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Technical Note

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

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ABSTRACT

The air-side performance of crimped spiral fin and tube heat exchangers at high Reynolds number (3000–13,000) is investigated in this study. The test heat exchangers have a new type of multipass parallel and counter cross-flow water flow arrangement which is a combination of parallel cross-flow and counter cross-flow. The test samples are made from copper and aluminium with different number of tube rows (N_{row} = 2, 3, 4 and 5). The effects of number of tube rows and fin material on the heat transfer and friction characteristics are studied. The results show that no significant effect for either number of tube rows or fin materials on the heat transfer performance is found at high Reynolds number. In addition, the correlation of the air-side performances of this type of the heat exchangers at high Reynolds number is developed for industrial applications.

1. Introduction

The heat exchanger is a basic component that is used for thermal systems in many industrial processes involving heat transfer. The most favourable type of heat exchanger used in industrial applications is the fin-and-tube heat exchanger. It is very important to consider the heat transfer rate, which is normally limited by the thermal resistance on the air-side of the heat exchanger. One way to augment the heat transfer rate on the air-side of the heat exchanger is to improve the fin geometry. However, this method may lead to the need for increased fan power because of the penalty associated with the pressure drop. Many researchers had studied the effect of the plate fin geometry, such as plain fin, slit fin, wavy fin, louvered fin, compounded fin, etc., on the heat transfer performance and frictional characteristic. On the other hand, there are only few researches on the crimped spiral fin-and-tube heat exchanger as found in [1–4].

Nuntaphan et al. [1] studied the air-side of crimped spiral fin heat exchanger to analyse the effect of tube diameter, fin spacing, transfer tube pitch, and tube arrangement and proposed correlations between heat transfer and friction characteristics in the case of low Reynolds numbers. Moreover, heat exchangers using crimped spiral finned tubes were investigated by Srisawad and Wongwises [2] who focused on the air-side performance of helically coiled crimped spiral finned tube heat exchangers in dry-surface conditions. Pongsoi et al. [3] studied the effect of fin pitch on the air-side heat transfer characteristics of crimped spiral fin-tube heat exchangers having multipass parallel and counter cross-flow under sensible heating conditions and purpose the ε–*NTU* relation equation for this new water flow arrangement. Tang et al. [4] studied the air-side heat transfer and friction characteristics of five kinds of fin-and-tube heat exchangers i.e. crimped spiral fin, plain fin, slit fin, fin with delta-wing longitudinal vortex generators, and mixed fin. The number of tube rows (N_{row}) was 12. The diameter of tubes (d_o) was 18 mm. Reynolds number was varied from 4000 to 10,000. According to their results, although the crimped spiral fin gave high pressure drop, it provided higher air-side heat transfer performance than several types of heat exchangers.

According to these literatures, crimped spiral fin is proved to be quite reliable in industrial applications. The crimped spiral fin

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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
Gc	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_{do}	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_{do}(3000–13,000)
Inlet-water temperature, °C	55–70
Water flow rate, LPM	12–14

features a sine-shaped fin which gives rise to a higher heat transfer rate on the air-side through the increase in the contact surface between the fin base and tube. However, up to now, none of the experimental investigations found in the literature had reported the effect of the number of tube rows on the air-side performance of crimped spiral fin-and-tube heat exchangers. Upon the foregoing studies, Nuntaphan et al. [1] is the only experimental work that examined the effect of fin geometry and tube arrangement on the air-side performance of crimped spiral fin-and-tube heat

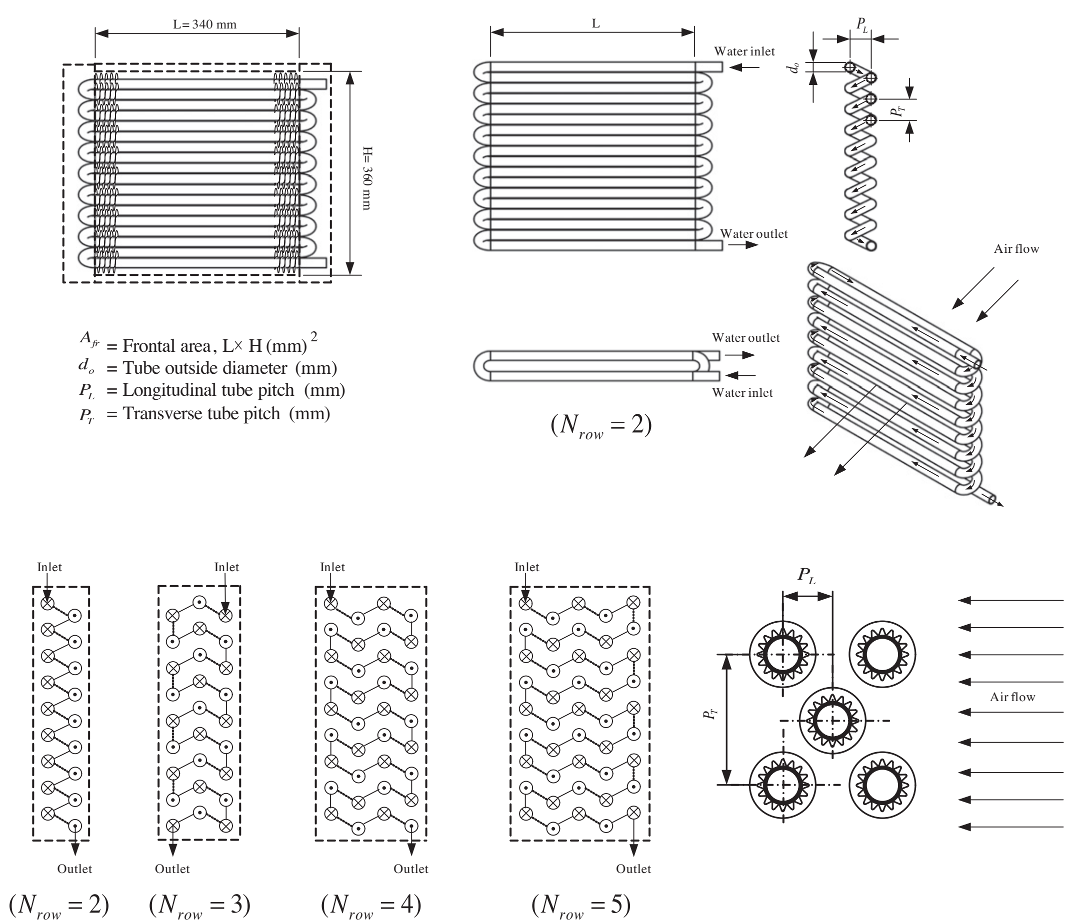

Fig. 1. Geometric details of multipass parallel-and-counter cross flow heat exchangers and water flow circuit inside the heat exchanger. N_{row} = 2, 3, 4 and 5 (× and • signs indicates that water flows into or out of the paper, respectively).

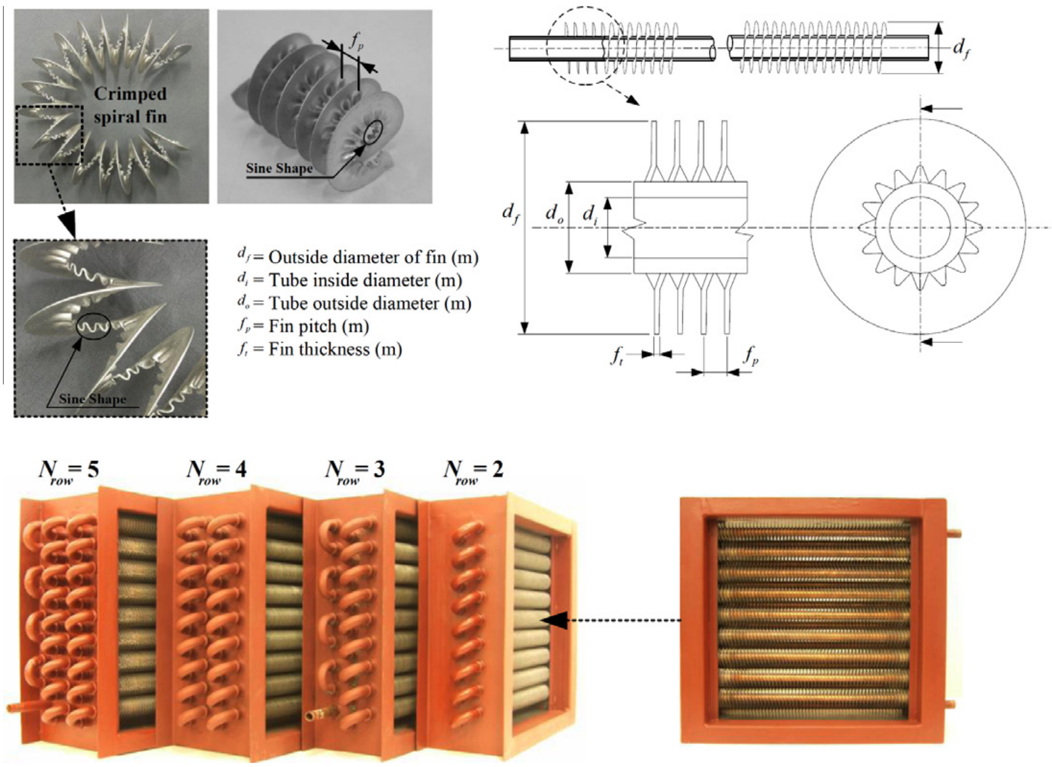

Fig. 2. The photos of the tested crimped spiral fin and tube heat exchangers and schematic diagram of crimped spiral fin.

exchangers. However, their tests were conducted at a very low air frontal velocity (0.5–1.5 m/s) which limited the applicability of the test results. In practice, especially in industrial service, the operation velocity is normally much higher as reported by Xie et al. [5]. Therefore, it is the main purpose of this study is to extend the applicable range (V_{fr} up to 6 m/s) of the crimped spiral fin-and-tube heat exchangers subject to the influence of the number of tube rows and fin materials. Moreover, the test samples are of this study is of large diameter tubes (about 16 mm) which are very popularly used in ventilator and fan-coil units [6,7].

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:

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

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. Therefore, the ε–NTU relationships developed by Pongsoi et al. [3] is selected to calculate the heat exchanger effectiveness. This relation combines both effect of parallel and counter cross-flow configuration of water flow arrangement from [9–11], as shown in Eqs. (2)–(9):

For multipass parallel cross-flow with N_{row} = 2, 3, 4 and 5:

$$(N_{row} = 2), \quad \varepsilon_P\left(1 - \frac{K}{2}\right)\left(1 - e^{-2K/C_A^*}\right), \quad K = 1 - e^{-NTU_A(C_A^*/2)} \tag{2}$$

Table 2
Detailed geometric parameters of the test samples.

No.	Fin type	d_i (mm)	d_o (mm)	d_f (mm)	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	35.0	35	40	0.5	9	3, 4	6.3	Cu
3–6	Crimped	13.5	16.35	35.0	35	40	0.5	9	2, 3, 4, 5	6.3	Al

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. Tube layouts of all heat exchangers are staggered layout (Al = aluminium, Cu = copper).

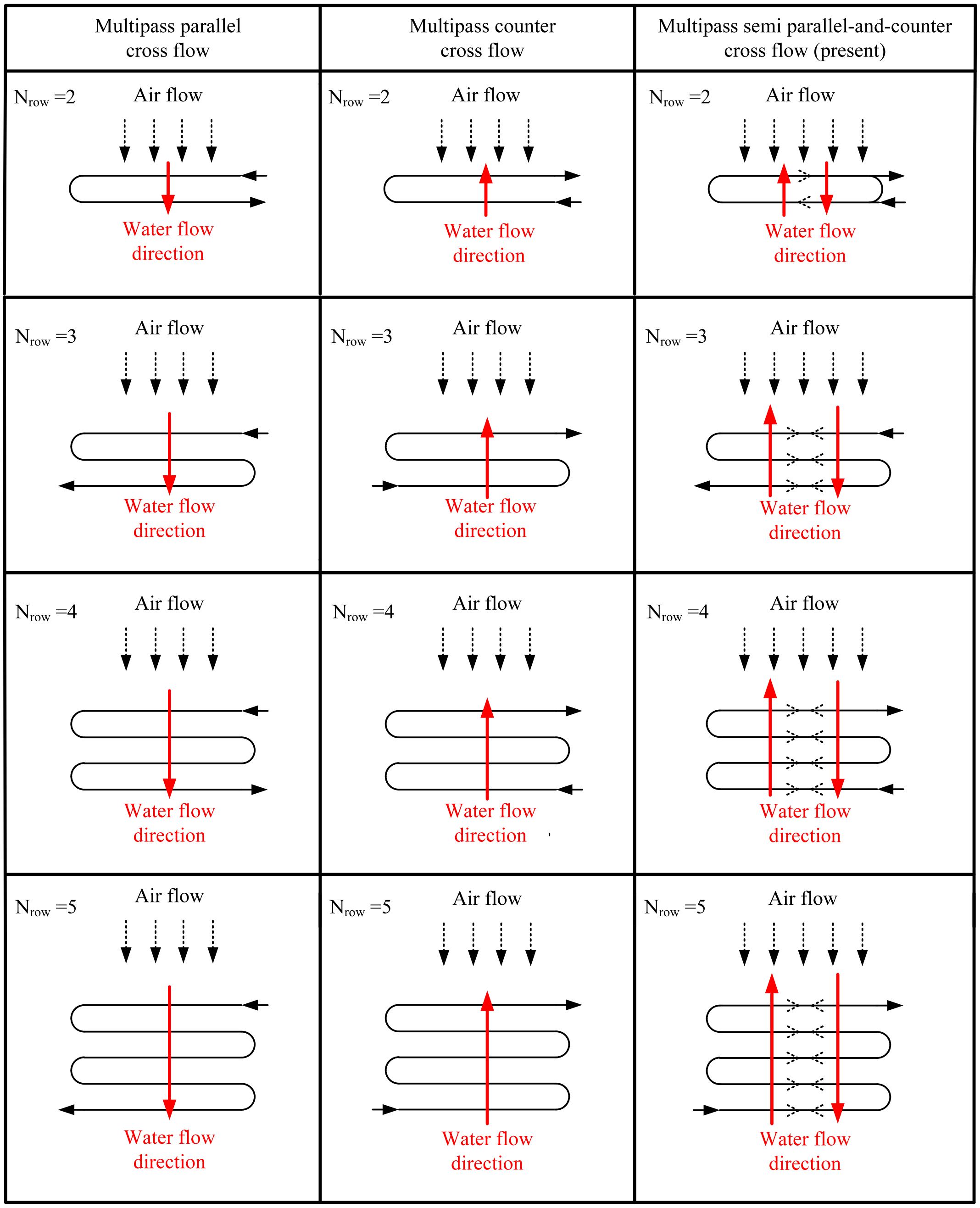

Fig. 3. Schematic diagram of the heat exchangers algorithm for multipass parallel cross flow, multipass counter cross flow and multipass parallel and counter cross-flow (N_{row} = 2, 3, 4 and 5).

$$(N_{row}=3),\quad \varepsilon_P = 1-\left(1-\frac{K}{2}\right)^2 e^{-3K/C_A^*} - K\left[1-\frac{K}{4}+\frac{K}{C_A^*}\left(1-\frac{K}{2}\right)\right]e^{-K/C_A^*},\quad K=1-e^{-NTU_A(C_A^*/3)} \tag{3}$$

$$(N_{row}=4),\quad \varepsilon_P = 1-\frac{K}{2}\left(1-\frac{K}{2}+\frac{K^2}{4}\right)-K\left(1-\frac{K}{2}\right)\left(1+2\frac{K}{C_A^*}\left[1-\frac{K}{2}\right]\right)e^{-2K/C_A^*}-\left(1-\frac{K}{2}\right)^3 e^{-4K/C_A^*},\quad K=1-e^{-NTU_A(C_A^*/4)} \tag{4}$$

$$(N_{row}=5 \text{ or } \infty),\quad \varepsilon_P = \frac{1-e^{-NTU_A(1+C_A^*)}}{1+C_A^*} \tag{5}$$

For multipass counter cross-flow with N_{row} = 2, 3, 4 and 5:

$$(N_{row}=2),\quad \varepsilon_c = 1-\left[\frac{K}{2}+\left(1-\frac{K}{2}\right)e^{2K/C_A^*}\right]^{-1},\quad K=1-e^{-NTU_A(C_A^*/2)} \tag{6}$$

$$(N_{row}=3),\quad \varepsilon_c = 1-\left\{\left(1-\frac{K}{2}\right)^2 e^{3K/C_A^*} + k\left[K\left(1-\frac{K}{4}\right)-\left(1-\frac{K}{2}\right)\frac{K^2}{C_A^*}\right]e^{K/C_A^*}\right\}^{-1},\quad K=1-e^{-NTU_A(C_A^*/3)} \tag{7}$$

$$(N_{row}=4),\quad \varepsilon_c = 1-\left\{\frac{K}{2}\left(1-\frac{K}{2}+\frac{K^2}{4}\right)+K\left(1-\frac{K}{2}\right)\times\left[1-2\frac{K}{C_A^*}\left(1-\frac{K}{2}\right)\right]e^{2K/C_A^*}+\left(1-\frac{K}{2}\right)^3 e^{4K/C_A^*}\right\}^{-1},\quad \times K=1-e^{-NTU_A(C_A^*/4)} \tag{8}$$

$$(N_{row} = 5 \text{ or } \infty), \quad \varepsilon_c = \frac{1 - e^{-NTU_A(1-C_A^*)}}{1 - C_A^* e^{-NTU_A(1-C_A^*)}} \tag{9}$$

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

$$\varepsilon_{pc} = \frac{\varepsilon_P + \varepsilon_C}{2} \quad \text{for } N_{row} = 2, 3, 4 \text{ and } 5 \tag{10}$$

The schematic diagram of circuitry arrangement for N_{row} = 2, 3, 4 and 5 are shown in Fig. 3.

The heat transfer performance is calculated by the method of Kays and London. 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.,

$$\eta_f = \frac{2\psi}{\phi(1+\psi)} \frac{I_1(\phi R_o)K_1(\phi R_i) - I_1(\phi R_i)K_1(\phi R_o)}{I_0(\phi R_i)K_1(\phi R_o) + I_1(\phi R_o)K_0(\phi R_i)} \tag{11}$$

where

$$\phi = (r_o - r_i)^{3/2} \left(\frac{2h_o}{k_f A_p}\right)^{1/2} \tag{12}$$

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

$$j = \frac{Nu}{Re_{do} \mathrm{Pr}^{1/3}} = \frac{h_o}{\rho_a V_{\max} c_P} (\mathrm{Pr})^{2/3} \tag{13}$$

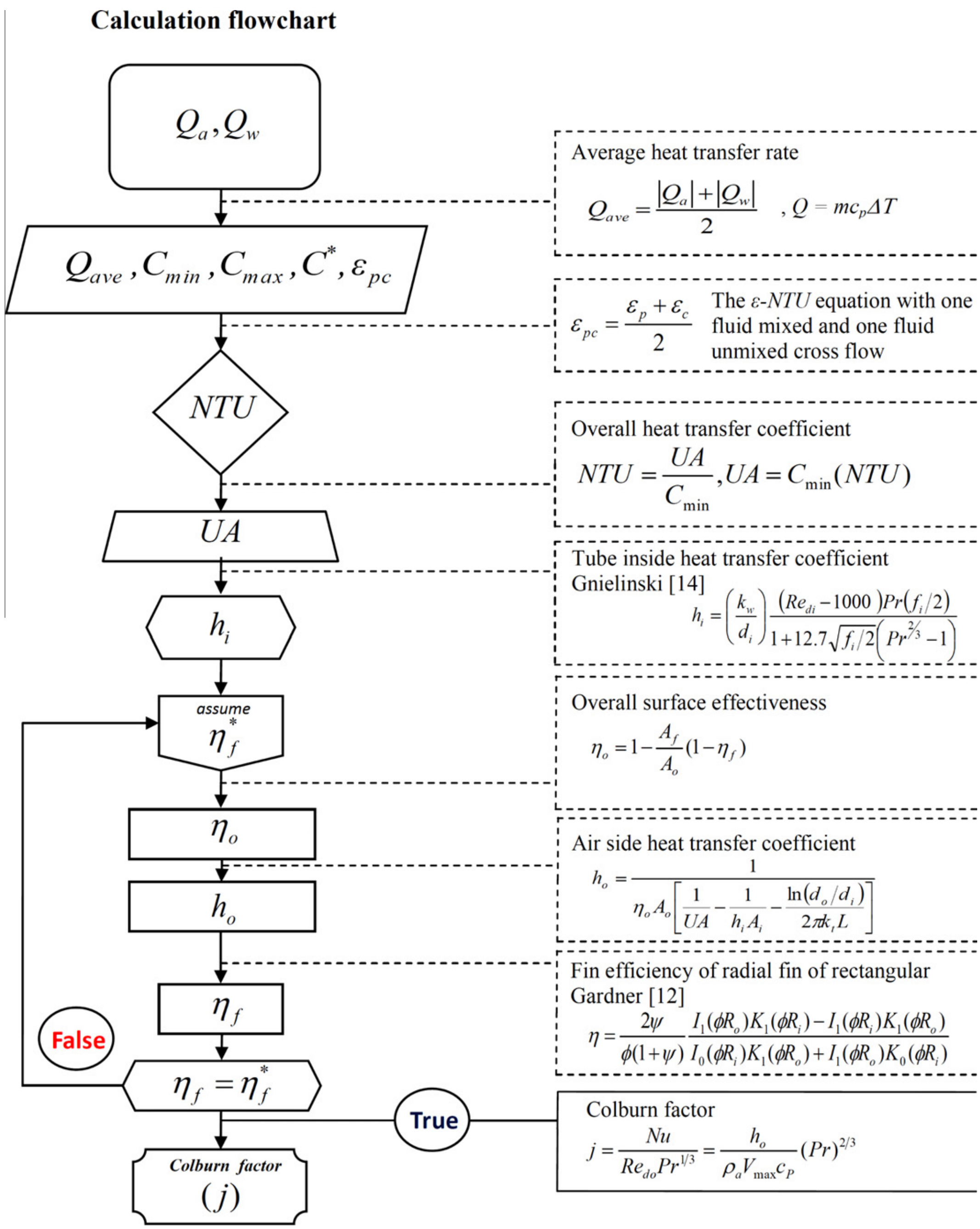

Fig. 4. Flowchart of the data reduction for air-side heat transfer performance (j-factor). (See above-mentioned reference for further information.)

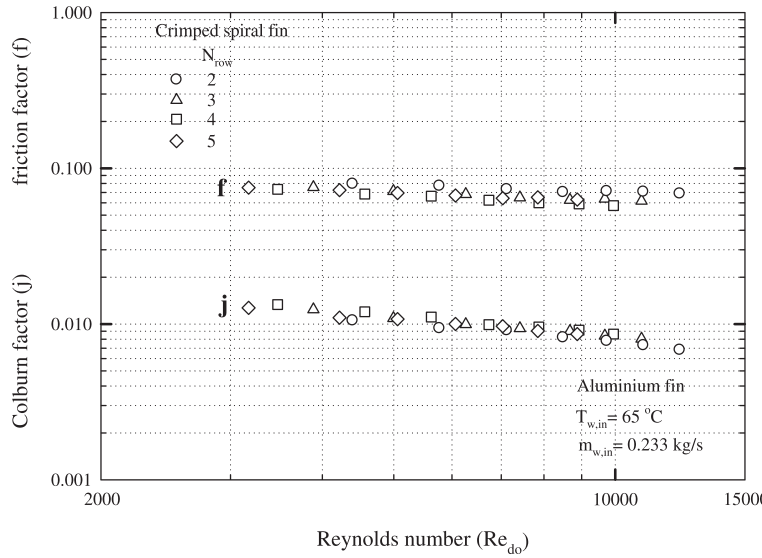

Fig. 5. Effect of number of tube rows on the Colburn factor and friction factor of crimped spiral fin and tube heat exchangers.

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

$$f = \left(\frac{A_{\min}}{A_o}\right)\left(\frac{\rho_m}{\rho_1}\right)\left[\frac{2\Delta P\rho_1}{G_c^2} - (1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right)\right] \tag{14}$$

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 flow chart showing the reduction of the data is shown in Fig. 4.

The experiments are conducted following the ANSI/ASHRAE 33 Standards [15] 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 3.5% for the Re_{do}, 13.0% for the j-factor and 11.5% for f-factor. The uncertainties decreased as Reynolds number increased.

3. Results and discussion

In experiments, the analysis of air-side heat transfer and friction characteristics of all the heat exchangers examined are presented in terms of a dimensionless number including the Colburn factor (j) and friction factor (f) plotted against the Reynolds number (Re_{do}) and the Reynolds number based on the outside diameter of the tube. As expected, both the Colburn factor (j) and the friction factor (f) decrease an increasing Reynolds number.

Fig. 5 shows the effect of the number of tube rows on the heat transfer and friction characteristic. The number of tube row of the heat exchangers are from 2 to 5. As can be seen in Fig. 5, the number of tube rows casts no significant effect on the Colburn factor (j) at high Reynolds numbers (3000–13,000). This phenomenon is similar to the plain fin data as reported by Rich [16] and Wang et al. [17] in which the effect of the number of tube rows diminishes as the Reynolds number is increased over 2000. This is due to the downstream turbulence eddies shed from the tubes that

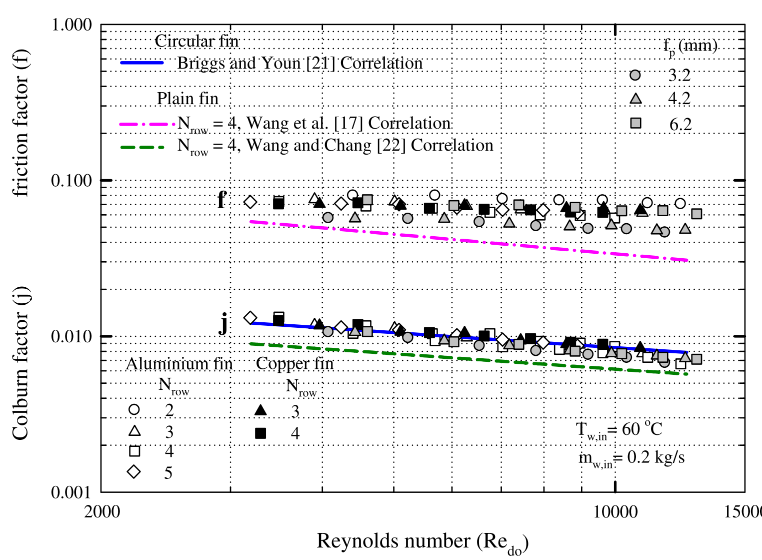

Fig. 6. The comparison of j and f factors between presented data with correlation of plain fin and circular fin.

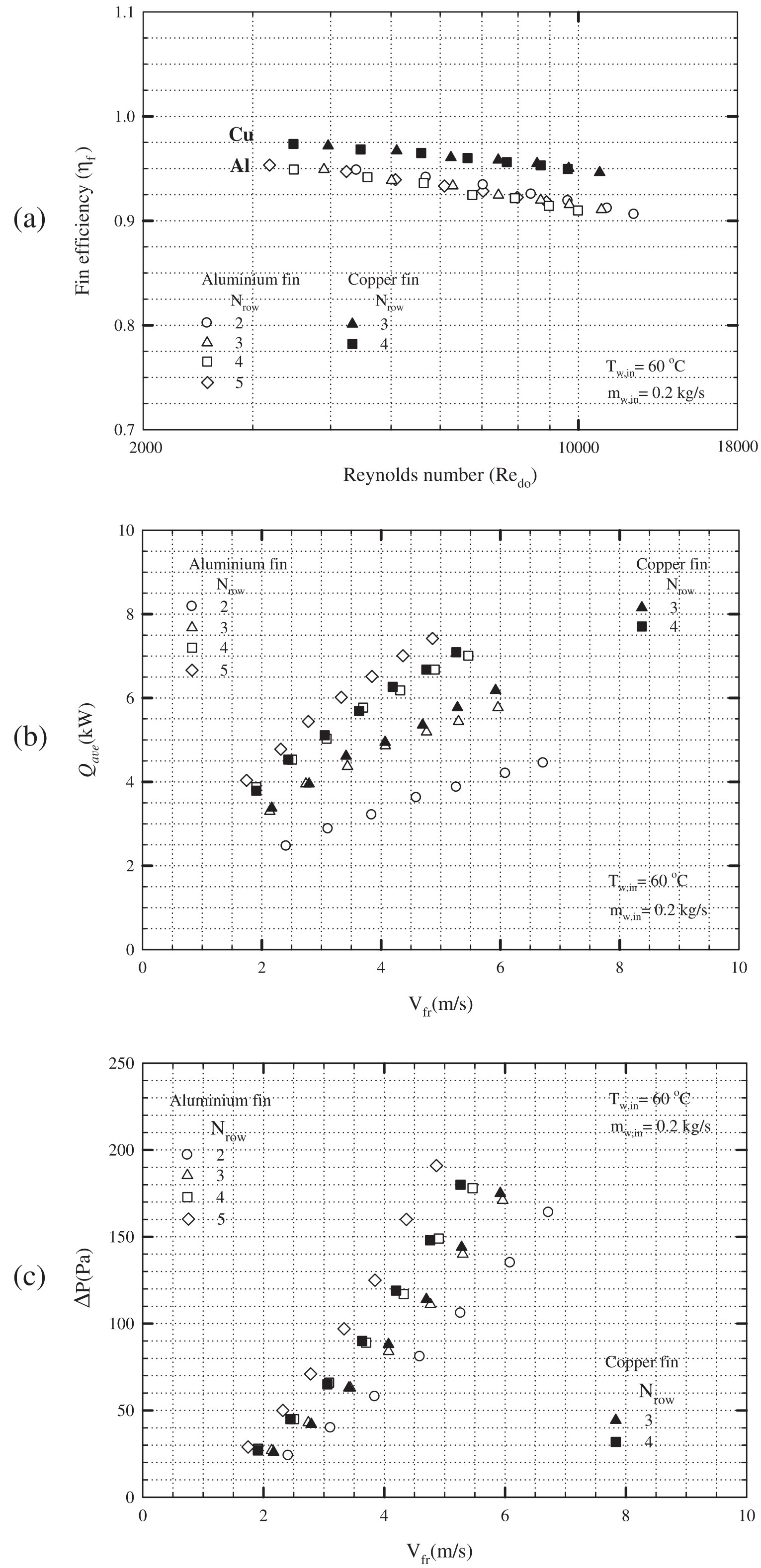

Fig. 7. Effect of number of tube rows on the fin efficiency (a), heat transfer rate (b) and pressure drop (c) of crimped spiral fin and tube heat exchangers (N_{row} = 2, 3, 4 and 5).

cause good mixing in the downstream fin region. Moreover, the fact that the effect of the number of tube rows on the heat transfer characteristics vanishes for Reynolds numbers over 2000 can also be found in the works of [18–20] with fin-and-tube heat exchangers having wavy, louver and slit geometry, respectively. The data point in Fig. 5 shows that the number of tube row (N_{row}) does not affect the friction factor (f) over the range of Reynolds numbers being examined.

For a comparison of the present test samples with previous studies, the Briggs and Young [21] correlation for circular fin-and-tube heat exchanger is chosen to compare with the present experimental results as illustrated in Fig. 6. Note that the configuration of circular fin is very similar to that of the tested crimped spiral fin. Therefore, the calculated Colburn factor (j) of circular fins reveals a similar trend and magnitude as the present crimped spiral fins. Also, the Colburn factor (j) and friction factor (f) correlations of plain fin-and-tube heat exchangers, which is proposed by Wang and Chang [22] and Wang et al. [17], respectively, are selected to compare with the tested results. As shown in Fig. 6, it is found that crimped spiral fins have a similar trend with the plain fins as well. However, due to the corrugated folding at the base of the present fin configuration, it appears that crimped spiral fins give a higher friction factor (f) than that plain fins at the same Reynolds number. Moreover, as shown in Fig. 6, there is no significant effect of fin material on both the Colburn factor (j) and friction factor (f).

Fig. 7(a) illustrated the effect of the number of tube rows on the fin efficiency (η_f) of both fin material. It can be seen that no significant effect of number of tube rows is found under the same experimental conditions. The reason may be explained by Eqs. (11) and (12), which show that an increase in the number of tube rows has no significant differences in terms of ϕ and fin efficiency (η_f) at the same Reynolds number because the fin efficiency depends only on size, shape and material of fin. In the meantime, it can be clearly seen that the fin efficiency (η_f) of the copper fin is higher than that of aluminium fin due to a higher thermal conductivity of copper.

Moreover, the heat transfer rate for both fins is shown in Fig. 7(b). It seems that the heat transfer rate for copper fin only marginally higher than that of aluminium fin due to its very minor difference in fin efficiency. On the other hand, the number of tube rows had significant effect on average heat transfer rate at the same air frontal velocity.

A comparison of the results for the effect of the number of tube rows on the pressure drop of crimped spiral fin-and-tube heat exchangers is presented in Fig. 7(c). As seen, the pressure drop increases with the frontal air velocity while the friction factor decreases with increasing Reynolds number. It can be seen that the pressure drop for $N_{row} = 5$ is two times higher than those for $N_{row} = 2$. The reason for this result can be explained by the increase in the blocking flow area when N_{row} is increased.

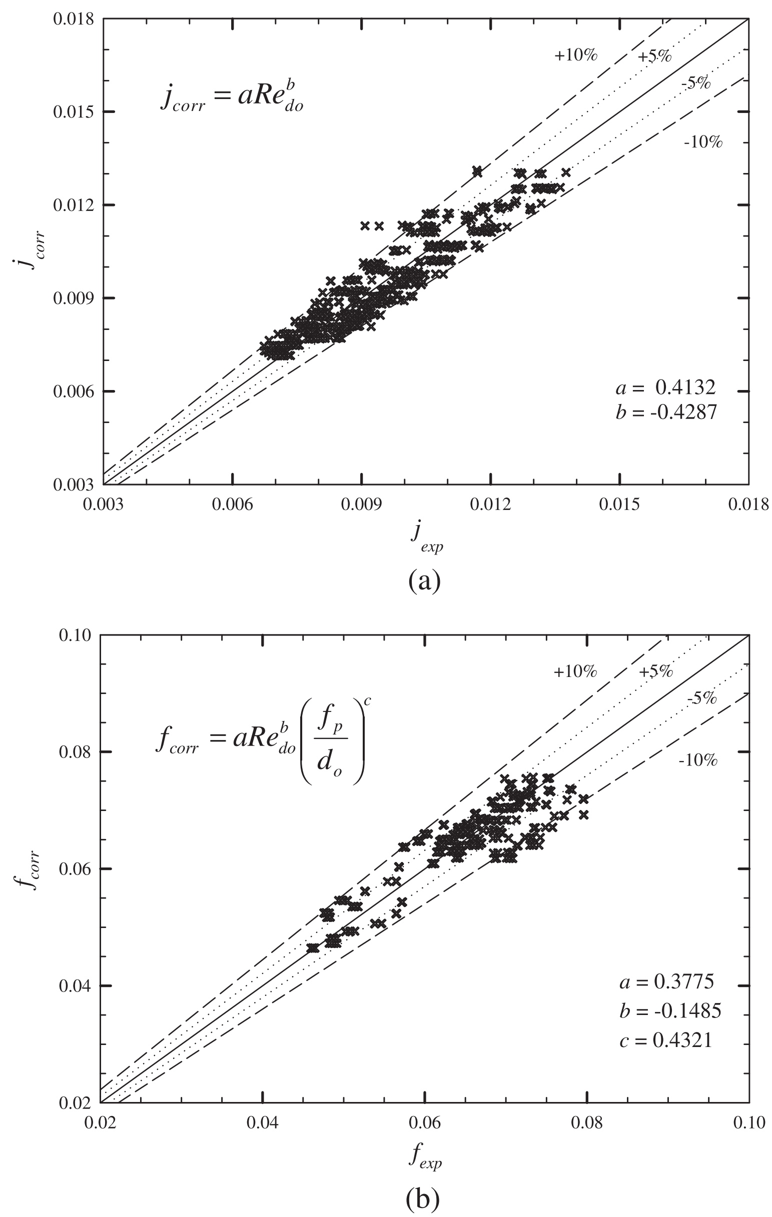

Fig. 8. Comparison of the proposed correlations with experimental data: (a) Colburn factor and (b) friction factor.

To determine the heat transfer coefficient of crimped spiral fin-and-tube heat exchangers, an empirical correlation between the Colburn factor (j) and the friction factor (f) is proposed in this work. The correlations may be valuable for industrial usage in order to design the heat exchangers operating at air flows of high Reynolds number.

An empirical correlation for j and f factors for crimped spiral fin-and-tube heat exchanger is developed in this work. The correlation also includes the experimental data used to study the effect of fin pitch as reported in Pongsoi et al. [3]. It must be noted that number of tube rows cast negligible influence on both j and f factors as foregoing discussions. As a result, the number of tube row is not included in the correlations. The test results are correlated as suggested by Wang et al. [23] in the form of $j = aRe_{do}^{b}$ and $f = aRe_{do}^{b}$ where a and b are the empirical constants obtained from least-square fitting of the experimental data. Furthermore, based on the data analysis, the f factor of crimped spiral fin-and-tube heat exchanger were also proportional to the fin pitch. Therefore, the present correlation for f factor is modified by including the dimensionless fin pitch (f_p) normalized by the tube outside diameter (d_o). The final correlation for the j and f factors takes the following form:

$$j_{corr} = 0.4132Re_{do}^{-0.4287} \tag{15}$$

$$f_{corr} = 0.3775Re_{do}^{-0.1485}\left(\frac{f_p}{d_o}\right)^{0.4321} \tag{16}$$

$$\text{Mean deviations} = \frac{1}{M}\left[\sum_{1}^{M}\frac{|\Phi_{corr} - \Phi_{exp}|}{\Phi_{exp}}\right] \times 100\% \tag{17}$$

As shown in Fig. 8, the proposed heat transfer and friction correlations Eqs. (15) and (16) can describe 95.04% and 91.07% of the j and f factors to be within ±10%, respectively. The mean deviations Eq. (17) of the proposed heat transfer and friction correlations are 5.09% and 4.53%, respectively. However, Eq. (15) and (16) can be used only (N_{row} = 2–5), (f_p = 3.2–6.3 mm) when staggered arrangement.

4. Conclusion

This study has investigated the effect of number of tube rows on the air-side performance of crimped spiral fin and tube heat exchangers having multipass parallel and counter cross-flow under sensible heating conditions at high Reynolds number (3000–13,000) based on the tube outside diameter. The following conclusions based on the tests are obtained:

– The number of tube rows cast a negligible influence on the air-side heat transfer performance at high Reynolds number (3000–13,000). Moreover, it could be clearly seen that the airside performance is independent of fin material.
– The average heat transfer rate and pressure drop increase with increasing the number of tube rows.
– The correlation of Colburn factor (j) and friction factor (f) of crimped spiral fin-and-tube heat exchangers at high Reynolds number are proposed in this paper. The mean deviations of the proposed heat transfer and friction correlations are 5.09% and 4.53%, respectively.

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