International Journal of Heat and Mass Transfer 54 (2011) 2234–2240

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Technical Note

Effect of fin pitches on the air-side performance of crimped spiral fin-and-tube heat exchangers with a multipass parallel and counter cross-flow configuration

Parinya Pongsoi [a], Santi Pikulkajorn [b], Chi-Chuan Wang [c], Somchai Wongwises [a,*]

[a] Fluid Mechanics, Thermal Engineering and Multiphase Flow Research Lab. (FUTURE), Department of Mechanical Engineering, King Mongkut's University of Technology Thonburi, Bangmod, Bangkok 10140, Thailand

[b] Somchai Industry Co., Ltd, Bangkok 10150, Thailand

[c] Department of Mechanical Engineering, National Chiao Tung University, Hsinchu 300, Taiwan

ARTICLE INFO

Article history:
Received 4 November 2010
Received in revised form 3 December 2010
Accepted 3 December 2010
Available online 27 January 2011

Keywords:
Fin
Heat transfer
Heat exchanger
Air–water
Extended surface

ABSTRACT

This study investigates the effect of fin pitches and fin materials on the air-side performance of crimped fin-and-tube heat exchangers in the range of high Reynolds numbers (4000–13000). The test samples are made from copper and aluminium with different fin pitches (f_p = 3.2, 4.2 and 6.2 mm). It is found that the proposed simple average effectiveness equation from the pure counter and parallel circuitry arrangement can well represent the effectiveness-NTU relationship for the current z-shape arrangement. The experimental results reveal that the fin pitch casts insignificant effect on the heat transfer characteristics (Colburn *j* factor). However, a detectable rise of the friction factor is seen when the fin pitch is increased to f_p = 6.2 mm. On the other hand, the effect of fin material on the airside performance is negligible.

1. Introduction

The heat exchanger is a basic equipment for thermal systems in many industrial processes involving heat transfer. One of the most favourable configurations of heat exchanger used in industrial applications takes the form as fin-and-tube heat exchanger. Normally for this type of heat exchanger, the dominant thermal resistance is on the air-side of the heat exchanger. Hence one way to augment the heat transfer performance is via enhanced fin geometry. There have been numerous fin configurations such as plain fin, slit fin, wavy fin, louvered fin, circular fin, annular fin, spiral fin, compounded fin and the like that were already implemented in various industrial applications. Upon the foregoing fin configurations, the spiral fin featuring easy production, is quite common in industrial services. However, there is only a few studies concerning the air-side performance of the spiral fin-and-tube heat exchanger [1–6]. According to these literatures, crimped spiral fin is proved to be quite reliable in industrial applications [1–5]. Upon the foregoing studies, Nuntaphan et al. [1] is the only experimental work that examines the effect of fin pitches on the air-side performance of crimped spiral fin-and-tube heat exchangers. However, this study only discussed the effect at a very low air frontal velocity (0.5–1.5 m/s). In practice, especially in industrial service, the operation velocity is normally much higher [7]. Therefore, it is the main purpose of this study is to extend the applicable range (V_{fr} up to 6 m/s) of the spiral heat exchangers subject to the influence of fin pitch. Moreover, the effect of fin materials on the air-side performance is also examined.

2. Data reduction

This present work is conducted by using the experimental apparatus of Wongwises and Chokeman [8], including the test section, air supply, water loop, instrumentation, and data acquisition. Air and hot water are used as working fluids. Detailed descriptions of the relevant components can be seen from the previous study.

In the experiment, the inlet water temperature and the water flow rate are fixed while varying the air flow rate. Tests are then conducted at the steady state with tested conditions being tabulated in Table 1.

The tests heat exchangers are of fin-and-tube configurations with copper tube being finned with either copper or aluminium. The water-side circuitry arrangement and detailed dimensions of the tested fin-and-tube heat exchangers are shown in Fig. 1. Photos of the crimped spiral fin pattern are shown in Fig. 2. The geometric parameters of the heat exchangers are summarized in Table 2. Tests are performed under steady state condition, and the overall resistance can be obtained from the *UA* product of transfer units (ε-NTU), yet the total resistance is the sum of the individual resistances as follows:

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{\ln(d_o/d_i)}{2\pi k_t L} + \frac{1}{\eta_o h_o A_o} \tag{1}$$

* Corresponding author. Tel.: +66 24709115.
E-mail address: somchai.won@kmutt.ac.th (S. Wongwises).

0017-9310/$ - see front matter © 2010 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijheatmasstransfer.2010.12.027

Nomenclature

$A_{\min}$	minimum free flow area, m^2
A_o	total surface area, m^2
A_p	cross-sectional or profile area of fin, m^2
Al	aluminium material
C^*	capacity rate ratio, dimensionless
Cu	copper material
f	Fanning friction factor, dimensionless
G_c	mass flux of the air based on minimum free flow area, kg/m^2 s
h	heat transfer coefficient, W/(m^2 K)
j	Colburn factor, dimensionless
NTU	number of transfer units, dimensionless
Nu	Nusselt number, dimensionless
Pr	Prandtl number, dimensionless
ΔP	pressure drop, Pa
Re_{d_o}	Reynolds number based on tube outside diameter(d_o)
U	overall heat transfer coefficient, W/(m^2 K)
V_{fr}	air frontal velocity, m/s
$V_{\max}$	maximum velocity across heat exchanger, m/s
ε_c	heat exchanger effectiveness for multipass counter cross-flow
ε_p	heat exchanger effectiveness for multipass parallel cross-flow
ε_{pc}	heat exchanger effectiveness for multipass parallel and counter cross-flow
η_f	fin efficiency, dimensionless
ϕ	combination of terms, dimensionless

Table 1
Experimental conditions.

Inlet-air-dry bulb temperature, °C	31.5 ± 0.5
Inlet-air frontal velocity, m/s	2–6 or Re_{d_o} (4000–13000)
Inlet-water temperature, °C	55–70
Water flow rate, LPM	12–14

The ε-NTU relationships with one fluid mixed and one fluid unmixed in cross-flow was employed to determine the overall heat transfer coefficient. From Fig. 1, the present mixed circuitry arrangement is a combination of parallel and counter cross-flow. From the previous discussion, the ε-NTU relationships for multipass parallel cross-flow and multipass counter cross-flow configuration are available from [9–11], as shown in Eqs. (2) and (3):

For multipass counter cross-flow with $N_{\text{row}} = 2$:

$$\varepsilon_c = 1 - \left[\frac{K}{2} + \left(1 - \frac{K}{2}\right)e^{2K/C_A^*}\right]^{-1}, \quad K = 1 - e^{\text{NTU}_A(C_A^*/2)} \tag{2}$$

For multipass parallel cross-flow with $N_{\text{row}} = 2$:

$$\varepsilon_P = \left(1 - \frac{K}{2}\right)\left(1 - e^{-2K/C_A^*}\right), \quad K = 1 - e^{-\text{NTU}_A(C_A^*/2)} \tag{3}$$

where $C^* = C_{\min}/C_{\max}$ is equal to C_c/C_h or C_h/C_c depending on the value of hot and cold fluid heat capacity rates. However, the multipass parallel and counter cross-flow used in this experiment is a combination of multipass parallel cross-flow. Hence it may be reasonable to use the average value of the relationships shown in Eq. (4) as follows:

$$\varepsilon_{pc} = \frac{\varepsilon_P + \varepsilon_C}{2} \quad \text{for } N_{\text{row}} = 2 \tag{4}$$

The schematic diagram of circuitry arrangement (parallel, counter, cross) for $N_{\text{row}} = 2$ are shown in Fig. 3.

Further details about the data reduction can be seen from Wongwises and Chokeman [8]. The efficiency of a radial fin with rectangular profile is based on the derivation of Gardner [12], i.e.,

$$\eta_f = \frac{2\psi}{\phi(1+\psi)} \frac{I_1(\phi R_o)K_1(\phi R_i) - I_1(\phi R_i)K_1(\phi R_o)}{I_0(\phi R_i)K_1(\phi R_o) + I_1(\phi R_o)K_0(\phi R_i)} \tag{5}$$

where

$$\phi = (r_o - r_i)^{3/2}\left(\frac{2h_o}{k_f A_p}\right)^{1/2} \tag{6}$$

Accordingly, the air-side heat transfer coefficient (h_o) can then be calculated from Eq. (1). The air-side heat transfer characteristics of the heat exchanger are often in terms of dimensionless Colburn j factor:

$$j = \frac{Nu}{Re_{d_o}Pr^{1/3}} = \frac{h_o}{\rho_a V_{\max} C_P}(Pr)^{2/3} \tag{7}$$

The frictional characteristics are termed with Fanning friction factor, as depicted by Kays and London [13]:

$$f = \left(\frac{A_{\min}}{A_o}\right)\left(\frac{\rho_m}{\rho_1}\right)\left[\frac{2\Delta P\rho_1}{G_c^2} - (1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right)\right] \tag{8}$$

where G_c is the mass flux of the air based on minimum free flow area, A_o is the total heat transfer area, $A_{\min}$ is the minimum free flow area. The experiments are conducted following the ANSI/ASHRAE 33 Standards [14] in which the energy un-balance between air and water of the crimped spiral fin and tube heat exchangers, denoting $|Q_a - Q_w|/Q_{ave}$, is less than 0.05. The uncertainties are calculated from the root mean sum square method, the maximum uncertainties are 12.30% for the j-factor and 11.13% for f-factor.

3. Results and discussion

As aforementioned in Eq. (4) which was used to calculate the Colburn factor (j) for $N_{\text{row}} = 2$. The reasons for using Eq. (4) can be explained by Fig. 4(a) which shows the calculation of heat exchanger effectiveness using Eqs. (2)–(4) subject to variation of NTU and C^*. Apparently, it is not suitable to use either counter flow (Eq. (2)) or parallel flow (Eq. (3)) when operating at a high Reynolds number. For the present flow configuration, the heat exchanger effectiveness lies between the multipass parallel cross-flow and counter cross-flow. Note that using inappropriate ε-NTU relationships for complex flow arrangement may lead to an error up to 10% as pointed out by Navarro and Cabezas-Gomez [15]. Moreover, in Fig. 4(a), the heat exchanger effectiveness using Eq. (4) is found to be in excellent agreement with the simulated result of the z-shaped arrangement from [15]. Notice that the z-shape arrangement is analogous to the present circuitry arrangement. As a consequence, Eq. (4) can be proposed as the empirical ε-NTU relationship for the multipass parallel and counter cross-flow arrangement.

Fig. 4(b) and (c) shows the effect of fin pitches on the performance of heat exchangers having a two-row configuration with copper and aluminium fin, respectively. The corresponding fin pitches were 3.2, 4.2 and 6.2 mm. Test results indicated that the effect of fin pitch on the Colburn factor is very small. In contrast, for the effect of the fin pitch on the heat transfer performance, Lee et al. [6] (spiral fin), Mon and Gross [16] (annular fin) and Kim and Kim [17] (plate fin) all revealed a noticeable decline of heat transfer performance with respect to the smaller fin pitch. At the first glance, it seems that there exist some controversies amid

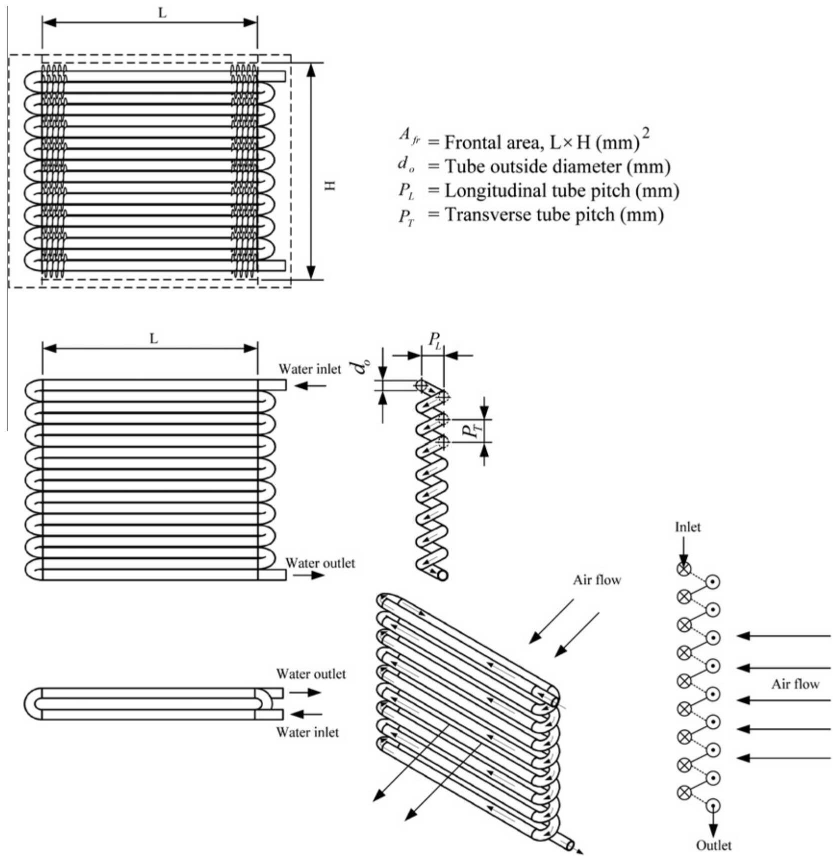

Fig. 1. Geometric details of multipass parallel-and-counter cross flow heat exchangers and water flow circuit inside the heat exchanger. N_{row} = 2 (× and • signs indicates that water flows into or out of the paper, respectively).

the present study and those studies. The difference arises from two different aspects, and an elaboration is given subsequently.

Firstly, there is a departure of the present Reynolds number and those aforementioned studies. The tests Reynolds number ($Re_{d_o} < 1000$) of previous studies were considerably lower than this study. Note that the present Reynolds number ranges approximately from 4000 to 13000. In this regard, higher operating velocities promote better mixing and lead to a better heat transfer performance, and this phenomenon prevails irrespective of change of fin pitch. Secondly, even for the Reynolds number is low (below 1000), Kim and Kim [17] found that the dependence of Colburn *j* factor on the fin pitch is rather small for a one row coil. The results are related to the boundary layer development. Kim and Kim [17] provided an analysis of the boundary layer development for the flat plate surface, and concluded that the boundary layer interruption could not occur for a fin-and-tube heat exchanger having large fin pitches, indicating the whole heat exchangers are in the developing region where the corresponding heat transfer performance is high, resulting a negligible dependence of *j* factor on the fin pitch. This is actually similar to a high operation velocity. Conversely, their 4-row coils show that the Coburn *j* increases with the rise of fin pitch at the same range of the Reynolds number. With varying fin pitches, the entrance region is in developing region while the rest are in fully developed region. The percentage of the length of developing/fully developed varies for different fin pitches, hence a detectable influence of fin pitch is encountered.

Lee et al. [6] argues that the convection heat transfer coefficient decreases with smaller fin pitches due to the boundary layer becomes thicker with a decrease in fin pitch, which can cause easier boundary layer interruption between the fins. However, this phenomenon diminishes as the high Reynolds number. Moreover, the fact that the effect of fin pitch on the heat transfer characteristics vanishes for the Reynolds numbers is above 1000 can also be found in the louvered fin-and-tube heat exchanger of Wang et al. [18]. In the meantime, Fig. 4(b) and (c) also shows that there is no significant effect of fin pitch on the friction factor for f_p = 3.2 and 4.2 mm. On the other hand, a fin pitch of 6.2 mm casts a noticeable effect on the friction factor (*f*). The friction factor for a fin pitch of 6.2 mm is higher than those having smaller fin pitch (f_p = 3.2, 4.2 mm). The reason can be seen from Eq. (8), since the friction factor depends on the dynamic effects of the ratio of minimum free-flow area with the total heat transfer area (A_{min}/A_o) and mass flux of air (G_c). Notice that A_o is significantly increased as the fin pitch is varied from 6.2 to 4.2 or 3.2 mm whereas the reduction of A_{min} is comparatively small. On the other hand, there is no significant effect of fin pitch on G_c over the entire area ratio [19]. In summary of the foregoing analysis engenders the detectable rise of friction factor for f_p = 6.2 mm.

For a comparison of the present test samples with previous studies, Fig. 4(b) and (c) also include the Wang and Chang [20], Wang et al. [21] and Briggs and Young [22] correlations for plain and circular fin-and-tube heat exchangers. It is found that crimped spiral fins shows a similar trend with those of circular fins or plate fins when f_p = 4.2 mm. However, due to the corrugated folding at the base of the present fin configuration, it appears that the crimped spiral fins give a higher friction factor (*f*) than that of plate fins at the same Reynolds number.

The effect of the material on the air-side heat transfer characteristics is shown in Fig. 5(a). The results show that there is an insignificant effect of fin material on either the Colburn factor (*j*)

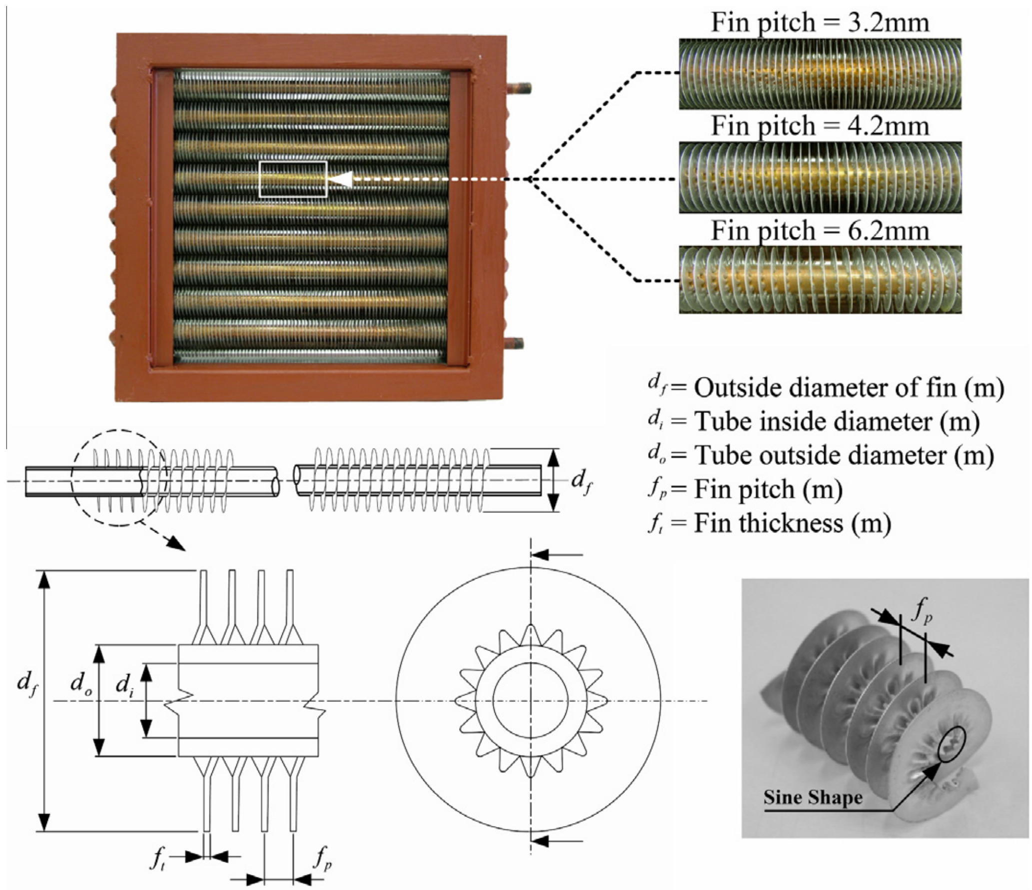

Fig. 2. The photos of the tested crimped spiral fin and tube heat exchangers (f_p = 3.2, 4.2 and 6.2 mm) and schematic diagram of crimped spiral fin.

Table 2
Detailed geometric parameters of the test samples.

No.	Fin type	d_i (mm)	d_o (mm)	d_f (mm)	A_{fr} (mm^2)	P_L (mm)	P_T (mm)	f_t (mm)	n_t	N_{row}	f_p (mm)	Fin material
1&2	Crimped	13.5	16.35	34.8	320 × 350	35	39	0.4	9	2	3.2	Al&Cu
3&4	Crimped	13.5	16.35	34.8	320 × 350	35	39	0.4	9	2	4.2	Al&Cu
5&6	Crimped	13.5	16.35	34.8	320 × 350	35	39	0.4	9	2	6.2	Al&Cu

Remarks: d_f = Outside diameter of fin; d_i = Tube inside diameter; d_o = Tube outside diameter; f_p = Fin pitch; f_t = Fin thickness; P_L = Longitudinal tube pitch; P_T = Transverse tube pitch; n_t = number of tubes in row; N_{row} = number of tube rows; A_{fr} = Frontal area (L × H).
Notes: Tube layouts of all heat exchangers are staggered layout (Al = aluminium, Cu = copper).

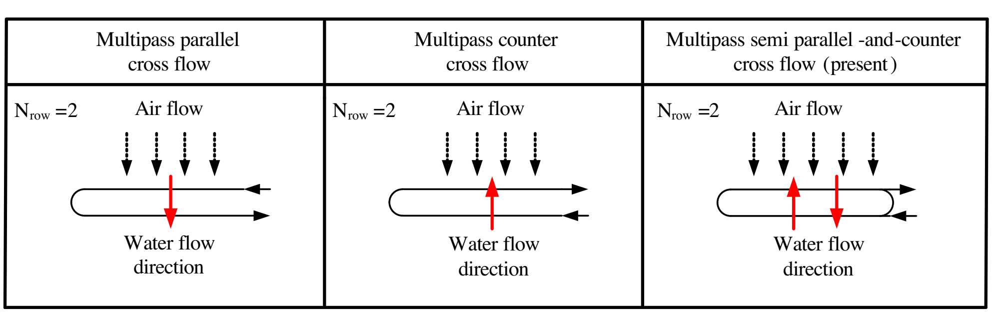

Fig. 3. Schematic diagram of the heat exchangers algorithm for multipass parallel cross flow, multipass counter cross flow and multipass parallel and counter cross-flow (N_{row} = 2).

or friction factor (f) at the same experimental condition. The results are somehow expected for the convection heat transfer performance is independent of fin materials. In contrast, it can be clearly seen that the fin efficiency (η_f) of the copper fin is higher than that of aluminium fin. The reason may be clearly explained by Eqs. (5) and (6) through which a higher thermal conductivity will result in a higher fin efficiency (η_f) accordingly. This phenomenon is confirmed in Fig. 5(b).

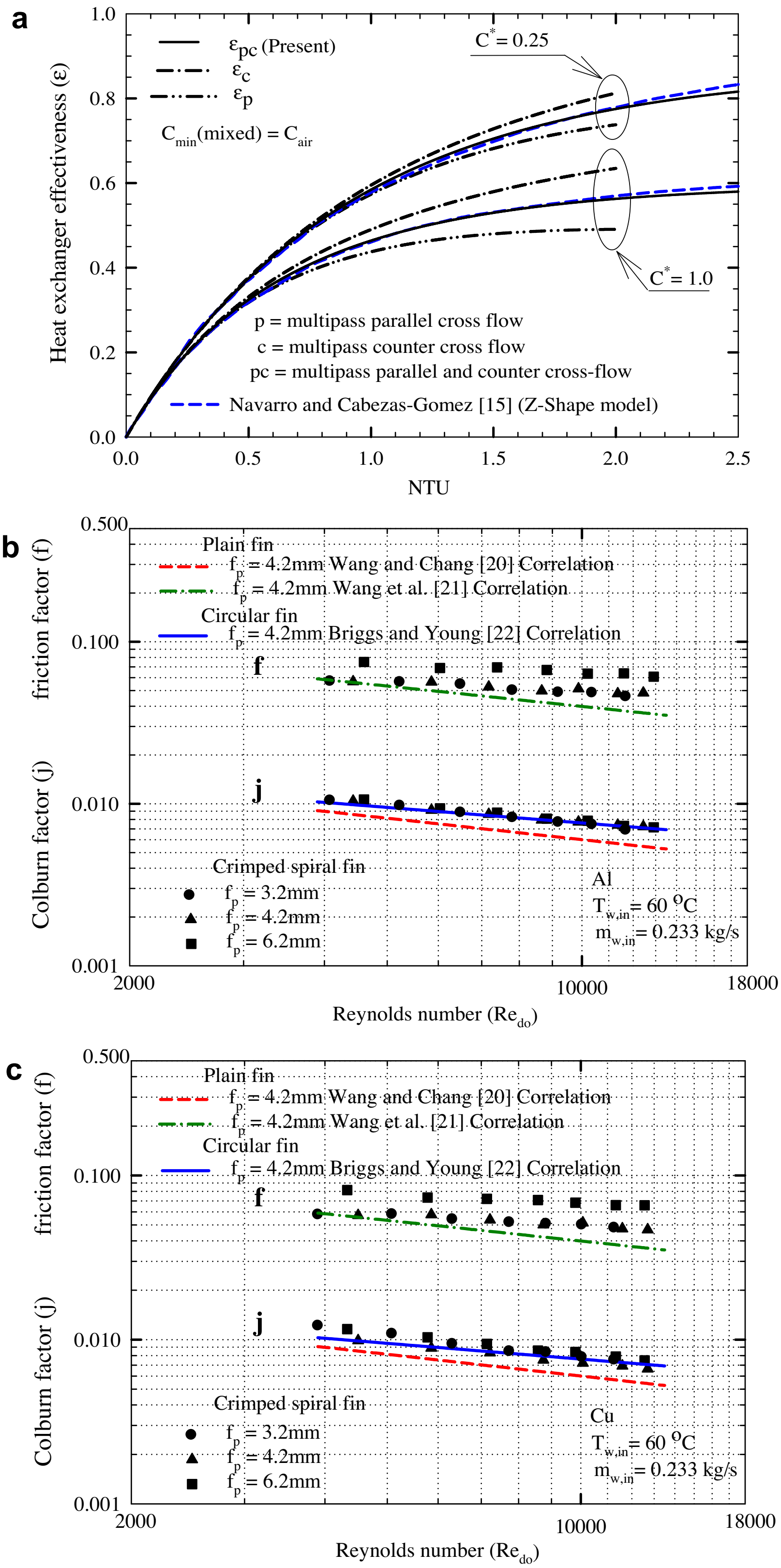

Fig. 4. The comparison chart for ε-NTU relationships graphic (a) and effect of fin pitch on the Colburn factor and friction factor for (b) aluminium fin, and (c) copper fin.

Moreover, the heat transfer rate for both fins is shown in Fig. 5(c). It is clearly seen that the heat transfer rate for copper fin is only marginally higher than that of aluminium fin due to very minor difference in fin efficiency.

4. Conclusion

This study has investigated the effect of fin pitch on the air-side performance of crimped spiral fin-and-tube heat exchangers with

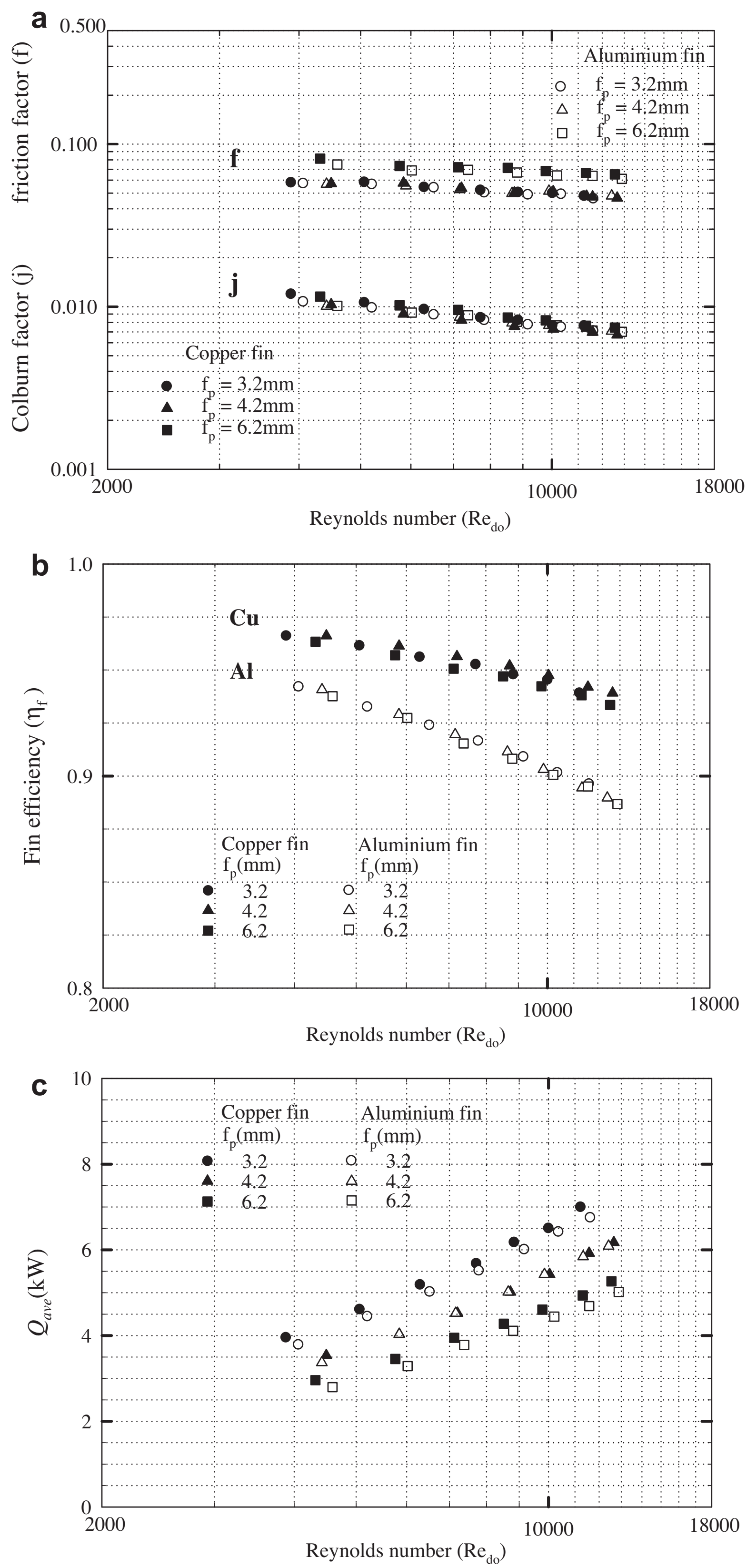

Fig. 5. Effect of fin material on the *j* and *f* factors (a), fin efficiency (b) and heat transfer rate (c) of crimped spiral fin and tube heat exchangers at $T_{w,in}$ = 65 °C and $m_{w,in}$ = 0.2 kg/s.

multipass z-shape cross-flow under sensible heating conditions with Reynolds number ranging from 4000 to 13000. A total of 6 samples were tested with associated fin materials of copper or aluminium, respectively. The associated fin thickness and outside diameter is 0.4 mm and 34.8 mm, respectively. The number of tube rows is two and fin pitches are 3.2, 4.2 and 6.2 mm, respectively. It is found that the proposed simple average effectiveness equation from the pure counter and parallel circuitry arrangement can well represent the effectiveness-NTU relationship for the current z-shape arrangement. Based on the test results, it is found that the effect of fin pitch on the Coburn j factor is insignificant. This is because the high Reynolds number accentuates good mixing, leading to a better heat transfer performance irrespective changes of fin pitch. In the meantime, there is negligible difference for friction factor amid $f_p = 3.2$ and 4.2 mm. However, a detectable increase of friction factor is seen when the fin pitch is increased to 6.2 mm. Moreover, it could be clearly seen that the airside performance is independent of fin material.

Acknowledgments

The authors are indebted to King Mongkat's University of Technology Thonburi, the Thailand Research Fund, the Office of the Higher Education Commission and the National Research University Project for supporting this study. The third author also expresses gratitude for supporting funding from the National Science Council of Taiwan (99-2218-E-009-012-MY2).

References

[1] A. Nuntaphan, T. Kiatsiriroat, C.C. Wang, Air side performance at low Reynolds number of cross-flow heat exchanger using crimped spiral fins, Int. Commun. Heat Mass Transfer 32 (2005) 151–165.

[2] P. Naphon, S. Wongwises, Investigation of the performance of a spiral-coil finned tube heat exchanger under dehumidifying conditions, J. Eng. Phys. Thermophys. 76 (2003) 83–92.

[3] S. Wongwises, P. Naphon, Heat transfer characteristics of a spiral-coil finned tube heat exchanger under dry-surface conditions, Heat Transfer Eng. 27 (2006) 25–34.

[4] S. Wongwises, P. Naphon, Thermal performance of a spiral-coil finned tube heat exchanger under wet-surface conditions, J. Mech. Sci. Technol. 20 (2006) 212–226.

[5] K. Srisawad, S. Wongwises, Heat transfer characteristics of a new helically coiled crimped spiral finned tube heat exchanger, Heat Mass Transfer 45 (2009) 381–391.

[6] M. Lee, T. Kang, Y. Kim, Air-side heat transfer characteristics of spiral-type circular fin-and-tube heat exchangers, Int. J. Refrig. 33 (2010) 313–320.

[7] G. Xie, Q. Wang, B. Sunden, Parametric study and multiple correlations on air-side heat transfer and friction characteristics of fin-and-tube heat exchangers with large number of large-diameter tube rows, Appl. Therm. Eng. 29 (2009) 1–16.

[8] S. Wongwises, Y. Chokeman, Effect of fin pitch and number of tube rows on the air side performance of herringbone wavy fin and tube heat exchangers, Energy Convers. Manage. 46 (2005) 2216–2231.

[9] ESDU 86018, Effectiveness-NTU Relations for the Design and Performance Evaluation of two-stream heat exchangers, Engineering Science Data Unit 86018 with Amendment, London ESDU International plc (1991) 92-107.

[10] H.A. Navarro, L. Cabezas-Gomez, A new approach for thermal performance calculation of cross-flow heat exchanger, Int. J. Heat Mass Transfer 48 (2005) 3880–3888.

[11] L. Cabezas-Gomez, H.A. Navarro, J.M. Saiz-Jabardo, Thermal performance of multipass parallel and counter-cross-flow heat exchangers, J. Heat transfer 129 (2007) 282–290.

[12] K.A. Gardner, Efficient of extended surface, ASME Trans. 67 (1945) 621.

[13] W.M. Kays, A. London, Compact Heat Exchangers, third ed., McGraw-Hill, New York, 1984.

[14] ANSI/ASHRAE Standard 33-2000, Method of Testing Forced Circulation Air Cooling and Air Heating Coils, 2000.

[15] H.A. Navarro, L. Cabezas-Gomez, Effectiveness-NTU computation with a mathematical model for cross-flow heat exchangers, Brazilian J. Chem. Eng 24 (4) (2007) 509–521.

[16] M.S. Mon, U. Gross, Numerical study of fin-spacing effects in annular-finned tube heat exchangers, Int. J. Heat Mass Transfer 47 (2004) 1953–1964.

[17] Y.H. Kim, Y. Kim, Heat transfer characteristics of flat plate finned-tube heat exchangers with large fin pitch, Int. J. Refrig. 28 (2005) 851–858.

[18] C.C. Wang, K-Y. Chi, Y-J. Chang, Y-P. Chang, An experimental study of heat transfer and friction characteristics of typical louver fin-and-tube heat exchangers, Int. J. Heat Mass Transfer 41 (1998).

[19] K. Torikoshi, G.N. Xi, Y. Nakazawa, H. Asano, Flow and heat transfer performance of a plate fin and tube heat exchanger (First Report: Effect of Fin Pitch), Heat Transfer 1994, in: Proceedings of the 10th Int. Heat Transfer Conf. 4, 1994, pp. 411–416.

[20] C.C. Wang, C-T. Chang, Heat and mass transfer for plate fin-and-tube heat exchangers with and without hydrophilic coating, Int. J. Heat Mass Transfer 41 (1998) 3109–3120.

[21] C.C. Wang, Y.J. Chang, Y.C. Hsieh, Y.T. Lin, Sensible heat and friction characteristics of plate fin-and-tube heat exchangers having plane fins, Int. J. Refrig. 19 (4) (1996) 223–230.

[22] D.E. Briggs, E.H. Young, Convective heat transfer and pressure drop of air flowing across triangular pitch banks of finned tubes, Chem. Eng. Prog. Symp. Ser. 59 (41) (1963) 1–10.