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A Numerical Investigation of Compact Serrated Spiral Fin Heat Exchanger for Circular Tubes with Twisted Tap Inserts

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Abstract: This study presents the airside performance of fin and tube heat exchangers for perforated serrated spiral fin with twisted tap inserts, this research focus on heat and fluid flow analysis by various fin pitches. A steady –state three-dimensional numerical model is going to be used to study the heat transfer and pressure drop characteristics of spiral fin on the tube surfaces with Reynolds number in the range of 1000-1600.A numerical study is going to be perform on compact fin and tube heat exchanger having circular tube with serrated spiral fins. The air side performance for heat exchanger with combination of with hydrophilic coating and without hydrophilic coating on a fin and different fin pitches and also different twist angle of fins. The increase in the heat transfer performance will be analyzed for various fin pitches of 2mm to 3mm with combined coating. The effect of number of tube rows, fin pitch on airside performance will be analyzed.
Keywords: Fin Pitch, Serrated fin, Pressure drop, Heat Exchanger, Fin Thickness, Twisting Angle, Hydrophilic Coating

I. INTRODUCTION

The normal function of a spiral fin-and-tube heat exchanger is to transfer heat from one fluid to another. The basic component of a spiral fin-and-tube heat exchanger can be viewed as a tube with one fluid running through it and another fluid flowing by on the outside. There are thus two modes of heat transfer that need to be described: convective heat transfer from fluid to the inner wall of the tube, conductive heat transfer through the tube wall, and convective heat transfer from the outer tube wall to the outside fluid. Spiral fin-and-tube heat exchangers are typically classified according to fin configurations and tube arrangements, as shown in Pongsoi et al [2]. Generally for this kind of the spiral fin-and-tube heat exchanger, the dominant thermal resistance is on the air-side. Decreasing air side thermal resistance can be done by increasing air speed, air turbulence, and heat transfer area. Moreover, a more efficient and compact heat exchanger of the spiral fin-and-tube heat exchanger can be done via enhanced fin geometry. There have been many researches related to the experimental study of the serrated spiral-finned tube heat exchangers. On the other hand, the only paper to describe the Vortex generation characteristics of serrated finned tube banks with staggered arrangement 1st report by K. Kawaguchi. They presented the influence of fin pitch on the heat transfer performance and friction characteristic for perforated serrated spiral fin-and-tube heat exchangers by varying the fin pitches of 1.8 and 3.5 mm. It was confirmed that the fin pitch increases with decreasing heat transfer coefficients and pressure drops at the same Reynolds number. The experimental results given by Parinya Kiatpachai, the maximum heat transfer at a fin pitches of 2-3.5mm.

Over the years, most of the studies have focused on the flow characteristics of air flowing through the serrated fins while the heat transfer performance has received comparatively little attention. Very few studies with heat transfer performance are available. However, although some information is currently available, there still remains room for further research. Up to now, there has been only one work, carried out by Parinya Kiatpachai [2], dealing with the effects of the fin pitch. However, their study focused only on serrated spiral fins at two different fin pitches. In real application, serrated fins can be perforated with tubes. With this configuration, serrated spiral fins can reduce the thermal resistance between bases of fins with tube surfaces. In the present study, the effect of fin pitch of a serrated spiral fin-and-tube heat exchanger at high Reynolds number on heat transfer performance, which has never before appeared in open literature, is presented. Both frontal velocity and fin pitches used in the present study cover the range of real applications and real manufacturing

II. DATA REDUCTION

In this study, the perforated serrated spiral fin-and-tube heat exchangers having Z shape tube arrangements and 2 tube rows are investigated by using experimental method to measuring the inlet and outlet water temperature using thermometer, flow rate, including the water flow loop, air flow supply, water suction pump and data acquisition system. The detailed geometrical parameters of the test section are shown according to table 1.The test samples are made from copper tube and Aluminium fin.The working fluids are ambient air and hot water for air-side and water side, respectively. The experimental conditions are shown in the table 2 and temperature and water flow rates are fixed while varying ambient air flow rate in range of 200-1600A from Parinya kiatpachi (2). The present study's spiral fin-and-tube heat exchanger is a type of perforated serrated spiral finned tube heat exchanger. The dimensions of serrated spiral fin and tube heat exchanger shown in fig.1 and water flow arrangements shown in fig.2. The tests are performed under steady state conditions. And overall resistance can be obtained from the UA product of transfer units (ε-NTU). Yet total resistance is a sum of individual resistance as follows.

Table 1

Detailed geometric parameters of the test samples

Fin type	d_i (mm)	d_o (mm)	d_f (mm)	p_L (mm)	p_T (mm)	f_t (mm)	n_t (mm)	N_{row} (mm)	W_s (mm)	h_s (mm)	f_p (mm)
Serrated	8	9.52	30	40	38	0.5	7	2	1	3	2.5,3 and 3.5

Remarks: d_f = outside diameter of fin; d_i = Tube inside diameter; d_o = Tube outside diameter; p_L = longitudinal p_T = Transverse tube pitch;

f_t = Fin thickness; n_t = no of tubes in row; N_{row} = number of tube row; W_s = segment width; h_s = segment hight f_p = Fin pitch

Table 2

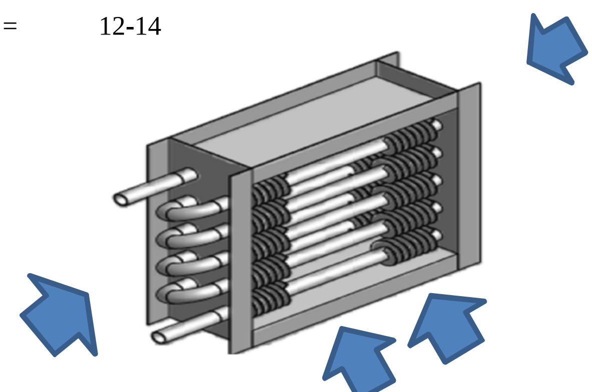
Experimental conditions

Inlet air dry bulb temperature, ºC = 31ºC

Inlet air frontal velocity, m/s = 2-4

Inlet water temperature, ºC = 46-50

Water flow rate, LPM = 12-14



Water outlet and Inlets

Air Flow Direction

Fig.1. Serrated spiral fin and tube heat exchanger

III. METHODOLOGY

The heat transfer and pressure drop characteristics are studied in the spiral fin heat exchangers, with and without hydrophilic coating. In the type of spiral fin heat exchanger having two different cases viz. without twist of fins, with twist of fins and without coating and with coating will be subjected to analysis. The analysis will performed in three different patterns.

Spiral fin heat exchanger without twist of fins.

Spiral fin heat exchanger without twist of fins and with hydrophilic coating.

Spiral fin heat exchanger with twist of fins and with hydrophilic coating.

A. Hydrophilic Coating

The serrated spiral fin and tube heat exchanger without hydrophilic coating in the previous experiments the condensate water may adhere as droplets on the fin surfaces without hydrophilic coating, and this phenomenon will cause bridging between the fins and increasing air pressure drop and the condensate water may corrode the aluminium fins and produce corrosion problems. So solving this problem the hydrophilic coating applied to the fin surface due to this increase the condensate water drainage and decrease pressure drop. The air side performance is different between spiral fin and tube with coating and without coating.

B. Serration Of Fins

The serrated spiral fin means the spiral fins are having cutting sections like teeth gear, the serrated spiral fin and having same performance when compared to full spiral fin.The seerated fin reduce material using and give good mechanical strength for the tube surfaces.

Fig 2 fabricated serrated fin with above dimensions.

C. Twisting Fins

The twisting of fins mean the fins tips are twisted for some angle for getting higher heat transfer,to find out performance of fin twisting the experimental analysis going to perform two different combinations with twisting and without twisting.

IV. NUMERICAL CALCULATION

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi k_t L} + \frac{1}{\mu_o h_o A_o} \quad (4.1)$$

The ε-NTU relations for multipass parallel cross-flow and multipass counter cross-flow configuration are available from (17-19),as shown in equations (2) and (3);

For multipass counter cross-flow with $N_{row} = 2$;

$$\varepsilon_c = 1 - \left(\frac{K}{2} + \left(1 - \frac{k}{2}\right) e^{2k/C_A}\right)^{-1}, K = 1 - e^{-NTU_A\left(C_A/2\right)} (4.2)$$

For multipass parallel cross-flow with $N_{row} = 2$;

$$\varepsilon_p = \left(1 - \frac{k}{2}\right)\left(1 - e^{-2k/C_A}\right), K = 1 - e^{-NTU_A\left(C_A/2\right)} (4.3)$$

Where C* = Cmin/Cmax is equal to Cc/Ch or Ch/Cc 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 parallel cross-flow and counter cross-flow. Hence it may be reasonable to use the average value of the relationships shown in Eq. (4) as follows:

$$\varepsilon_{pc} = \frac{\varepsilon_p + \varepsilon_c}{2} \quad \text{For } N_{row} = 2 \ (4.4)$$

The schematic diagram of circuitry arrangement (parallel, counter, cross) for Nrow = 2. Further details about the data reduction can be seen from Pongsoi et al. [11–16]. The efficiency of a radial fin with rectangular profile is based on the derivation of Gardner [20], i.e.,

$$\mu_f = \frac{2\Psi}{\phi(1+\Psi)} \frac{I_1(\phi R_O)K_1(\phi R_O) - I_1(\phi R_O)K_1(\phi R_O)}{I_O(\phi R_O)K_1(\phi R_0) + I_1(\phi R_O)K_O(\phi R_O)} \ (4.5)$$

Where

$$\phi = (r_o - r_i)^{3/2} \left(\frac{2h_o}{k_f A_p}\right)^{1/2} \quad (4.6)$$

Accordingly, the air-side heat transfer coefficient (ho) 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:

$$j = \frac{Nu}{Re_{do} pr^{1/3}} = \frac{}{\rho_a V_{max} c_p} (pr)^{2/3} \quad (4.7)$$

The frictional characteristics are termed with f-fanning friction factor, as depicted by Kays and London [21]:

$$f = \left(\frac{A_{min}}{A_o}\right)\left(\frac{\rho_m}{\rho_1}\right)\left(\frac{2\Delta p \rho_1}{G_c^2}\right) - (1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right) (4.8)$$

Where
A_{min} = minimum free flow area
A_o = Total heat transfer area
G_c = Maximum flux based on the free flow area
The heat transfer rate $q_v = m_f c_p (t_{out} - t_{in})/v$ (4.9)
The Power input per unit volume $e_v = m_f \Delta p / \rho_m \eta v$ (4.10)

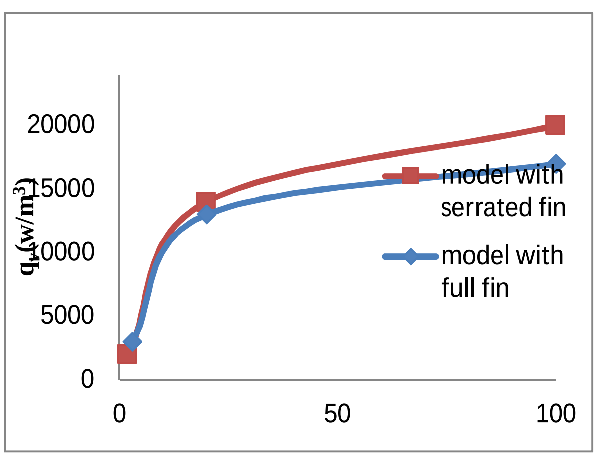
V. RESULTS AND DISCUSSION

The study focuses the effect of fin pitches (2, 2.5, and 3), serration of fins, twisting of fins on air side performance of serrated spiral fins with hydrophilic coating perforated on 6 tube pass compact fin and tube heat exchanger, this type of heat exchangers used in wide applications like automobile radiator ,economizers, refrigeration's.

A. *Effect Of Fin Serration*

The performance plot for the three configurations investigated is depicted in Fig.3 for Reynolds numbers 1000, 1200 and 1600, respectively. It is clear that for the same heat transfer area, the case with serrated fins delivers better performances than the case with full fins. The configuration with serrated fins shows an about 9% better heat transfer rate for the same power input than the configuration with full fins having the same heat transfer area. For the same fin height, even though the model with full fins has a higher area than the reference model with serrated fins, the performances of both configurations are close to each other. However, for a similar performance, the configuration with serrated fins has the advantage of savings in material of 12.3% compared to the case with full fin.

The tube surface at the base of conventional spiral fin is not covered by the fin, leading to tube corrosion and affect the heat transfer rate. So we need to find the alternate method, the serrated spiral fins have some distance from tube surface, this result no fouling in and increase heat transfer



$E_v(w/m^3)$

Fig.3.Performance plot of cases with and without fin serration

B. Effect Of Twisting Of Fin

The effect of fin twisting investigated for six different angles $\beta = 0°, 5°, 10°, 20°$.The twisting of fins on the tip of serrated fin increase the vortex generation due to this the natural convection heat transfer is increases. The twisting angle of fins 5º-10º gives better heat transfer performance. The twisting angles greater than 15˚result in a deterioration of the serrated finned tube performance. Fig.4. shows the performance of configurations with different angles.

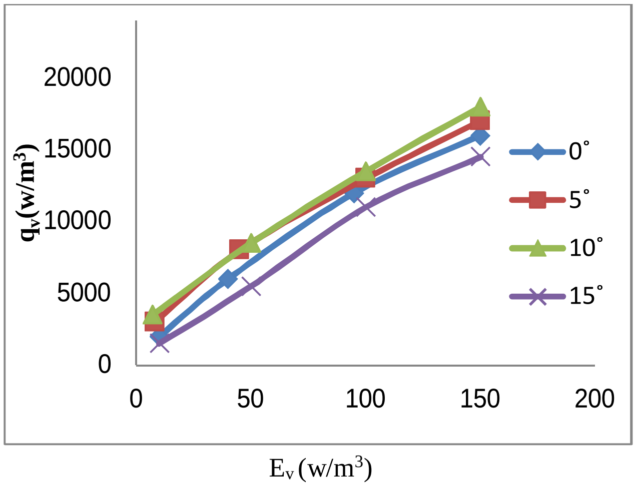


Fig.4. Performance of configurations different twist angles

C. Effect Of Fin Pitch

The experimental was carried out different fin pitches of 2, 2.5, 3, 3.5, from that fin pitches of 2.5,3 gives higher heat transfer rate and fin pitches affect heat transfer rate if fin pitch increases to increase pressure drop. The reduction of fin pitch increase heat transfer if fin pitch reduced more the is restricted to reduce the heat transfer rate. From the author Parinya Pongsoi, Somchai Wong wises gives the optimum fin pitches for better heat transfer rate (2.5, 3, and 3.5).

VI. CONCLUSIONS

The numerical investigation was carried out in this study show the advantage of perforated serrated fins in improving the performance of finned tubes or compact fin and tube heat exchanger. This is mainly due to the fact that the interruption of fins in these devices improves the re-build of the boundary layer close to heat transfer surfaces and increases the level of fluid mixing in the flow domain. The fin Pitches 2.5 to 3.5 gives better performance and also twisting of serrated fins between the angle 5º-10º gives more heat transfer rate

The results obtained show that for the same heat transfer area, the perforated serrated fin tubes have better performances than the full fins.The hydrophilic coating gives better performance in under wet conditions, there no corrosion on the finned tube when used for number of days in practical Application. It is useful for manufacturing industries and power plants to increase performance in future days.

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