minor effect on the air-side heat transfer coefficient improvement. Smaller diameter is beneficial for achieving a higher heat transfer coefficient while larger diameter design can result in a higher percentage improvement. To achieve higher air side hA value percentage improvement, larger tube diameter, larger bifurcation angle, larger air velocity and smaller length ratio are preferred. HCHX designs with larger diameter, smaller bifurcation angle, lower air velocity and larger length ratio are beneficial for minimize air-side pressure drop.

NOMENCLATURE

BTHX	bare tube heat exchanger	(–)
D	diameter	(mm)
D_1	main tube diameter	(mm)
D_2	branch tube diameter	(mm)

DR	diameter ratio	(–)
HCHX	honeycomb heat exchanger	(–)
LR	length ratio	(–)
L_1	main tube length	(mm)
L_2	branch tube length	(mm)
P_l	longitudinal tube pitch	(mm)
P_t	transversal tube pitch	(mm)
V_a	air velocity	(m/s)
θ	bifurcation angle	(deg)

REFERENCES

ANSYS® Academic Research, Release 14.5 (2012). ANSYS Fluent User's Guide, ANSYS, Inc.

Bacellar, D., Aute, V., Radermacher, R. (2014). CFD-Based Correlation Development For Air Side Performance Of Finned And Finless Tube Heat Exchangers With Small Diameter Tubes, *Int. Refrig. Air Cond. Conf.* 1410.

Chen, Y., and Cheng, P. (2002). Heat transfer and pressure drop in fractal tree-like microchannel nets. *International Journal of Heat and Mass Transfer*, 45(13), 2643–2648.

Du, Y.J., Wang, C.C. (2000). An experimental study of the air side performance of the superslit fin-and-tube heat exchangers, *International Journal of Heat and Mass Transfer.* 43: 4475–4482.

Joarda,r A., Jacobi, A. M. (2008). Heat transfer enhancement by winglet-type vortex generator arrays in compact plain-fin-and-tube heat exchangers, *International Journal of Refrigeration.* 31: 87–97.

Kwak, K., Torii, K., Nishino, K. (2002). Heat transfer and flow characteristics of fin-tube bundles with and without winglet-type vortex generators, *Exp. Fluids.* 33: 696–702.

Paitoonsurikarn, S., Kasagi, N., Suzuki, Y. (2000). Optimal design of micro bare-tube heat exchanger, *Proceedings of the Symposium on Energy Engineering (SEE2000)*, vol. 3, pp. 972-979.

Singh, V., Abdelaziz, O., Aute, V., Radermacher, R. (2011). Simulation of air-to-refrigerant fin-and-tube heat exchanger with CFD-based air propagation, *International Journal of Refrigeration*, 34, pp. 1883-1897.

Singh, V., Aute, V., Radermacher, R. (2009). A heat exchanger model for air-to-refrigerant fin-and-tube heat exchanger with arbitrary fin sheet, *International Journal of Refrigeration*, 32, pp. 1724-1735.

Tang, L.H., Zeng M., Wang, Q.W. (2009). Experimental and numerical investigation on air-side performance of fin-and-tube heat exchangers with various fin patterns, *Exp. Therm. Fluid Sci.* 33: 818–827.

Torii, K., Kwak, K.M., Nishino, K. (2002). Heat transfer enhancement accompanying pressure-loss reduction with winglet-type vortex generators for fin-tube heat exchangers, *International Journal of Heat and Mass Transfer*. 45: 3795–3801.

Wang, C.C., Chi, K.Y., Chang, C.J. (2000). Heat transfer and friction characteristics of plain fin-and-tube heat exchangers, part I: New experimental data, *International Journal of Heat and Mass Transfer*, 43, pp. 2681-2691.

Wang, C.C., Tao, W.H., Chang, C.J. (1999). Investigation of the airside performance of the slit fin-and-tube heat exchangers, *International Journal of Refrigeration.* 22, 595–603.

Wang, X. Q., Xu, P., Mujumdar, A. S., and Yap, C. (2010). Flow and thermal characteristics of offset branching network. *International Journal of Thermal Sciences*, 49(2), 272–280.

Yu, X. F., Zhang, C. P., Teng, J. T., Huang, S. Y., et al. (2012). A study on the hydraulic and thermal characteristics in fractal tree-like microchannels by numerical and experimental methods. *International Journal of Heat and Mass Transfer*, *55*(25-26), 7499–7507.

Zhang, Y. H., Wu, X., Wang, L.B., Song, K.W., et al. (2008). Comparison of heat transfer performance of tube bank fin with mounted vortex generators to tube bank fin with punched vortex generators, *Exp. Therm. Fluid Sci.* 33: 58–66.

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