

Air-Side Heat Transfer Coefficient of Thermosyphon Heat Pipe with Crimped Spiral Fins: A Case Study of Staggered Arrangement

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ABSTRACT

This research work studies the air-side heat transfer coefficient of thermosyphon heat pipe in case of staggered-arrangement tubes bank. Normally, the air-side heat transfer coefficient is the lowest and it controls the overall heat transfer of the system. To enhance the overall performance, extended surface is used at the air-side and the crimped spiral finned tube is normally selected in case of thermosyphon heat pipe. In this research work, the heat transfer coefficient of the thermosyphon heat pipe with crimped spiral fins is investigated. The parameters affecting the performance of heat exchanger such as tube diameter, fin spacing, fin height and tube spacing have been studied. Moreover the empirical correlation for evaluating the air-side heat transfer coefficient is also developed in this work.

Keywords: air-side heat transfer coefficient, thermosyphon heat pipe, crimped spiral fins, heat transfer model

1. INTRODUCTION

Many types of heat exchangers are used as industrial waste heat recovery units such as cross flow, rotary, run around coil, and especially thermosyphon heat exchanger which has high performance and low operating cost. The thermosyphon heat exchanger is used to recover heat from flue gas of the boiler or furnace and transfer this energy to increase the temperature of combustion air (air-preheater) or boiler feed water (economizer).

The thermosyphon heat exchanger composes of a set of thermosyphon heat pipes with in-line or staggered arrangement. In case of air-preheater, The effective method to improve the performance is to increase the air-side surface area with extended surface. The circular finned is normally designed for the thermosyphon and many correlations are developed for calculating the air-side heat transfer performance of this finned tube [1-3]. However, the process to construct the real circular finned tube is quite complicated with high cost. Thus the crimped spiral fins is used in

practice and Figure 1 shows the shape of the finned tube.

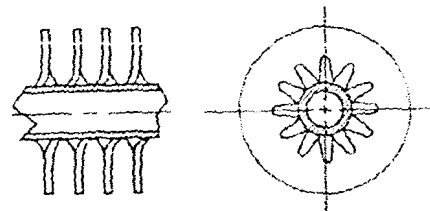


Figure 1. The shape of crimped spiral fins.

In Thailand, there are a few factories construct this kind of fin and no available heat transfer performance data. Therefore, in this study, the performance of the thermosyphon heat exchanger using crimped spiral fins is investigated and the correlation for evaluating the heat transfer performance of the finned tube is also formulated.

This work can be divided in to two parts. The first part is to test the performance of the crimped spiral fins and correlate the heat transfer data. The second part is to find out the performance of the thermosyphon heat

exchanger using this kind of finned tubes by using the data from Nuntaphan[4].

2. PERFORMANCE TEST OF CRIMPED SPIRAL FIN

2-1 Theory

This research work follows the ANSI/ARI standard [5] for testing the performance of the finned tubes. The air-side heat transfer coefficient of the crimped spiral fins can be evaluated by arranging it as a cross flow heat exchanger shown in Figure 2. Hot water is flowing inside the tube bank and transfers heat to the cross flow air stream. The heat transfer rate (Q) of heat exchanger can be calculated as

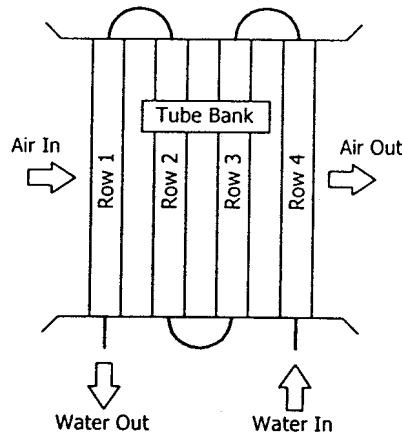


Figure 2. Cross flow heat exchanger.

$$Q = \dot{m}_a C_p (T_{ao} - T_{ai}) \quad (1)$$

$$Q = \dot{m}_w C_p (T_{wi} - T_{wo}) \quad (2)$$

where \dot{m}_a is the mass flow rate of air, \dot{m}_w is the mass flow rate of water, T_{ai} , T_{ao} are inlet and outlet temperatures of air stream, T_{wi} , T_{wo} are inlet and outlet temperature of water and C_p , C_p are specific heats of air and water respectively.

The performance indicator of heat exchanger is the effectiveness which is defined as

$$\varepsilon = \frac{Q}{Q_{\max}} \quad (3)$$

$$\varepsilon = \frac{Q}{(\dot{m}Cp)_{\min} \Delta T_{\max}} \quad (4)$$

where ε is the effectiveness. The relation between the effectiveness and the air side heat transfer coefficient in case of 4 tube rows is as follow

$$\varepsilon = \frac{1}{C^*} \left\{ 1 - e^{-4KC^*} \left[\frac{1 + C^*K^2(6 - 4K + K^2)}{+ 4(C^*)^2K^4(2 - K)} + \frac{8(C^*)^3K^6}{3} \right] \right\} \quad (5)$$

$$K = 1 - e^{-NTU/4} \quad (6)$$

$$NTU = \frac{UA}{(\dot{m}Cp)_{\min}} \quad (7)$$

$$C^* = \frac{(\dot{m}Cp)_{\min}}{(\dot{m}Cp)_{\max}} \quad (8)$$

$$\frac{1}{UA} = \frac{1}{\eta_o h_o A_o} + \frac{\ln(d_o / d_i)}{2\pi kL} + \frac{1}{h_i A_i} \quad (9)$$

where h is the heat transfer coefficient, A is area, d is tube diameter, L is tube length, k is thermal conductivity of tube material, η_o is surface efficiency of the airside and subscripts o, i are defined as the air side and the tube side respectively.

The tube side heat transfer coefficient can be calculated from Gnielinski correlation [6] as

$$h_i = \left(\frac{k}{d} \right)_i \frac{(Re_{Di} - 1000) Pr(f_i / 2)}{1 + 12.7 \sqrt{f_i / 2} (Pr^{2/3} - 1)} \quad (10)$$

$$f_i = [1.58 \ln(Re_{Di}) - 3.28]^2 \quad (11)$$

where Re_{Di} is the tube-side Reynolds number. The surface efficiency from equation (9) can be estimated from

$$\eta_o = 1 - \frac{A_f}{A_o} (1 - \eta) \quad (12)$$

$$A_o = A_f + A_b \quad (13)$$

where A_o is the total surface area of finned tube, A_f is surface area of fin, A_b is surface area of bare tube and η is fin efficiency which can be calculated from Schmidt approximation [7] as follows:

$$\eta = \frac{\tanh(mr\phi)}{mr\phi} \quad (14)$$

$$m = \sqrt{\frac{2h_o}{k_f f_t}} \quad (15)$$

$$\phi = \left(\frac{R_{eq}}{r} - 1 \right) \left[1 + 0.35 \ln \left(\frac{R_{eq}}{r} \right) \right] \quad (16)$$

$$\frac{R_{eq}}{r} = 1.27 \frac{X_M}{r} \left(\frac{X_L}{X_M} - 0.3 \right)^{1/2} \quad (17)$$

$$X_L = \frac{\sqrt{(S_t/2)^2 + S_l}}{2} \quad (18)$$

$$X_M = 0.5S_t \quad (19)$$

where k_f is thermal conductivity of fin material, S_t and S_l are transverse and longitudinal pitches of tube bank respectively and f_t is fin thickness.

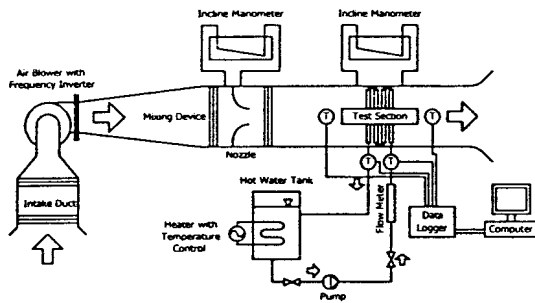


Figure 3. Schematic sketch of experimental setup.

2-2 The experimental setup

Figure 3 shows the experimental setup. The air stream at room temperature is fed through the tube bank while the hot water is flowing inside the tube. The water flow rate is kept constant at 8 l/min and the inlet temperature is at 65°C. The mass flow rate of air is varied in the range of 0.1-0.5 kg/s. By measuring the inlet and the outlet temperatures of the water and the air streams including the mass flow rate of each side, the air side heat transfer coefficient could be calculated.

Table 1 shows dimensions of the finned tubes and their arrangement. The effect on the heat transfer performance of the parameters such as tube diameter, fin height, fin spacing, fin thickness and tube arrangement are studied. Moreover the empirical model including these parameters is formulated for calculating the air side heat transfer coefficient

Table 1 Dimensions of various type of heat exchanger.

d_o	d_i	f_s	f_h	f_t	S_t	S_l	n_r	n_t
21.7	16.5	6.10	10.0	0.4	72.0	36.0	4	6
21.7	16.5	3.85	10.0	0.4	72.0	36.0	4	6
21.7	16.5	2.85	10.0	0.4	72.0	36.0	4	6
21.7	16.5	6.10	10.0	0.4	84.0	24.2	4	5
21.7	16.5	3.85	10.0	0.4	84.0	24.2	4	5
21.7	16.5	2.85	10.0	0.4	84.0	24.2	4	5
21.7	16.5	6.10	10.0	0.4	50.0	43.3	4	9
21.7	16.5	3.85	10.0	0.4	50.0	43.3	4	9
21.7	16.5	2.85	10.0	0.4	50.0	43.3	4	9
21.7	16.5	6.10	10.0	0.4	55.6	48.2	4	8
21.7	16.5	3.85	10.0	0.4	55.6	48.2	4	8
21.7	16.5	2.85	10.0	0.4	55.6	48.2	4	8
21.7	16.5	3.85	15.0	0.4	55.6	48.2	4	8
27.2	21.6	3.85	10.0	0.4	50.0	43.3	4	9

Note: All dimensions are in millimeter.

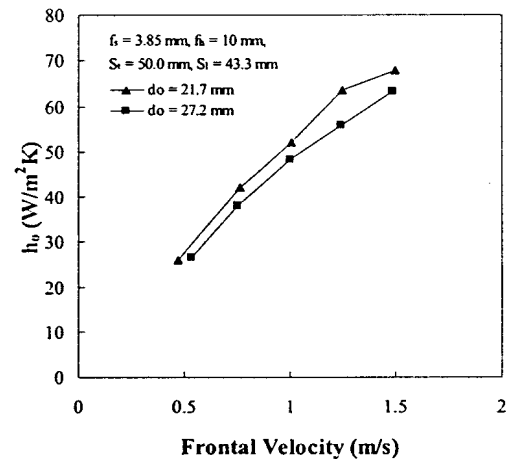


Figure 4. Effect of tube diameter on air-side heat transfer coefficient.

2-3 Results and Discussion

Figures 4-6 shows the relation of the air-side heat transfer coefficient and the mass flow rate of air stream. All of the mentioned Figures give the same undoubted results that higher the mass flow rate of air stream results in higher the heat transfer coefficient. From Figure 4, it is found that smaller diameter of tube gives higher heat transfer coefficient. This result comes from the re-circulation of air stream behind the tube that increases with the tube diameter. This affect pronounces the ineffective area of tube and it brings to get lower heat transfer coefficient. Wang et al. [8] also found this phenomena by using flow visualization.

Figure 5 shows the effect of fin spacing, and S_t on the airside heat transfer coefficients. The tube diameter is 21.7 mm and the fin height is 10 mm. For a smaller transverse tube pitch

($S_t = 50$ mm), one can see the effect of fin spacing on the heat transfer coefficients is negligible. This result is analogous to that of continuous fin geometry as reported by Rich [9] and Wang et al. [10]. However, for a larger transverse pitch of 84 mm, the heat transfer coefficients decrease with the decrease of fin spacing. This phenomenon may arise from the influence of airflow bypass effect. As is known, the corresponding pressure drop rises with the decrease fin spacing. As a consequence, although the airflow is directed by the tube row, the airflow is prone to flowing the portion where the flow resistance is smaller. For a very large of transverse tube pitch of 84 mm, part of the directed airflow just bypass the tube row and fin without effective contribution to the heat transfer, thereby causing a drop of heat transfer coefficients at smaller fin spacing. This flow bypass phenomenon becomes much where the transverse tube pitch is increased. Therefore, one can see no appreciable change of heat transfer coefficients for $S_t = 50$ mm.

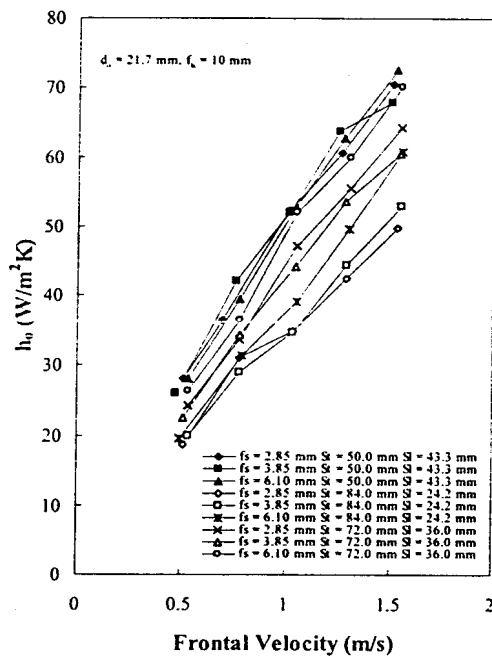


Figure 5. Effect of fin spacing and tube arrangement on the air-side heat transfer coefficient.

The effect of fin height on the airside performance is shown in Figure 6. It is found that the heat transfer coefficients of $f_h = 10$ mm are higher than that of $f_h = 15$ mm and the

explanation is as follow. In case of $f_h = 15$ mm, the airflow resistance around fin tube is larger than $f_h = 10$ mm. Therefore, part of the directed airflow just bypass the tube row without effective contribution to the heat transfer and lower heat transfer coefficient is obtained.

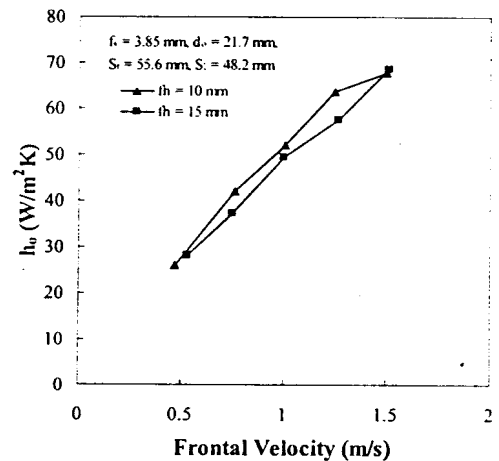


Figure 6. Effect of fin height on air-side heat transfer coefficient.

In this research work, the empirical model for predicting the heat transfer coefficient for various types of the finned tubes is also developed. The empirical model is as follow.

$$Nu = 0.0276 Re_D^{0.8834} \left(\frac{f_t}{f_s} \right)^{-0.1430} \left(\frac{S_t}{S_t} \right)^{1.1866} \left(\frac{S_t}{d_o} \right)^{0.7815} \left(\frac{d_c}{d_o} \right)^{-0.1250} \quad (20)$$

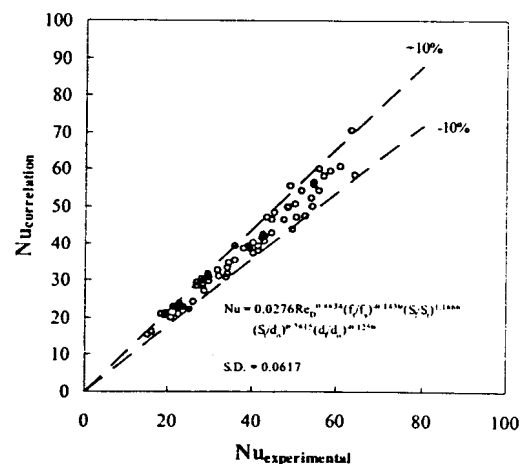


Figure 7. Comparison of Nu from the experimental data and the model.

Figure 7 shows the comparison of Nu from the experimental data and the model. It is found that the model can predict 98.6% of the experimental data with in $\pm 15\%$ error.

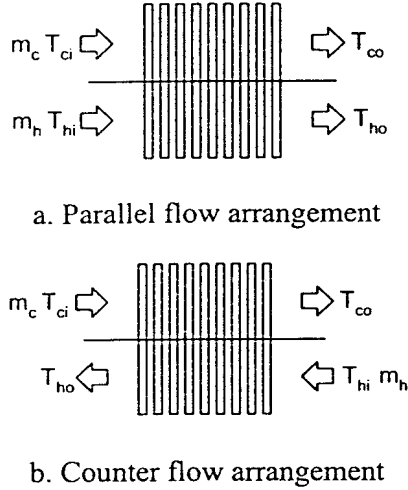


Figure 8. Thermosyphon heat exchanger. Water is working fluid inside.

3. PERFORMANCE OF THERMOSYPHON HEAT EXCHANGER USING CRIMPED SPIRAL FIN

In this part, the crimped spiral fin tube is used as in the air-to-air thermosyphon heat exchanger as shown in Figure 8. The heat transfer rate of the thermosyphon heat exchanger can be calculated as follows:

$$\begin{aligned}
 Q &= \dot{m}_h C_{p_h} (T_{hi} - T_{ho}) \\
 &= \dot{m}_c C_{p_c} (T_{co} - T_{ci}) \\
 &= (UA) \Delta T_{lmtcd}
 \end{aligned}
 \quad (21)$$

where UA is the product of overall heat transfer coefficient-area of the thermosyphon heat exchanger. In this research work, the simulation program for calculating the performance of thermosyphon heat exchanger developed by Nuntaphan [4] is selected for evaluating the performance. Figures 9-12 show the simulation results at various conditions.

From the Figures, it is found that the performance of the system depends on the mass flow rate of air and the inlet temperature of hot gas. Note that in this part the mass flow rate of the hot gas equals to the mass flow rate of the cold gas. Moreover, it is also found that the

counter flow gives higher performance than the parallel flow.

Figure 9 shows the effect of tube arrangement on the performance of thermosyphon heat exchanger. It is found that triangular arrangement gives the highest heat transfer rate because of its highest air-side heat transfer coefficient.

Figures 10-11 show the effect of fin spacing and fin height on the heat transfer rate. The outputs show the same results as those described in the previous section.

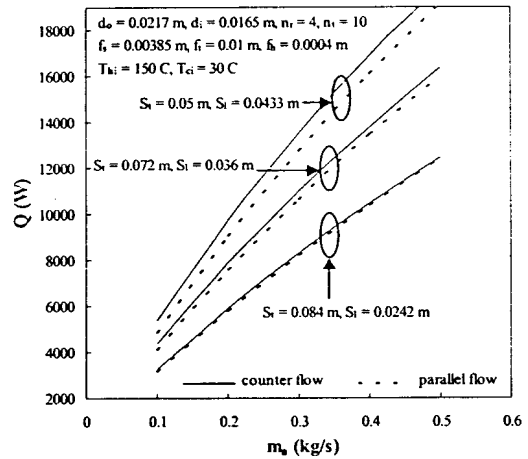


Figure 9. Effect of tube arrangement on the performance of thermosyphon heat exchanger.

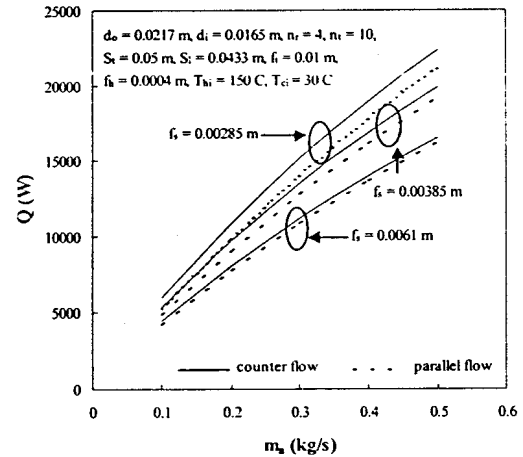


Figure 10. Effect of fin spacing on the performance of thermosyphon heat exchanger.

Figure 12 shows the effect of the inlet hot gas temperature on the heat rate. Higher the temperature difference between the heat source and heat sink results in higher heat transfer rate.

Anyway, the temperature must be controlled not to exceed the critical heat flux condition.

direction of air streams give high effect to the heat transfer rate of heat exchanger.

5. ACKNOWLEDGEMENT

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NOMENCLATURES

A	Area (m ²)
C _p	Specific heat (J/kgK)
d _f	Outside diameter of finned tube (mm)
d _i	Inside diameter of bare tube (mm)
d _o	Outside diameter of bare tube (mm)
f _h	Fin height (mm)
f _s	Fin Spacing (mm)
f _t	Fin thickness (mm)
h	Heat transfer coefficient (W/m ² K)
k	Thermal conductivity (W/mK)
L	Length (m)
\dot{m}	Mass flow rate (kg/s)
n _r	Number of tube rows
n _t	Number of tubes in row
NTU	Number of transfer unit
Nu	Nusselt Number
Pr	Prandtl number
Q	Heat transfer rate (W)
Re _D	Reynolds number
S _t	Transverse pitch (mm)
S _l	Longitudinal pitch (mm)
T	Temperature (°C)
U	Overall heat transfer coefficient (W/m ² K)

Greek symbols

ϵ	Effectiveness
η	Efficiency

Subscripts

a	Air
b	Bare tube
f	Fin
i	Inlet, tube side
o	Outlet, air side
w	Water

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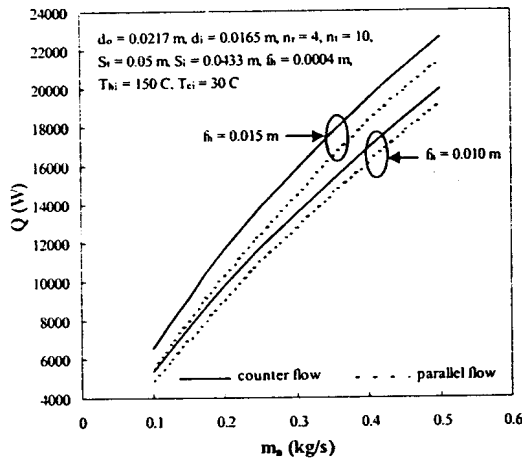


Figure 11. Effect of fin height on the performance of thermosyphon heat exchanger.

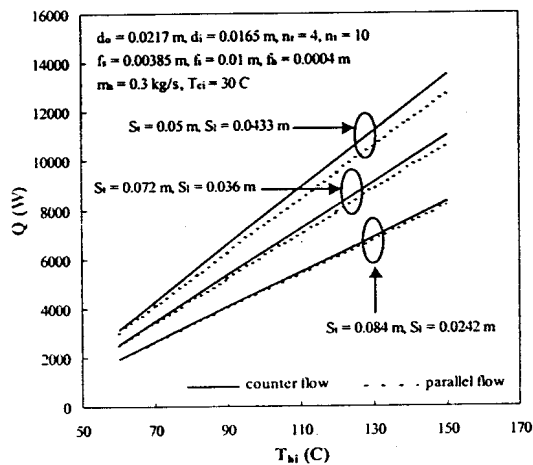


Figure 12. Effect of hot gas temperature on the performance of thermosyphon heat exchanger.

4. CONCLUSION

This work studies the performance of crimped spiral fin at various conditions such as fin spacing, fin height, tube arrangement, mass flow rate and temperature of air. The empirical model for predicting the air-side heat transfer coefficient is also developed and it can predict the results quite well. This research also studies the performance of thermosyphon heat exchanger using crimped spiral fin. It is found that fin spacing, fin height, tube arrangement, mass flow rate and temperature of air and the

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