



FINAL REPORT

Thermal Performance Testing of Finned Tube Producing in Thailand

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ABSTRACT

This research work studies the thermal performance of finned tube heat exchanger which is locally made. The crimped spiral finned tube is selected for this study. This work can be divided into two parts. The first part studies the performance of crimped spiral finned tube under the dry condition and the second part studies the performance under the dehumidifying condition.

A total of 23 cross flow heat exchangers having crimped spiral configurations is studied. The effect of tube diameter, fin spacing, fin height, transverse tube pitch, and tube arrangements are examined. From the experiment, it is found that these parameters play important roles not only on the heat transfer performance but also on the frictional characteristic. Moreover it is found that, in case of the dry surface heat exchanger, the thermal performance and the air stream pressure drop are slightly higher lower than the wet surface.

The empirical correlations for evaluating the heat transfer coefficient and pressure drop of both cases are also developed in this work. It is also found that the correlations can predict the experimental data quite well.

บทคัดย่อ

งานวิจัยนี้ศึกษาสมรรถนะเชิงความร้อนของท่อคัดกริบที่ผลิตในประเทศ โดยท่อกริบที่ศึกษาจะเป็นท่อกริบแบบเกลียวชนิดขอบหັก งานวิจัยนี้แบ่งออกเป็น 2 ส่วน โดยงานวิจัยส่วนแรกจะศึกษาสมรรถนะของกลุ่มท่อกริบในกรณีที่ไม่มีการควบแน่นของไอน้ำในอากาศบนพื้นผิวท่อ และงานวิจัยส่วนที่สองจะศึกษาในกรณีที่มีการควบแน่นของไอน้ำในบรรยากาศบนพื้นผิวของท่อกริบ

งานวิจัยนี้ได้ศึกษาเครื่องแลกเปลี่ยนความร้อนแบบไหลตามขวางที่ใช้ท่อกริบแบบเกลียวชนิดขอบหັกจำนวน 23 แบบ โดยได้ศึกษาผลของขนาดท่อ ระยะห่างระหว่างกริบ ความสูงของกริบ และการจัดเรียงท่อ ซึ่งจากการทดลองพบว่าพารามิเตอร์เหล่านี้ไม่เพียงแต่ส่งผลต่อสมรรถนะการถ่ายเทความร้อนเท่านั้น ยังส่งผลต่อความต้านทานการไหลของอากาศผ่านกลุ่มท่อด้วย และนอกจากนี้ยังพบว่าเครื่องแลกเปลี่ยนที่ไม่มีการควบแน่นของไอน้ำบนผิวท่อก็มีสมรรถนะเชิงความร้อนสูงกว่า และค่าความดันอากาศตกคร่อมกลุ่มท่อดำกว่าเครื่องแลกเปลี่ยนความร้อนที่มีการควบแน่นของไอน้ำบนพื้นผิวเล็กน้อย

งานวิจัยนี้ยังได้พัฒนาสมการสหสัมพันธ์เพื่อใช้คำนวณค่าสัมประสิทธิ์การถ่ายเทความร้อน และค่าความดันอากาศตกคร่อมกลุ่มท่อจากงานวิจัยทั้งสองส่วน ซึ่งสมการที่พัฒนาขึ้นสามารถใช้ทำนายผลการทดลองได้เป็นอย่างดี

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NOMENCLATURE

A	area (m^2)
$A_{\min}$	minimum free flow area (m^2)
A_o	total surface area (m^2)
$A_{p,i}$	inside surface area of tube (m^2)
$A_{p,m}$	mean surface area of tube (m^2)
$A_{p,o}$	outside surface area of tube (m^2)
h'_p	slope of straight line between the outside and inside tube wall temperature
h'_r	slope of the air saturation curved at the mean coolant temperature
$h'_{w,m}$	slope of the air saturation curved at the mean water film temperature of the external surface
$h'_{w,p}$	slope of the air saturation curve at the mean water film temperature of the primary surface
C_p	specific heat (J/kgK)
$C_{p,a}$	moist air specific heat at constant pressure (J/kgK)
$C_{p,w}$	water specific heat at coolant pressure (J/kgK)
d_f	outside diameter of finned tube (m)
d_i	tube inside diameter (m)
d_o	tube outside diameter (m)
f	friction factor
f_h	fin height (m)
f_i	in-tube friction factor of water
f_s	fin spacing (m)
f_t	fin thickness (m)
F	correction factor
G_c	mass flux of air base on minimum flow area (kg/sm^2)
$G_{\max}$	maximum mass velocity based on minimum flow area

h	heat transfer coefficient (W/m ² K)
$h_{c,w}$	sensible heat transfer coefficient for wet coil (W/m ² K)
h_i	inside heat transfer coefficient (W/m ² K)
$h_{o,w}$	total heat transfer coefficient for wet external fin (W/m ² K)
J_0	modified Bessel function solution of the first kind, order 0
J_1	modified Bessel function solution of the first kind, order 1
i	air enthalpy (J/kg)
$i_{a,in}$	inlet air enthalpy (J/kg)
$i_{a,out}$	outlet air enthalpy (J/kg)
$i_{i,m}$	saturated air enthalpy at the mean refrigerant temperature (J/kg)
$i_{r,in}$	saturated air enthalpy at the inlet of refrigerant temperature (J/kg)
$i_{r,out}$	saturated air enthalpy at the outlet of refrigerant temperature (J/kg)
$i_{s,p,i,m}$	saturated air enthalpy at the mean inside tube wall temperature (J/kg)
$i_{s,p,o,m}$	saturated air enthalpy at the mean outside tube wall temperature (J/kg)
$i_{s,w,m}$	saturated air enthalpy at the mean water film temperature of the external surface (J/kg)
Δi_m	mean enthalpy difference (J/kg)
j	the Colburn factor
K_0	modified Bessel function solution of the second kind, order 0
K_1	modified Bessel function solution of the second kind, order 1
k_f	thermal conductivity of fin (W/mK)
k_i	thermal conductivity of tube side fluid (W/mK)
k_p	thermal conductivity of tube (W/mK)
k_w	thermal conductivity of water (W/mK)
L	length (m)
m	parameter
$\dot{m}_a$	air mass flow rate (kg/s)
$\dot{m}_w$	water mass flow rate (kg/s)

n_r	number of tube row
n_i	number of tube in each row
NTU	number of transfer unit
Nu	Nusselt number
P	pressure (Pa)
ΔP	pressure drop (Pa)
Pr	Prandtl number
$\dot{Q}$	heat transfer rate (W)
$\dot{Q}_{avg}$	mathematical average heat transfer rate (W)
$\dot{Q}_a$	air-side heat transfer rate (W)
$\dot{Q}_w$	water side heat transfer rate (W)
r_i	distance from the center of the tube to the fin base (m)
r_o	distance from the center of the tube to the fin tip (m)
Re_{Di}	Reynolds number base on inside diameter of bare tube
Re_D	Reynolds number base on outside diameter of bare tube
S_l	longitudinal tube pitch (m)
S_t	transverse tube pitch (m)
S_{min}	minimum flow area (m ²)
T	temperature (°C)
$T_{w,m}$	mean temperature of water film (°C)
$T_{w,in}$	water temperature of at the tube inlet (°C)
$T_{w,out}$	water temperature of at the tube outlet (°C)
$T_{p,i,m}$	mean temperature of the inner tube wall (°C)
$T_{p,o,m}$	mean temperature of the outer tube wall (°C)
$T_{r,m}$	mean temperature of refrigerant coolant (°C)
U	overall heat transfer coefficient (W/m ² K)
$U_{o,w}$	overall heat transfer coefficient (W/m ² K)
V_{max}	maximum velocity (m/s)
x_p	thickness of tube wall (m)

δ_w thickness of condensate water film (m)

Greek symbols

ε effectiveness
 η efficiency
 $\eta_{f, wet}$ wet fin efficiency
 μ dynamic viscosity (Pas)
 ρ density (kg/m³)
 ρ_i density of inlet air (kg/m³)
 ρ_o density of outlet air (kg/m³)
 ρ_m mean density of air (kg/m³)
 σ contraction ratio of cross sectional area

Subscripts

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

EXECUTIVE SUMMARY

The thermal performance of locally manufactured finned tube is investigated in this research work. The crimped spiral finned tube, widely used in the industry, is selected in this study. Since there is a very few data about the performance of this finned tube, consequently, this work benefits to the heat exchanger designer.

The method for testing the performance of finned tube follows the ASHRAE and ARI standards. The finned tubes are arranged in a wind tunnel as a cross flow heat exchanger. The water circulates inside the tube while the air stream is flowing outside. By measuring the inlet and the outlet temperatures of the water and the air and also the mass flow rate of both fluids, the performance of the cross flow heat exchanger is evaluated. Note that the performance testing of the heat exchanger covers the heat transfer performance and the frictional characteristic.

This work can be divided into 2 parts. The first part is to study the performances of crimped spiral finned tube under the dry condition and the second part studies the performance under the dehumidifying condition.

For the first part, the approximately 65°C hot water is exchanging heat with the ambient air. While the second part, the cold water is exchanging heat with the approximately 65°C hot air. When the air exchanges heat with the water, the temperature of the air decreases. At the testing condition, the temperature decreasing of the air is designed to lower than the dew point of moisture. Therefore, there is a condensation of moisture from the air stream on the heat exchanger surface. The phenomenon of cooling is more complicated than that of the heating.

For heating and cooling, the heat exchangers have the same dimensions. In this work, the parameters affecting the performance of the heat exchanger are investigated such as tube diameter, fin height, fin spacing and tube arrangement. Note that, the mass flow rate of air is varied in the range of 0.1-0.5 kg/s while the volume flow rate of water is kept constant at 8 l/min. The empirical models for evaluating the heat transfer coefficient and the pressure drop of each part are also developed. The results of the performance testing are as follows:

The heating coil

1. For an inline arrangement, with the rise of tube diameter, the pressure drops increase but the associated heat transfer coefficients decrease with it. The increase of fin height

also gives rise to a considerable increase of the pressure drop but decreases the heat transfer coefficient.

2. For the inline arrangement, the effect of fin spacing on the airside performance varies with the transverse tube pitch. For a larger transverse tube pitch of 71.4 mm, there is an effect on heat transfer coefficient but no detectable influence of the fin spacing on frictional characteristics. It is likely that this phenomenon is related to the considerable airflow bypass between tube rows. On the contrary, at a smaller transverse tube pitch of 50 mm, one can see smaller fin spacing results in higher pressure drops and lower heat transfer coefficients.
3. For the staggered arrangement, the effect of tube diameter on the airside performance is analogous to that of the inline arrangement but to a comparatively small extent. This is because the recirculation zone behind the tube row is much smaller in a staggered arrangement. The effect of the fin height on the pressure drops much smaller than that of inline arrangement due to the major contribution to the pressure drops is from the blockage of the subsequent tube row in a staggered arrangement.
4. The effect of fin spacing on the airside performance for staggered arrangement also varies with the transverse tube pitch. For a smaller transverse tube pitch of 50 mm, there is no appreciable influence of the fin spacing on heat transfer. On the contrary, at a larger transverse tube pitch of 84 mm, one can see smaller fin spacing leads to lower heat transfer coefficients. This is also attributed to the presence of airflow bypass effect.
5. Correlations of the present crimped spiral fins in both staggered and inline arrangement are developed. The proposed correlations give quite good predictive ability against the present test data.

The cooling coil

1. The pressure drop of wet surface heat exchanger increases with the mass flow rate of air and the result is slightly higher or close to that of dry surface because only water condensate can be easily drained in the large fin spacing and individual finned configuration.
2. The heat transfer coefficient of the wet surface is slightly lower than that of the dry surface.
3. The effect of tube diameter on the airside performance is significant. Larger tube diameter not only gives lower heat transfer coefficient but also contributes

significantly to the increase of pressure drops. This phenomenon is applicable in both dry and wet condition.

4. For wet surface, the influence of fin height is negligible whereas there is a small effect in dry surface.
5. The effect of fin spacing on the heat transfer performance is rather small. However, the increasing of fin spacing tends to have a lower heat transfer coefficient.
6. The tube arrangement plays an important role on the heat transfer coefficient, the lower transverse pitch gives the higher heat transfer coefficient.
7. Airside performance in the present study is presented in terms of f and the j factor. The proposed correlations in case of staggered arrangement can predict 75% and 95% of experimental data within $\pm 15\%$. In case of inline arrangement, the proposed correlations can predict the j and the f factors can estimated about 85.7% and 82.3% of experimental data within $\pm 15\%$ accuracy.

CHAPTER 1

INTRODUCTION

1.1 Statement of Problems and Background

There are many types of heat exchanger applicable to the recovery of waste heat such as shell-and-tube, plate, and cross-flow heat exchangers. Among them, the cross flow type is especially suitable for recovering heat from high temperature air. Normally, the liquid always flows inside the tube whereas the gas flows across the tube bank. Because the dominant resistance occurs on the airside, it is a common practice to employ fins (such as circular fins) on the tube bundle to increase the gas side performance. For circular fin geometries, there are many empirical correlations available in the literature [1-5]. However, circular fin implemented in typically industrial application is usually in the form of crimped spiral fin as shown in Figure 1.1. Unfortunately, there is no airside data reported in the literature especially the finned tube manufactured in Thailand. In this regard, it is the objective of this study to present relevant airside data of the crimped spiral fins produced from Thai industry. Furthermore, relevant important geometric parameters such as tube diameter, fin spacing, fin height, and tube spacing influencing the airside performance are also investigated. Moreover the empirical correlations for evaluating the airside performance are also developed.

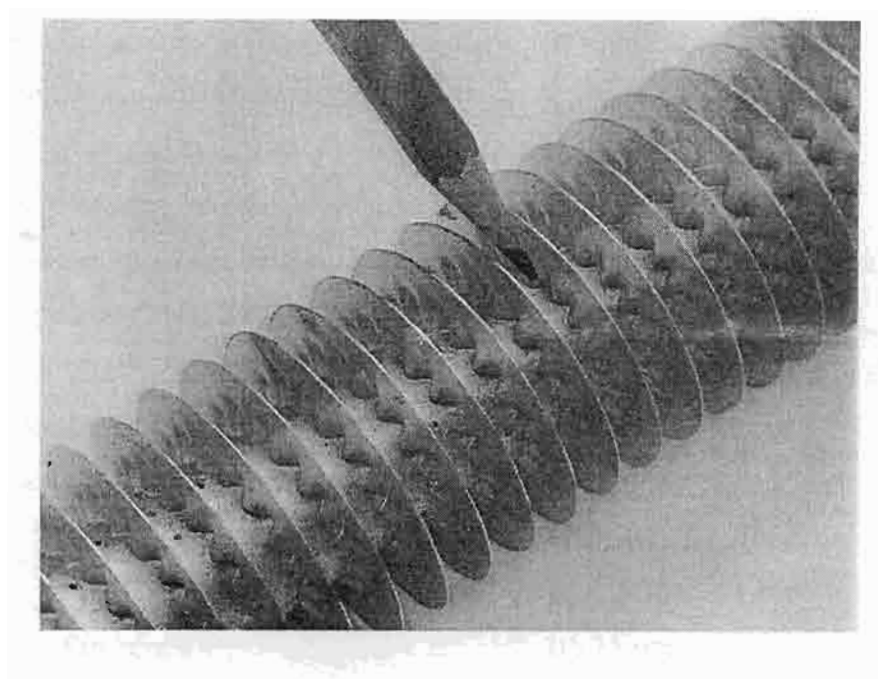


Figure 1.1 Schematic of the crimped spiral fin geometry.

The first part of this work is dealing with the performance testing of the heat exchanger operating under no condensation of moisture from the air stream on the heat exchanger surface. Actually, for practical waste heat recovery system, the heat exchanger may accompany with condensation of moisture air on the heat exchanger surface. Although the designers try to avoid this situation due to considerably corrosive problem associated with it, condensation may still takes place from time to time. This is commonly encountered if the load is not constant such as small boilers where the steam consumption varies with time and the the flue gas temperature fluctuates in a wide range. In that regard, the airside performance in the presence of dehumidification is rather important. Unfortunately, there are simply no data reported for the crimped spiral finned heat exchangers. Hence, it is the objective of the second part of this work. The heat transfer and friction characteristic of cross flow heat exchanger using crimped spiral fin in the presence of dehumidification are reported. Moreover, the heat transfer and friction correlations are also developed in this part.

1.2 Research Objectives

1. Develop the correlation for predicting the heat transfer coefficient of cross flow heat exchanger using crimped spiral fin under no condensation of moisture from the air stream and dehumidifying conditions.
2. Develop the correlation for predicting the air stream pressure drop of cross flow heat exchanger using crimped spiral fin under no condensation of moisture from the air stream and dehumidifying conditions.

1.3 Scope of Work

The aim of this work is to correlate the heat transfer and frictional characteristic of the cross flow heat exchanger using crimped spiral finned tube which are locally manufactured. The dimensions of finned tube are the same size as using in the industry. The dimensions and the testing conditions of this research are as follow:

1. Dimensions of finned tube

Outside diameter of tube	17.3, 21.7, 27.2 mm.
Fin spacing	2.85, 3.85, 6.10 mm.
Fin height	10, 15 mm.
Fin thickness	0.4 mm.
Tube arrangement	Inline, Staggered

2. Testing conditions

Part 1. No condensation of moisture from the air stream.

Inlet temperature of air	25 °C (approx.)
Inlet temperature of hot water	65 °C
Frontal velocity of air	0.5-2 m/s
Mass flow rate of water	0.12 kg/s

Part 2. Dehumidifying condition.

Inlet temperature of hot air	65 °C
Inlet temperature of water	25 °C
Frontal velocity of air	0.5-2 m/s
Mass flow rate of water	0.12 kg/s
Relative humidity of air	50% RH (approx.)

1.4 Methodology

1. Design the test rig for testing the performance of a cross flow heat exchanger using crimped spiral fin.
2. Calibrate all of the instrument by follow the standard method [6-9]
3. Performance testing of a cross flow heat exchanger under no condensation of moisture of air stream on heat exchanger surface and construct the empirical model for evaluating the heat transfer coefficient and the air stream pressure drop.
4. Performance testing of a cross flow heat exchanger under dehumidifying condition and construct the empirical model for evaluating the heat transfer coefficient and the air stream pressure drop.

1.5 Expected Benefits

The results from this research can help the heat exchanger designer select the suitable size of the heat exchanger using the crimped spiral finned tubes.

CHAPTER 2

EXPERIMENTAL SET-UP

2.1 Overview of the Experimental Apparatus

As mention in the previous chapter, this research work divides into two parts. The first part is to study the performance of the cross flow heat exchanger using crimped spiral finned tube operating under the condition having no condensation of moisture from the air stream on heat exchanger surface. Normally the heat exchanger operating under this condition is called heating coil. The second part investigates the performance in case of dehumidifying condition that having condensation of moisture on the heat exchanger surface. This heat exchanger is called cooling coil.

Therefor the experimental set-up of this work can be divided into two parts follow the testing conditions. The experimental apparatus is designed to support the testing conditions and the schematic sketch of the apparatus is shown in Figure 2.1. Note that this apparatus is designed and calibrated by following the ASHRAE and ARI standards [6-9]. The apparatus shown in Figure 2.1 is the wind tunnel. The cross flow heat exchanger is mount in this tunnel. This heat exchanger exchanges heat between the air stream and the water. The next sections will describe the operating methods and the testing conditions of this test rig. Figures 2.2-2.10 show the images of the experimental apparatus.

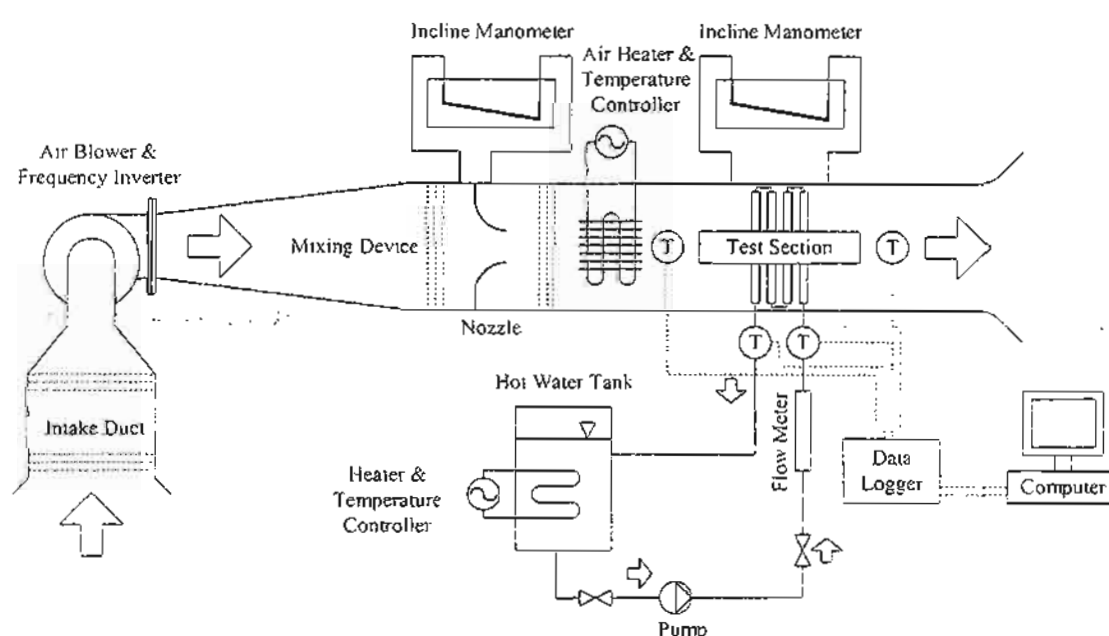


Figure 2.1 Schematic sketch of the experimental apparatus.



Figure 2.2 The front view of the apparatus.

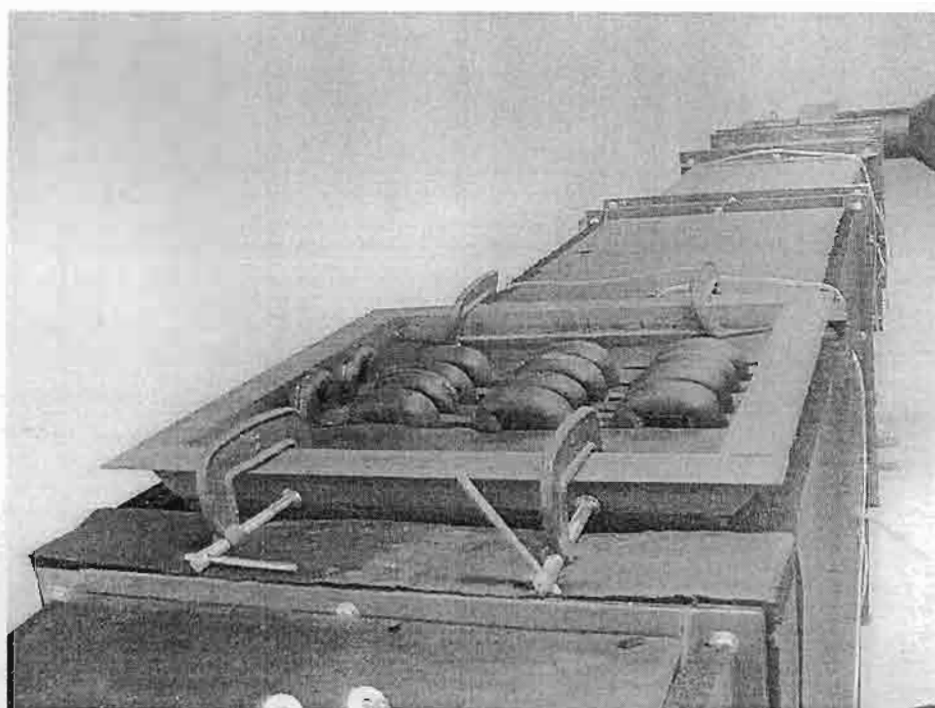


Figure 2.3 The top view of the apparatus at testing section.

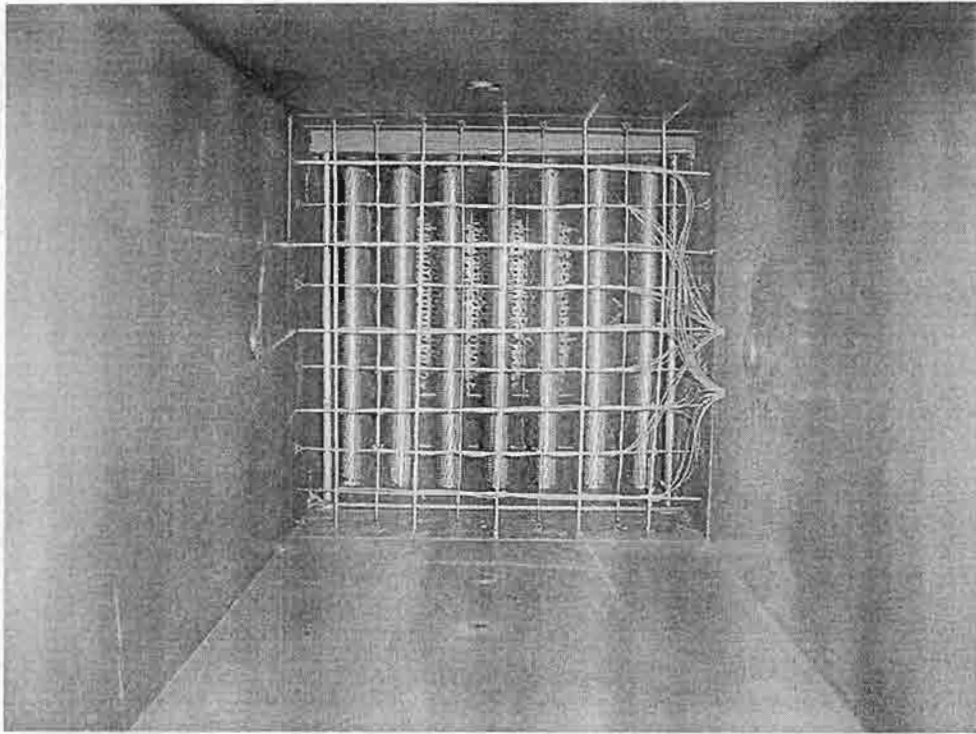


Figure 2.4 The cross flow heat exchanger setting in the wind tunnel.

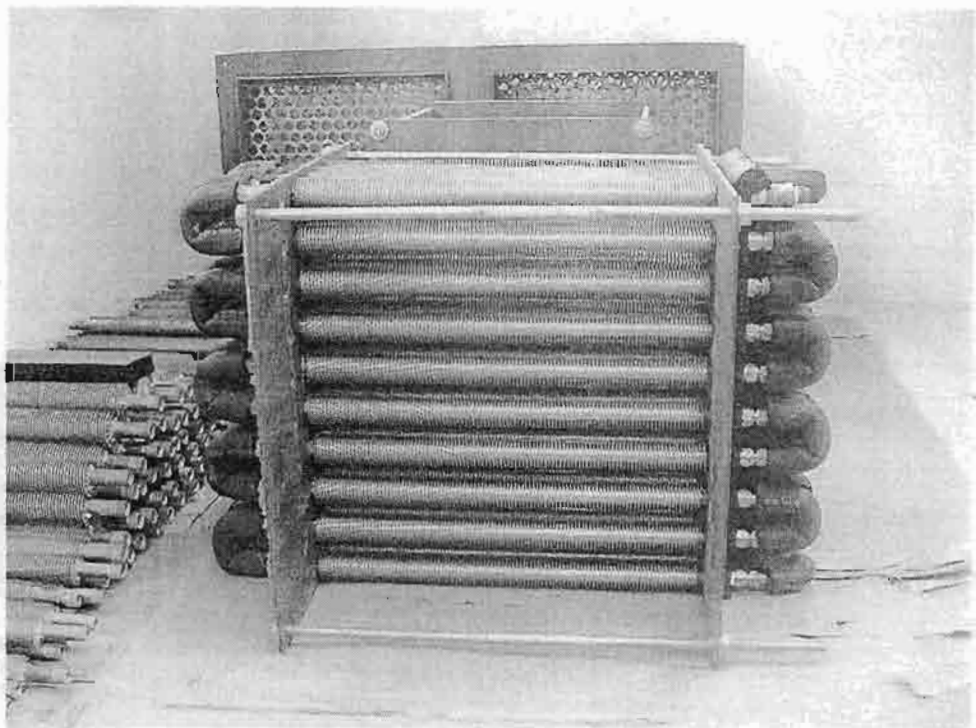


Figure 2.5 The testing specimen before mount in the wind tunnel.

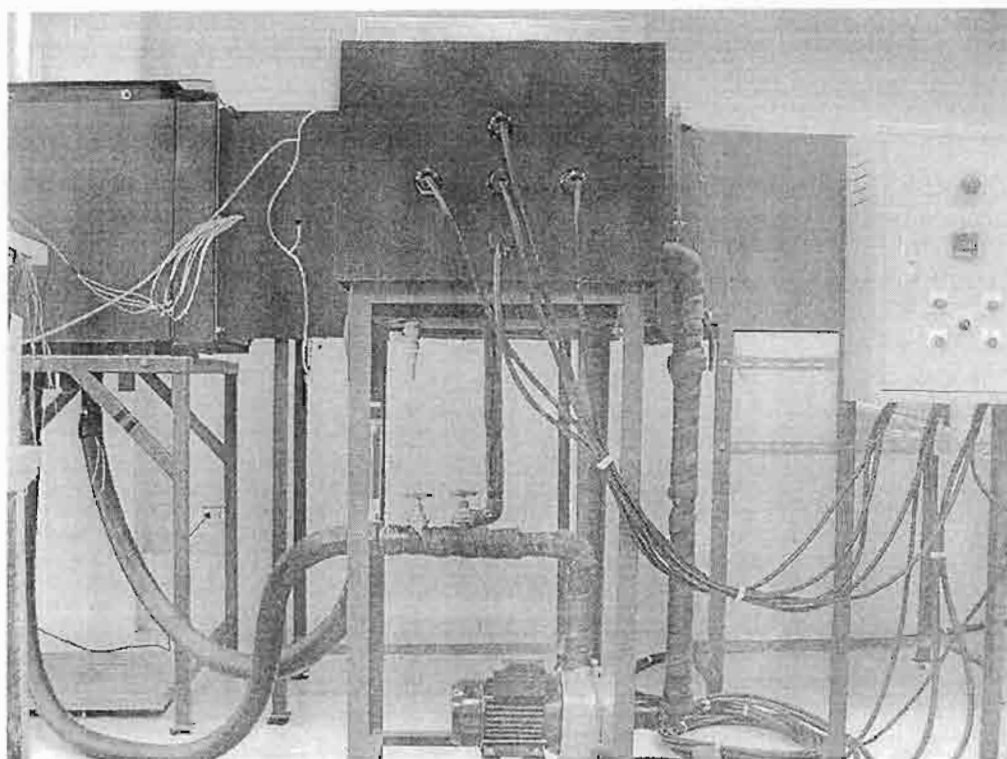


Figure 2.6 Hot water tank.

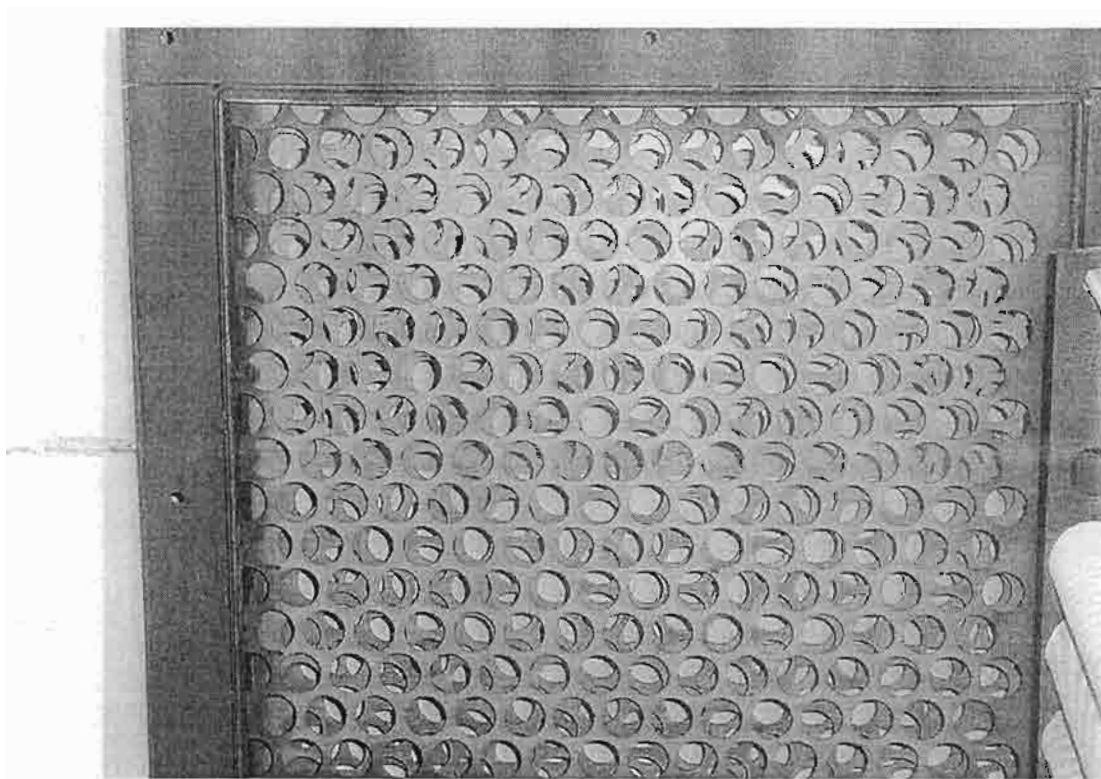


Figure 2.7 The mixing device.

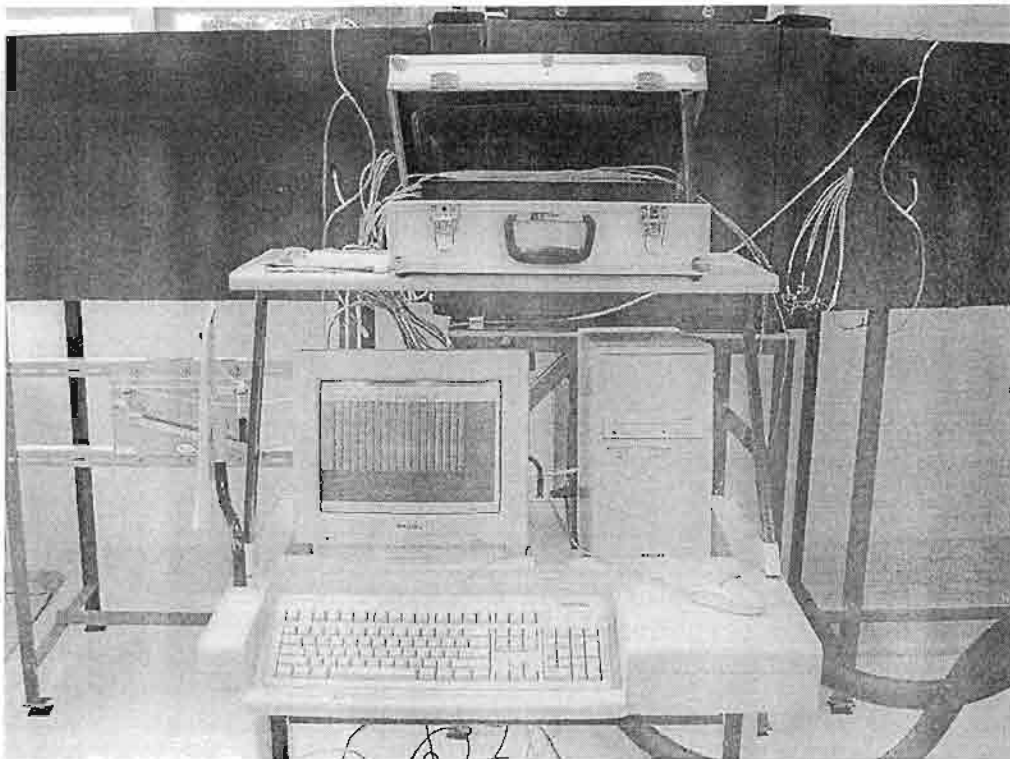


Figure 2.8 Temperature data logger and computer controller.

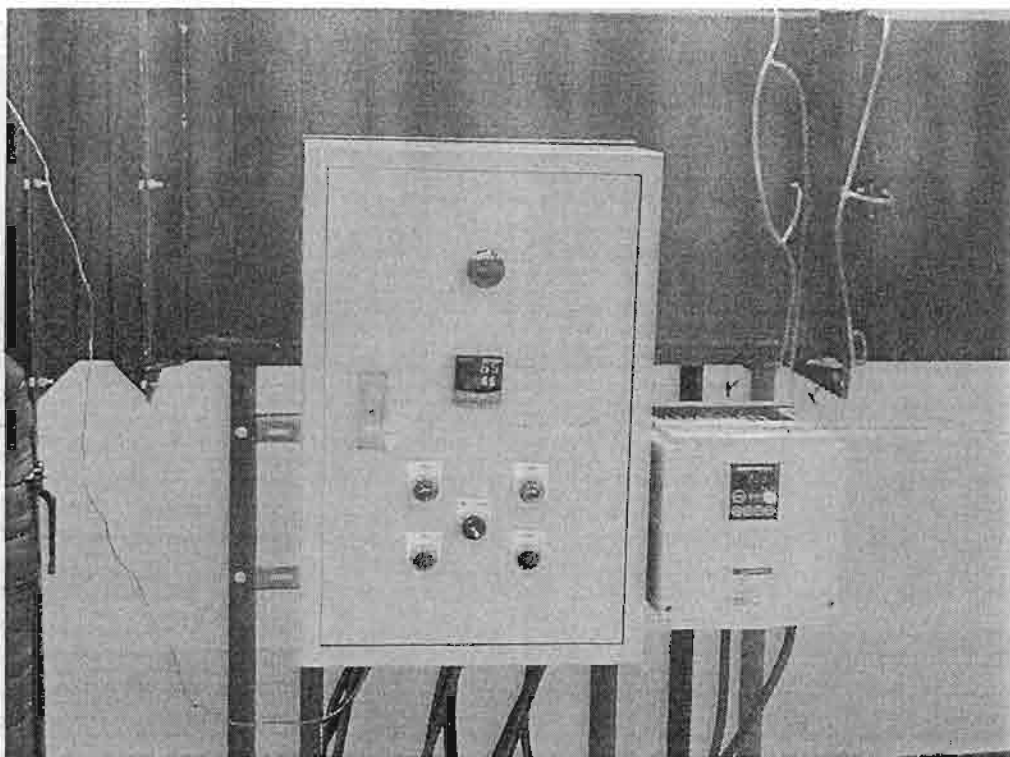


Figure 2.9 Control panel of the wind tunnel.

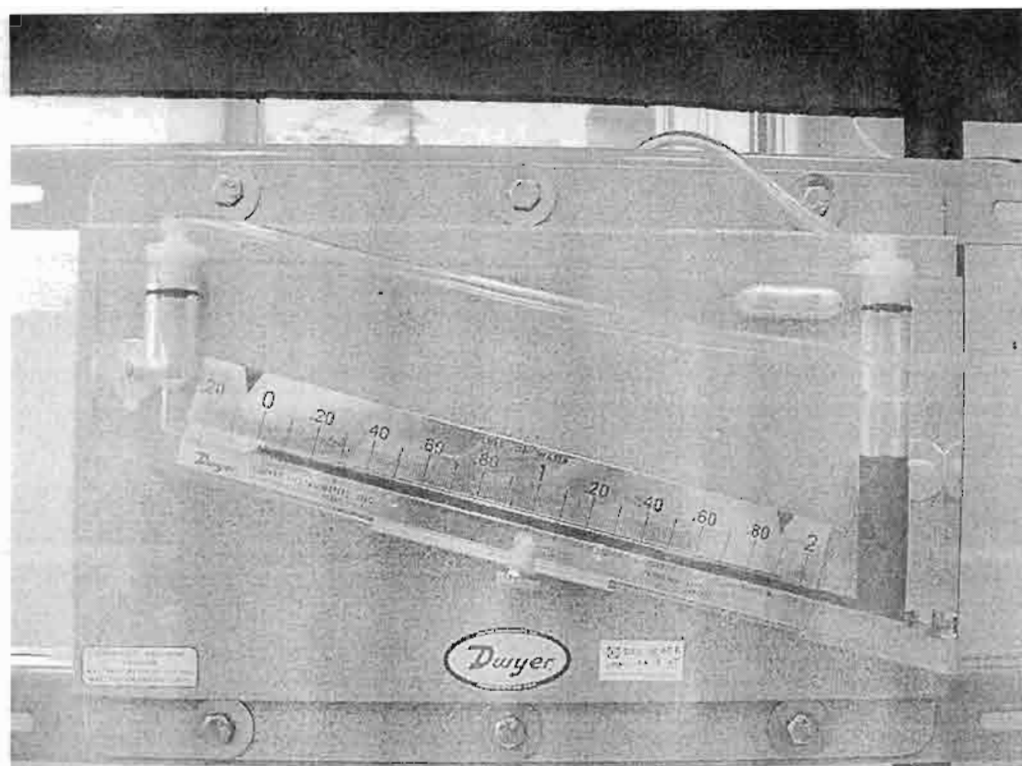


Figure 2.10 Inclined manometer for measuring the pressure drop.

2.2 The Experimental Set-up of the First Part

As mention in the previous section, the first part is to study the performance of the cross flow heat exchanger using crimped spiral finned tube operating under the condition having no condensation of moisture from the air stream on heat exchanger surface.

From Figure 2.1, the air stream at room temperature flows through the tube bank with hot water circulating inside the tubes. In this experiment, the water flow rate is kept at a constant flowrate of 8 L/min. An accurate flowmeter is used for the measurement with a precision of ± 0.1 L/min. The inlet temperature of water is maintained at 65°C . Both of the inlet and outlet temperatures of water are measured by a set of calibrated K-type thermocouples and the signals are recorded by a temperature data logger.

The airflow across the heat exchanger is generated by a 1.5 kW centrifugal air blower with the controllable range of 0.1-0.5 kg/s by using a frequency inverter. The mass flow rate of air stream is measured by a standard nozzle and an inclined manometer with ± 0.5 Pa accuracy. The inlet and the outlet temperatures of air stream are also measured by another set of K-type thermocouple mesh. The inlet and outlet temperature

measuring meshes consist of 16 and 41 thermocouples, respectively. Note that all of thermocouples have been calibrated to ± 0.1 °C accuracy. The pressure drop across the heat exchanger is also measured by the inclined manometer with ± 0.5 Pa accuracy.

A total of 23 crimped spiral fin heat exchangers having various geometric parameters are tested in this study. Table 2.1 lists the details of the tested samples. Relevant definitions of the geometrical parameters can be also shown in Figure 2.11. Notice that both inline and staggered arrangements are tested in this study. The effects of tube diameter, fin height, fin spacing, fin thickness, and tube arrangement on the airside performance are examined accordingly.

Table 2.1 Geometric dimensions of cross flow heat exchanger.

No	d_o (mm)	d_i (mm)	f_s (mm)	f_h (mm)	f_t (mm)	S_t (mm)	S_l (mm)	n_r	n_l	arrangement
1	17.3	13.3	6.10	10.0	0.4	50.0	50.0	4	10	inline
2	21.7	16.5	6.10	10.0	0.4	71.4	50.0	4	7	inline
3	21.7	16.5	6.10	10.0	0.4	50.0	50.0	4	10	inline
4	21.7	16.5	3.85	10.0	0.4	71.4	50.0	4	7	inline
5	21.7	16.5	3.85	10.0	0.4	50.0	50.0	4	10	inline
6	21.7	16.5	2.85	10.0	0.4	71.4	50.0	4	7	inline
7	21.7	16.5	2.85	10.0	0.4	50.0	50.0	4	10	inline
8	21.7	16.5	3.85	15.0	0.4	71.4	50.0	4	7	inline
9	27.2	21.6	3.85	10.0	0.4	50.0	50.0	4	10	inline
10	21.7	16.5	6.10	10.0	0.4	72.0	36.0	4	6	staggered
11	21.7	16.5	3.85	10.0	0.4	72.0	36.0	4	6	staggered
12	21.7	16.5	2.85	10.0	0.4	72.0	36.0	4	6	staggered
13	21.7	16.5	6.10	10.0	0.4	84.0	24.2	4	5	staggered
14	21.7	16.5	3.85	10.0	0.4	84.0	24.2	4	5	staggered
15	21.7	16.5	2.85	10.0	0.4	84.0	24.2	4	5	staggered
16	21.7	16.5	6.10	10.0	0.4	50.0	43.3	4	9	staggered
17	21.7	16.5	3.85	10.0	0.4	50.0	43.3	4	9	staggered
18	21.7	16.5	2.85	10.0	0.4	50.0	43.3	4	9	staggered
19	21.7	16.5	6.10	10.0	0.4	55.6	48.2	4	8	staggered
20	21.7	16.5	3.85	10.0	0.4	55.6	48.2	4	8	staggered
21	21.7	16.5	2.85	10.0	0.4	55.6	48.2	4	8	staggered
22	21.7	16.5	3.85	15.0	0.4	55.6	48.2	4	8	staggered
23	27.2	21.6	3.85	10.0	0.4	50.0	43.3	4	9	staggered

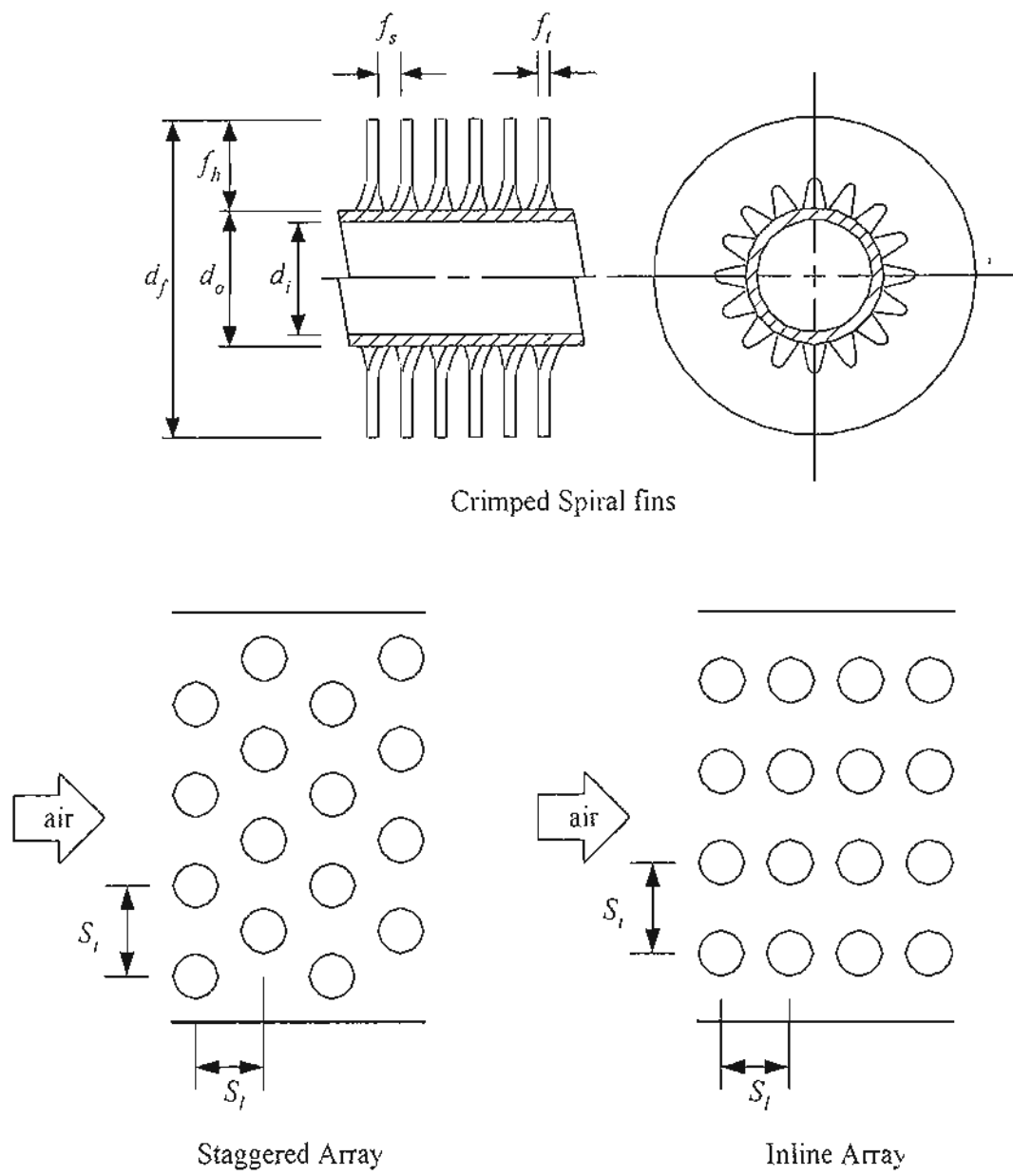


Figure 2.11 Details of crimped spiral fins geometry and tube arrangements.

2.3 The Experimental Set-up of the Second Part

The second part is to study the performance of the cross flow heat exchanger using crimped spiral finned tube operating under dehumidifying condition. In this part, the hot air stream flows through the tube bank and the water at room temperature circulates inside the tubes. In this experiment, the water flow rate is kept constant at 8 L/min. An accurate water flow meter is used for the measurement with a precision of ± 0.1 L/min. The inlet temperature of water is approximately 30°C. Both the inlet and outlet temperatures of water are measured by a set of calibrated K-type thermocouples and a temperature data logger records these signals.

A 1.5 kW centrifugal air blower accompanied with a frequency inverter having a controllable range of 0.1-0.5 kg/s air is used to conduct flowing air across the heat exchanger. A standard nozzle and an inclined manometer are adopted to measure the mass flow rate of air stream. The uncertainty of the inclined manometer is ± 0.5 Pa accuracy. The inlet temperature of air stream is kept constant at 65 °C by the set of heaters and the temperature controller. The inlet and the outlet dry bulb temperatures of air stream are also measured by another set of two meshes of K-type thermocouples. The inlet and the outlet wet bulb temperatures of air stream are also measured. Note that all of thermocouples have been calibrated to ± 0.1 °C accuracy. The pressure drop across the heat exchanger is measured by another set of inclined manometer with calibrated uncertainty of ± 0.5 Pa accuracy.

A total of 10 crimped spiral fin heat exchangers having various geometric parameters are tested in this study. The details of the tested samples is the same as shown in Table 2.1. Relevant definitions of the geometrical parameters can be also shown in Figure 2.11. The effects of tube diameter, fin height, fin spacing, fin thickness, and tube arrangements on the airside performance are examined accordingly.

CHAPTER 3

AIR-SIDE PERFORMANCE OF HEAT EXCHANGER USING CRIMPED SPIRAL FINNS, A CASE STUDY OF HEATING COIL

3.1 Introduction

This chapter describes the performance testing of cross flow heat exchanger using crimped spiral fins producing from Thai industry. Actually, there are many reports about the performance of this kind of heat exchanger, especially in case of circular fin. Briggs and Young [1] developed the air side heat transfer coefficient of the staggered tube bank in term of Colburn j factor as:

$$j = 0.134 \text{Re}_D^{-0.319} \left(\frac{f_s}{f_h} \right)^{0.2} \left(\frac{f_s}{f_l} \right)^{0.11} \quad (3.1)$$

Note that the above correlation is valid for four or more tube rows and applicable to the following ranges: $1100 \leq \text{Re}_D \leq 18000$, $1.0 \leq f_s/f_l \leq 6.6$, $0.09 \leq f_h/d_o \leq 0.69$, $0.01 \leq f_l/d_o \leq 0.15$, and $15 \leq S_l/d_o \leq 8.2$. The standard deviation of the Briggs and Young correlation is 5.1%.

Robinson and Briggs [2] provided the frictional performance termed as the Fanning friction factor for the circular fin tube bank having staggered arrangement:

$$f = 9.47 \text{Re}_D^{-0.316} \left(\frac{S_l}{d_o} \right)^{-0.927} \left(\frac{S_l}{S_d} \right)^{0.515} \quad (3.2)$$

Equation (2) is based on isothermal air flow with data from 17 staggered tube banks. Its applicable range is $2,000 \leq \text{Re}_D \leq 50,000$, $0.15 \leq f_s/f_h \leq 0.19$, $3.8 \leq f_s/f_l \leq 6.0$, $0.35 \leq f_h/d_o \leq 0.56$, $0.01 \leq f_l/d_o \leq 0.03$ and $1.9 \leq S_l/d_o \leq 4.6$. The standard deviation of Equation (2) is 7.8%.

Rabas et al. [3] developed the Colburn j factor and friction factor for circular finned tube bank having low fin height and small fin spacing:

$$j = 0.292 \text{Re}_D^n \left(\frac{f_s}{d_o} \right)^{1.12} \left(\frac{f_s}{f_h} \right)^{0.26} \left(\frac{f_l}{f_s} \right)^{0.67} \left(\frac{d_f}{d_o} \right)^{0.47} \left(\frac{d_f}{f_l} \right)^{0.77}, \quad (3.3)$$

$$n = -0.415 + 0.0346 \left(\frac{d_f}{f_s} \right), \quad (3.4)$$

$$f = 3.805 \text{Re}_D^{-0.234} \left(\frac{f_s}{d_o} \right)^{0.25} \left(\frac{f_s}{f_h} \right)^{0.76} \left(\frac{d_o}{d_f} \right)^{0.73} \left(\frac{d_o}{S_l} \right)^{0.71} \left(\frac{S_l}{S_t} \right)^{0.38} \quad (3.5)$$

Equations (3.3)-(3.5) are also valid with 4 or more rows of tube bank. The applicable range are $5,000 \leq Re_D \leq 25,000$, $1.3 \leq f_s / f_h \leq 1.5$, $0.01 \leq f_s / f_l \leq 0.06$, $f_h / d_o \leq 0.1$, $0.01 \leq f_l / d_o \leq 0.02$ and $1.3 \leq S_l / d_o \leq 1.5$. These equations could predict 94% and 90% of the experimental j and f data within $\pm 15\%$.

ESDU [4] recommended a correlation for heat transfer performance of high fin geometry:

$$Nu = 0.242 Re_D^{0.658} \left(\frac{f_s}{f_h} \right)^{0.297} \left(\frac{S_l}{S_l} \right)^{-0.091} Pr^{0.333} \quad (3.6)$$

This correlation is also valid for air-cooled heat exchanger with four or more tube rows and the suitable ranges for application are $2,000 \leq Re_D \leq 40,000$, $0.13 \leq f_s / f_h \leq 0.57$, $1.15 \leq S_l / S_l \leq 1.72$.

The preceding correlations were developed with the cross flow heat exchanger having staggered arrangement. In case of inline arrangement, despite of its comparatively low heat transfer performance, its lower pressure drop and high reliability (easy to maintain and clean) are very attractive in very severe environment. For the heat transfer performance, Schmidt [5] recommended a correlation for inline arrangement with high fin,

$$Nu = 0.30 Re_D^{0.625} \left(\frac{A}{A_l} \right)^{-0.375} Pr^{0.333} \quad (3.7)$$

Where A and A_l are the total surface area and that of bare tube. Range of applicability for Equation (3.7) is $5,000 \leq Re_D \leq 100,000$ and $5 \leq A / A_l \leq 12$.

However, circular fin implemented in typically industrial application is usually in the form of crimped spiral fin. Unfortunately, there is no airside data reported in the literature especially the crimped spiral fin producing from Thai industry. In this regard, it is the objective of this study to present relevant airside data of the crimped spiral fins. Furthermore, relevant important geometric parameters such as tube diameter, fin spacing, fin height, and tube spacing influencing the airside performance are also investigated. Moreover the empirical correlations capable of evaluating the airside performance are also developed.

3.2 Data Reduction

During the experiment, hot water flowing inside the tube bank transfers heat to the outside air and the the heat transfer rate (Q) can be calculated as

$$Q = \dot{m}_a C_{p_a} (T_{ao} - T_{ai}), \quad (3.8)$$

$$Q = \dot{m}_w C_{p_w} (T_{wi} - T_{wo}), \quad (3.9)$$

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 the inlet and outlet temperatures of air stream, T_{wi}, T_{wo} are the inlet and outlet temperature of water, and C_{p_a}, C_{p_w} are the specific heat of air and water, respectively.

The performance of the heat exchangers is analyzed by conventional ϵ -NTU technique, the effectiveness is defined as

$$\epsilon = \frac{Q}{Q_{\max}}, \quad (3.10)$$

or

$$\epsilon = \frac{Q}{(\dot{m}Cp)_{\min} \Delta T_{\max}}, \quad (3.11)$$

The relationship of the effectiveness and the number of transfer unit (NTU) for the present 4-tube configuration is as follow (ESDU [10]);

$$\epsilon = \frac{1}{C^*} \left\{ 1 - e^{-4KC^*} \left[1 + C^* K^2 (6 - 4K + K^2) + 4(C^*)^2 K^4 (2 - K) + \frac{8(C^*)^3 K^6}{3} \right] \right\}, \quad (3.12)$$

$$K = 1 - e^{-NTU/4}, \quad (3.13)$$

$$NTU = \frac{UA}{(\dot{m}Cp)_{\min}}, \quad (3.14)$$

$$C^* = \frac{(\dot{m}Cp)_{\min}}{(\dot{m}Cp)_{\max}}, \quad (3.15)$$

Once the overall resistance is obtained from Equations (3.12-3.14), the heat transfer coefficients can be obtained from the following overall resistance equation:

$$\frac{1}{UA} = \frac{1}{\eta_o h_o A_o} + \frac{\ln(d_o/d_i)}{2\pi k L} + \frac{1}{h_i A_i}, \quad (3.16)$$

Where h is the heat transfer coefficient, A is the surface area, d is tube diameter, L is total tube length, k is thermal conductivity of tube material, η_o is surface efficiency and the subscripts o, i denote the air side and the tube side, respectively. The tube side heat transfer coefficient can be calculated from Gnielinski correlation [11] as

$$h_i = \left(\frac{k}{d} \right) \frac{(\text{Re}_{Di} - 1000) \text{Pr}(f_i/2)}{1 + 12.7 \sqrt{f_i/2} (\text{Pr}^{2/3} - 1)}, \quad (3.17)$$

$$f_i = [1.58 \ln(\text{Re}_{Di}) - 3.28]^{-2}, \quad (3.18)$$

Where Re_{Di} is the tube-side Reynolds number. The relation of the surface efficiency in Equation (3.16) and the fin efficiency η is

$$\eta_o = 1 - \frac{A_f}{A_o} (1 - \eta), \quad (3.19)$$

$$A_o = A_f + A_b, \quad (3.20)$$

Where A_o is the total surface area of finned tube, A_f is surface area of fin, A_b is the surface area of the bare tube. The fin efficiency η can be approximated from the Schmidt approximation [12]:

$$\eta = \frac{\tanh(mr\phi)}{mr\phi}, \quad (3.21)$$

where

$$m = \sqrt{\frac{2h_o}{k_f f_i}}, \quad (3.22)$$

$$\phi = \left(\frac{R_{eq}}{r} - 1 \right) \left[1 + 0.35 \ln \left(\frac{R_{eq}}{r} \right) \right], \quad (3.23)$$

$$\frac{R_{eq}}{r} = 1.27 \frac{X_M}{r} \left(\frac{X_L}{X_M} - 0.3 \right)^{1/2}, \quad (3.24)$$

$$X_L = \frac{\sqrt{(S_t/2)^2 + S_L}}{2}, \quad (3.25)$$

$$X_M = 0.5S_t, \quad (3.26)$$

where k_f is thermal conductivity of the fin, S_t and S_L are transverse and longitudinal pitches of the tube bank and f_i is fin thickness.

The total surface area of finned tube and the total surface area of the fin can be estimated by assuming equal to that of the circular fin as [13];

$$A_o = \frac{nL\pi}{f_s + f_i} (0.5(d_f^2 - d_o^2) + d_f f_i + d_o f_s), \quad (3.27)$$

$$A_f = \frac{nL\pi}{f_s + f_i} (0.5(d_f^2 - d_o^2) + d_f f_i), \quad (3.28)$$

where n is the total number of tube, d_f is outer diameter of finned tube, f_s is spacing between adjacent fin and f_t is fin thickness.

In this study, the pressure drop across tube bank is in the form of Fanning friction factor. The relation of the friction factor including the entrance and exit pressure loss [14] relative to the measured pressure drop ΔP is

$$f = \frac{A_c \rho_i}{A_o \rho_m} \left[\frac{2 \rho_i \Delta P}{G_c^2} - \left(1 + \sigma^2 \right) \left(\frac{\rho_i}{\rho_o} - 1 \right) \right], \quad (3.29)$$

where A_c is the minimum flow area, σ is the contraction ratio, G_c is the mass flux of air flow based on the minimum flow area and the subscripts i , o , m represent the inlet, outlet, and mean value, respectively.

3.3 Results and Discussion

3.3.1 Inline Arrangement

In the case of inline arrangement, there are nine samples of inline arrangement as shown in Table 3.1. Figure 3.1 shows the effect of tube diameter on the airside performance. Results are termed as heat transfer coefficient and the pressure drop vs. frontal velocity. In this comparison, the fin spacing (3.85 mm), fin thickness (0.4 mm), and the fin height (10 mm) are all the same. The transverse and the longitudinal tube pitches are 50 mm. As expected, the pressure drop rises with the tube diameter. However, it is interesting to note that the heat transfer coefficient increases with the reduction of tube diameter. It is likely that this phenomenon is attributed to the ineffective area behind the tube increases with the tube diameter. The ineffective area is especially pronounced for an inline arrangement. Wang et al. [14] performed flow visualizations via dye injection technique for fin-and-tube heat exchangers having inline arrangement. Their visual results unveil a very huge flow circulation behind the tube row. Consequently this huge recirculation not only contributes to the decrease of heat transfer coefficient but also to the rise of pressure drop. In addition, the huge recirculation may also block the subsequent tube row and degrade the heat transfer performance hereafter.

Figure 3.2 shows the effect of fin height on the airside performance for inline arrangement. In this comparison, the associated fin heights are 10 and 15 mm and the fin spacing and the tube diameter are 3.85 mm and 21.7 mm with the transverse and the longitudinal pitches are 71.4 and 50 mm, respectively. As seen in the figure, the influence of fin height shows tremendous influence on the heat transfer performance and the

pressure drop. Although the increase of fin height is only 50%, the corresponding increase of fin surface area is roughly 53% that eventually leads to a dramatic increase of pressure drop (roughly 100%). Unlike those of pressure drops, the heat transfer coefficients are dropped drastically with the increase of fin height. This is probably due to the airflow bypass effect. Actually the airflow is prone to flowing the portion where the flow resistance is small. 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.

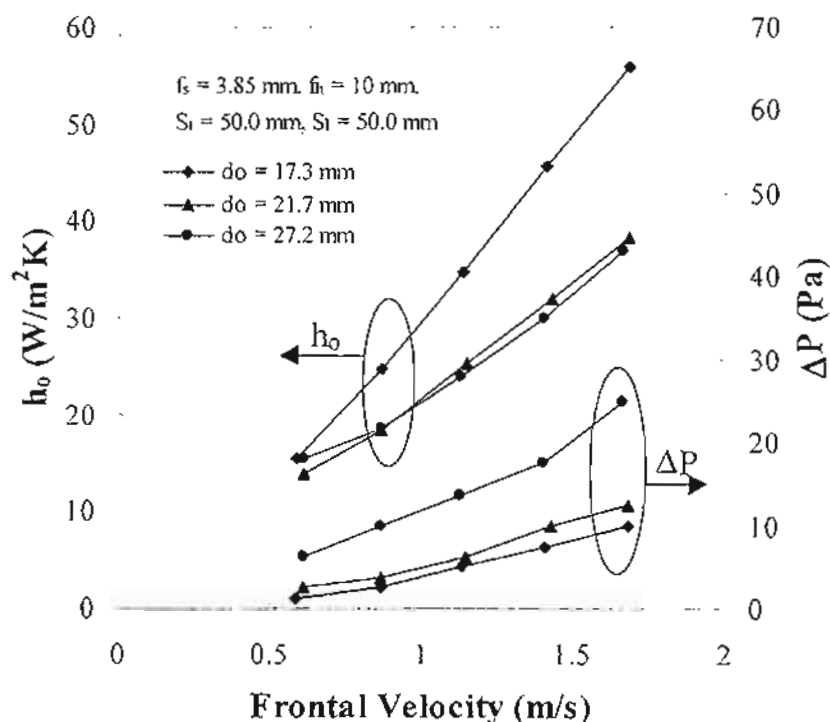


Figure 3.1 Effect of tube diameter on the airside performance for the inline arrangement.

The effects of the fin spacing on the airside performance at different transverse tube pitch (50 and 71.4 mm) are shown in Figure 3.3. At a larger transverse tube pitch of 71.4 mm, one can see the effect of fin spacing on pressure drops is rather small. Surprisingly, the reduction of the fin spacing does not give rise to the pressure drops. This may be related to the presence of the bypass airflow between the tubes. Since the bypass airflow does not increase the pressure drops, therefore no detectable change of pressure drops is seen at a very large transverse tube pitch of 71.4 mm. On the contrary, the

reduction of fin spacing give rise to pressure drops at a smaller transverse tube pitch of $S_t = 50$ mm. This is due to the significant decrease of bypass airflow and the reduction of the associated hydraulic diameter. From Figure 3.3, it is also found that the increase of fin spacing give rise to heat transfer coefficient. An explanation of this phenomenon is the same as the effect of fin height that the result comes from airflow bypass effect. As is known, the corresponding pressure drop rises with the decrease fin spacing. Therefore, lower fin spacing gets higher bypass airflow and the heat transfer coefficient is decreased at smaller fin spacing.

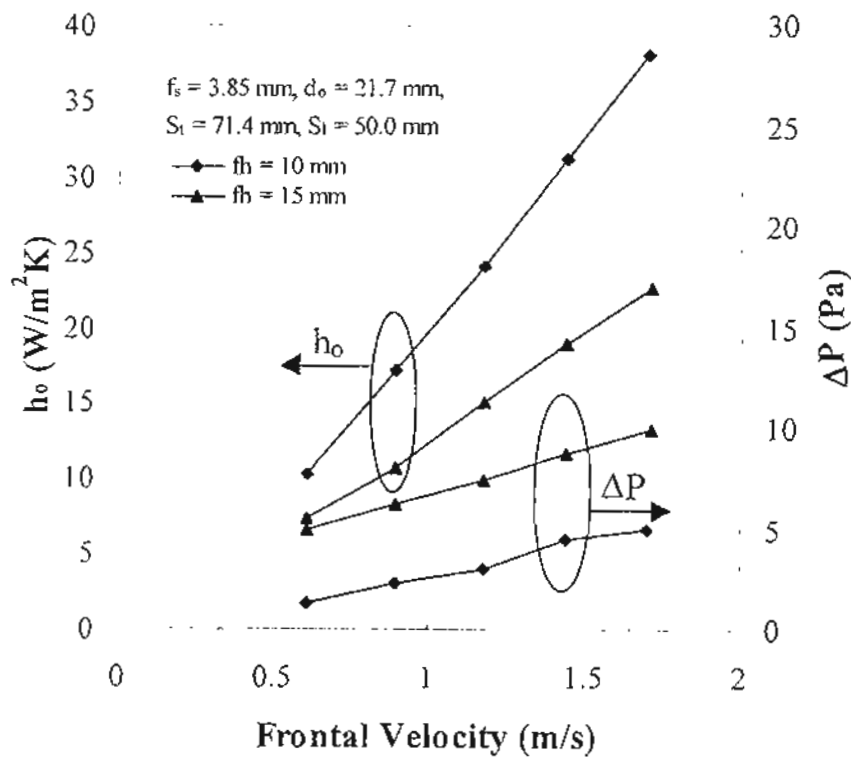


Figure 3.2 Effect of fin height on the airside performance for the inline arrangement.

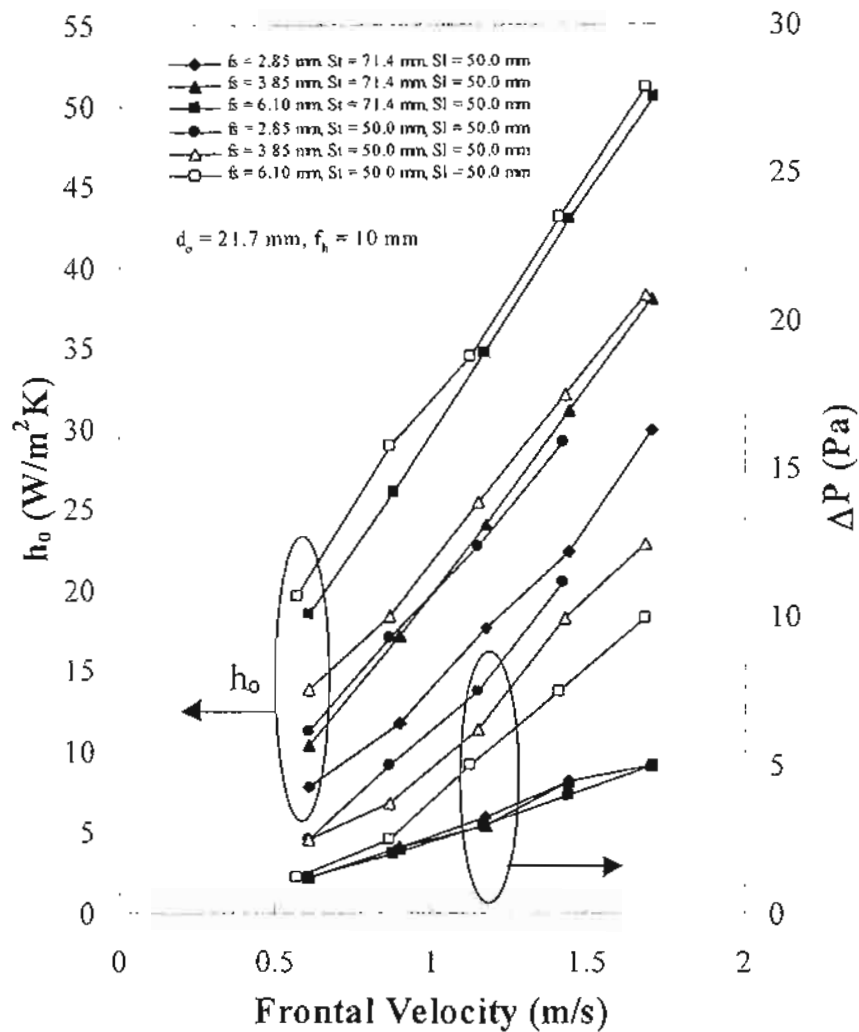


Figure 3.3 Effect of fin spacing and tube arrangement on the airside performance for the inline arrangement.

3.3.2 Staggered Arrangement

For staggered arrangement, Figure 3.4 shows effect of tube diameter on the air side performance. The geometrical parameters of fin spacing, fin height, transverse pitch and longitudinal pitch are 3.85 mm, 10 mm, 50 mm and 4.33 mm, respectively. Analogous to that of inline arrangement but the influence is comparatively less to some extent, the heat transfer coefficient and the pressure drop decrease with the tube diameter. Explanation of this phenomenon is the same as that of inline arrangement. However, the recirculation region for staggered arrangement behind the tube is much smaller than that of the inline arrangement due to the airflow across the adjacent tubes is directed by the subsequent

tube row. The directed oblique airflow will reduce the recirculation area behind the tube. This phenomenon can be also made clear from the flow visualization experiment by Wang et al. [15] in staggered arrangement. Their visualization indicated a much smaller recirculation area of the staggered arrangement than that of inline arrangement.

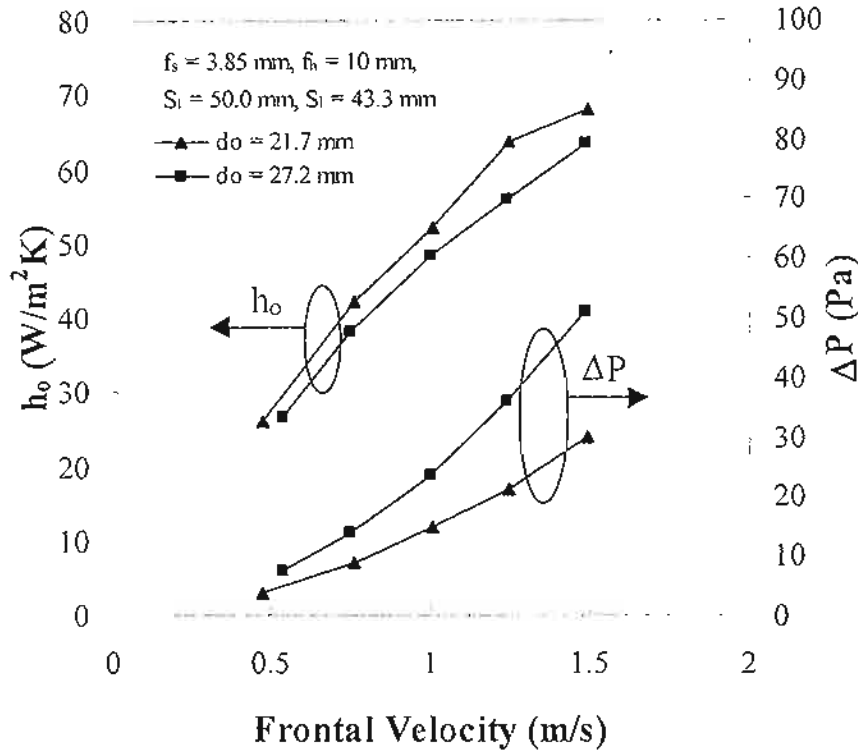


Figure 3.4 Effect of tube diameter on the airside performance for the staggered arrangement.

The effect of fin height on the airside performance is shown in Figure 3.5. It is found that the heat transfer coefficients of $f_h = 10 \text{ mm}$ are slightly higher than those of $f_h = 15 \text{ mm}$ and the explanation is similar to that of inline arrangement. However, the pressure drops for both fin height are about the same irrespective of the fin area of $f_h = 15 \text{ mm}$ are approximately 53% higher than that of $f_h = 10 \text{ mm}$. The results are quite different from those of inline arrangement in which one can see an approximately two times increase of pressure drops for $f_h = 15 \text{ mm}$ relative to that of $f_h = 10 \text{ mm}$. Explanation of this phenomenon can be made clear as follows. For the same fin height of $f_h = 10 \text{ mm}$, one can see a dramatic increase of pressure drops of six times for staggered arrangement relative to that of inline arrangement. This implies the major contribution to the total pressure drops comes from the type of arrangement. For staggered arrangement, the airflow is directed by the subsequent tube row in which most of the pressure loss is

generated. On the contrary, since the air flow across the inline arrangement is not directed by the subsequent tube row. The pressure drop is mainly affected by the friction caused by the fins and the recirculation area behind the tube. Thus one can see a very small increase of the pressure drop with the rise of fin height for staggered arrangement.

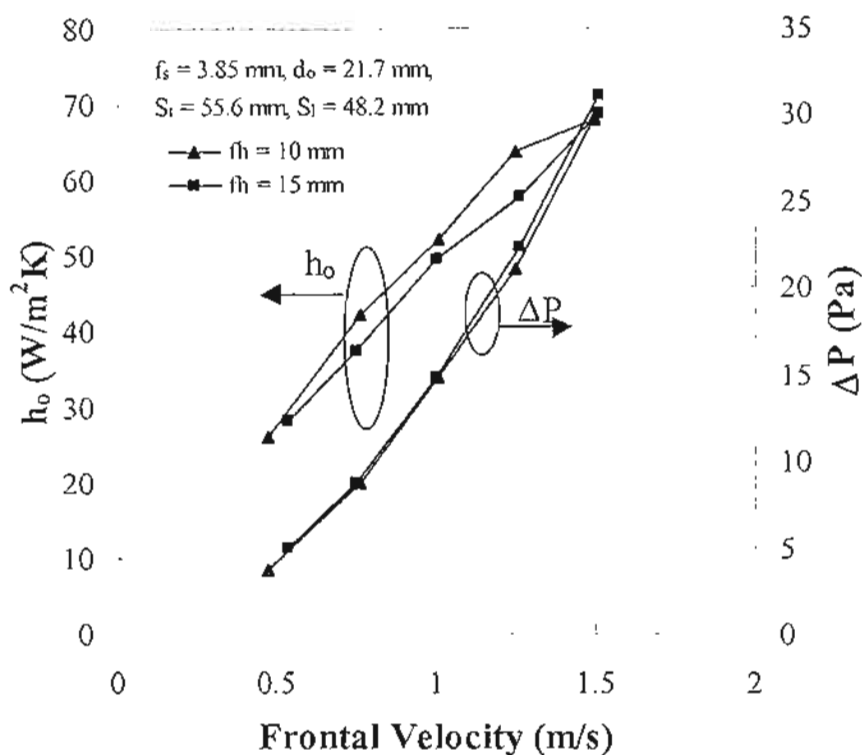


Figure 3.5 Effect of fin height on the airside performance for the staggered arrangement.

Figure 3.6 shows the effect of fin spacing, and S_t on the airside heat transfer coefficients and pressure drops. The tube diameter is 21.7 mm and the fin height is 10 mm. From Figure 3.5, it can be concluded that higher fin spacing gives lower pressure drops. 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 [16] and Wang et al. [17]. However, for a larger transverse pitch of 84 mm, the heat transfer coefficients decrease with the decrease of fin spacing. Again explanation of 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 reduced. Therefore, one can see no appreciable change of heat transfer coefficients for $S_t = 50$ mm.

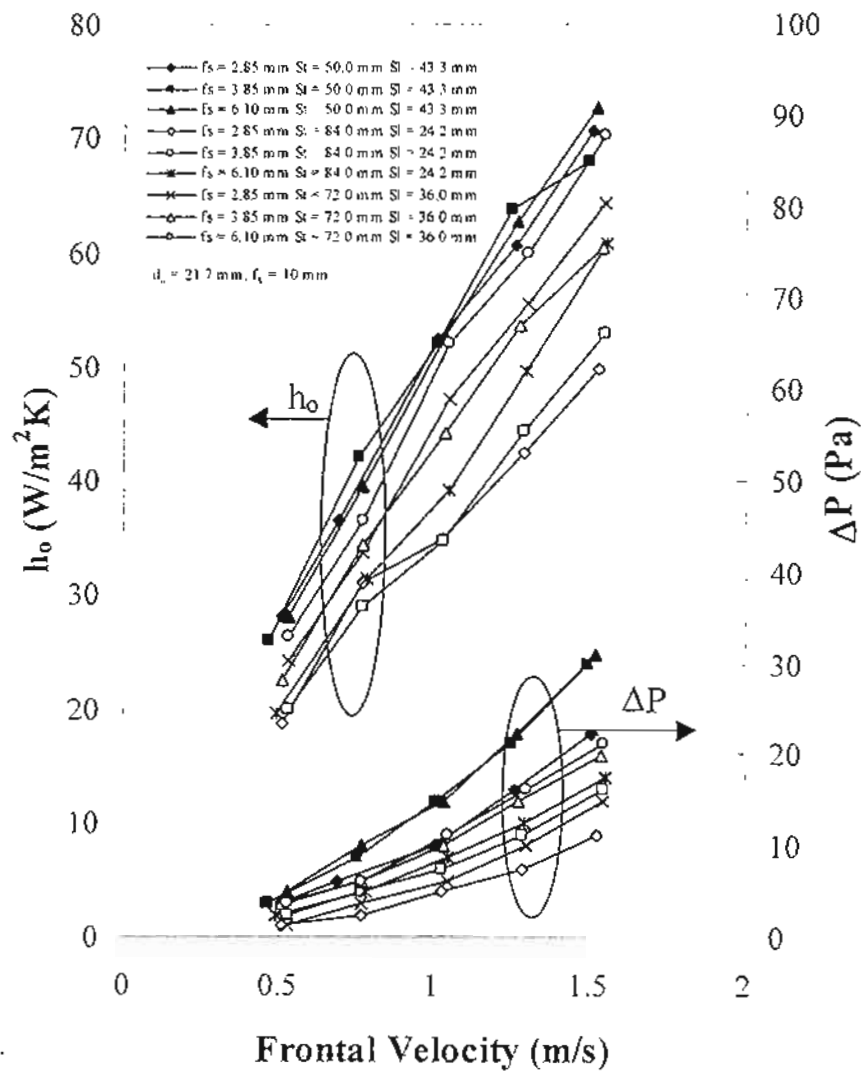


Figure 3.6 Effect of fin spacing and tube arrangement on the airside performance for the staggered arrangement.

3.3.3 Empirical Correlations

Based on the previous discussions, it is obvious from the test data that no single curve can be expected to describe the complex behaviors about the heat transfer and frictional characteristics in both inline and staggered arrangements. For easier engineering

calculations, we had performed multiple linear regression technique to obtain the relevant correlations. The corresponding correlations are given as follows:

Correlation of the heat transfer performance of the inline arrangement:

$$j = 3.9048 \times 10^{-4} \text{Re}_D^{0.0637} \left(\frac{f_t}{f_s} \right)^{-0.8363} \left(\frac{S_t}{S_l} \right)^{1.9926} \left(\frac{S_t}{d_o} \right)^{2.2810} \left(\frac{d_f}{d_o} \right)^{-2.1720} \quad (3.30)$$

Correlation of the heat transfer performance of the staggered arrangement:

$$j = 0.1970 \text{Re}_D^{-0.1295} \left(\frac{f_t}{f_s} \right)^{-0.1452} \left(\frac{S_t}{S_l} \right)^{1.1874} \left(\frac{S_t}{d_o} \right)^{0.8238} \left(\frac{d_f}{d_o} \right)^{0.0010} \quad (3.31)$$

Correlation of the frictional performance of the inline arrangement:

$$f = 0.1635 \text{Re}_D^{-0.4172} \left(\frac{f_t}{f_s} \right)^{-0.5215} \left(\frac{S_t}{S_l} \right)^{-1.2235} \left(\frac{S_t}{d_o} \right)^{-0.6334} \left(\frac{d_f}{d_o} \right)^{1.2000} \quad (3.32)$$

Correlation of the frictional performance of staggered arrangement:

$$f = 2.1768 \text{Re}_D^{-0.2679} \left(\frac{f_t}{f_s} \right)^{-0.2468} \left(\frac{S_t}{S_l} \right)^{1.8680} \left(\frac{S_t}{d_o} \right)^{0.3011} \left(\frac{d_f}{d_o} \right)^{-0.4470} \quad (3.33)$$

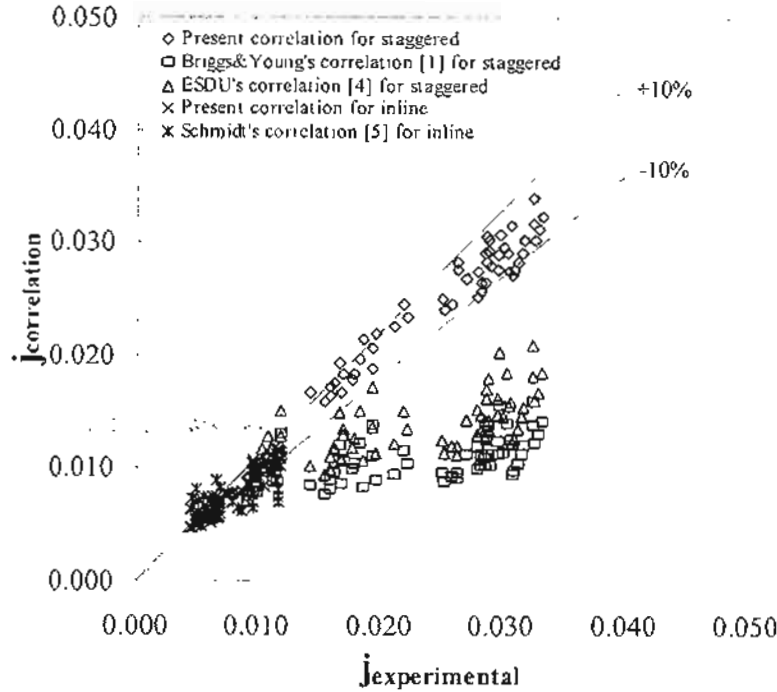


Figure 3.7 Comparison of the heat transfer correlations with experimental data.

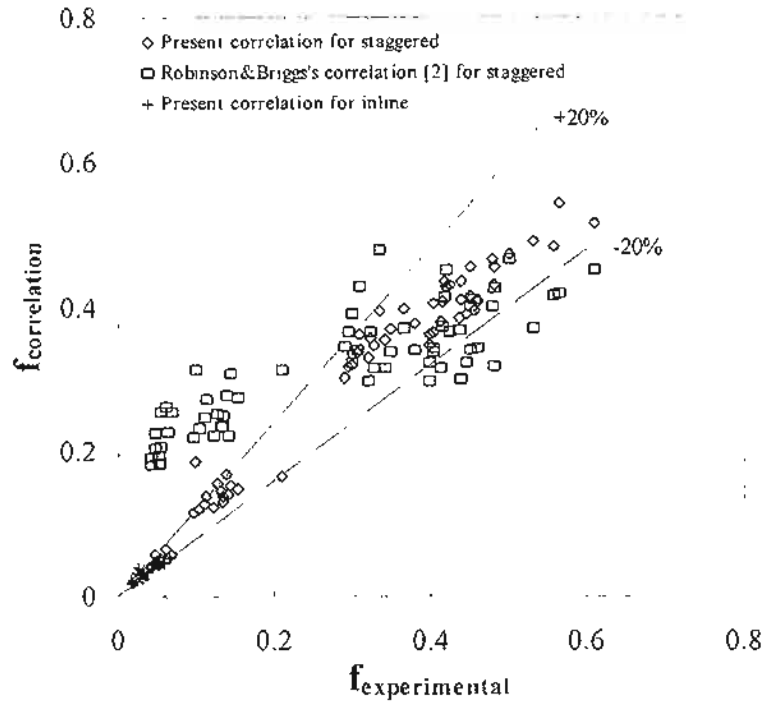


Figure 3.8 Comparison of the frictional correlations with experimental data.

Figures 3.7-3.8 show the comparison of j and f of the experimental results with the proposed correlations. For the heat transfer correlations, Equations (3.30)-(3.31) can predict 89.7% and 88.4% of the experimental data with $\pm 10\%$ and the friction factor correlations, Equations (3.32)-(3.33) give a predictive ability of 79.6% and 91.3% of the experimental data within $\pm 20\%$. The standard deviation of the correlations Equations (3.30)-(3.33) are 7.90%, 7.15%, 16.70%, and 13.7%, respectively.

The comparison of the proposed correlations and the previous correlations are also investigated. In case of the inline arrangement, it is found that Schmidt's correlation [5] can predicts 71.8% of the experimental data with in $\pm 30\%$. This result comes from the lower Reynolds number of air stream of the experimental. Moreover, Schmidt's correlation did not include the effect of the geometrical of finned tube. Therefore, lower accuracy is obtained.

In case of the staggered arrangement, the correlations of Briggs and Young [1] and ESDU [4] give approximately 30% lower than the present correlation. The explanation of this result is the shape of the crimped spiral finned tube differs from that of the circular finned tube and the effect of crimped edge pronounce the heat transfer rate of the heat exchanger. The friction factor correlation of Robinson and Briggs [2] gives over

predicting of the experimental value at low friction factor. However, at higher friction factor, the accuracy is close to that of the present correlation.

3.4 Conclusion

The present experimental study reports the airside performance of the crimped spiral fin heat exchanger. The effects of tube diameter, fin spacing, transverse tube pitch, and tube arrangements are examined. On the basis of previous discussions, the following conclusions are made:

5. For an inline arrangement, the pressure drops increase with the rise of tube diameter but the associated heat transfer coefficients decrease with it. The increase of fin height also gives rise to considerable increase of pressure drop but decrease of heat transfer coefficient.
6. For the inline arrangement, the effect of fin spacing on the airside performance varies with the transverse tube pitch. For a larger transverse tube pitch of 71.4 mm, there is an effect on heat transfer coefficient but no detectable influence of the fin spacing on frictional characteristics. It is likely that this phenomenon is related to the considerable airflow bypass between tube rows. On the contrary, at a smaller transverse tube pitch of 50 mm, one can see smaller fin spacing results in higher pressure drops and lower heat transfer coefficients.
7. For the staggered arrangement, the effect of tube diameter on the airside performance is analogous to that of inline arrangement but to a comparatively small extent. This is because the recirculation zone behind the tube row is much smaller in a staggered arrangement. The effect of the fin height on the pressure drops is much smaller than that of inline arrangement due to the major contribution to the pressure drops is from the blockage of the subsequent tube row in a staggered arrangement.
8. The effect of fin spacing on the airside performance for staggered arrangement also varies with the transverse tube pitch. For a smaller transverse tube pitch of 50 mm, there is no appreciable influence of the fin spacing on heat transfer. On the contrary, at a larger transverse tube pitch of 84 mm, one can see smaller fin spacing leads to lower heat transfer coefficients. This is also attributed to the presence of airflow pass effect.
6. Correlations of the present crimped spiral fins in both staggered and inline arrangement are developed. The proposed correlations give fairly good predictive ability against the present test data.

CHAPTER 4

AIR-SIDE PERFORMANCE OF HEAT EXCHANGER USING CRIMPED SPIRAL FINS, A CASE STUDY OF COOLING COIL

4.1 Introduction

For practical waste heat recovery system, the heat exchanger may accompany with condensation of moisture air on the heat exchanger surface. Although the designers try to avoid this situation due to considerably corrosive problem associated with it, condensation may still take place from time to time. This is commonly encountered if the load is not constant such as small boilers where the steam consumption varies with time and the flue gas temperature fluctuates in wide range. In that regard, the airside performance in the presence of dehumidification is rather important. Unfortunately, there are simply no data reported for the crimped spiral finned heat exchangers. Hence, it is the objective of this work is to report the heat transfer and friction characteristic of cross flow heat exchanger using crimped spiral fin in the presence of dehumidification. Moreover, the heat transfer and friction correlations are also developed in this work.

4.2 Data Reduction

The heat transfer rate of cross flow heat exchanger under dehumidifying condition can be calculated as follows:

$$Q_a = \dot{m}_a (i_{a,in} - i_{a,out}), \quad (4.1)$$

$$Q_w = \dot{m}_w C_{pw} (T_{w,out} - T_{w,in}). \quad (4.2)$$

Note that Equation (4.1) and Equation (4.2) denote heat transfer rate in the air-side and the tube-side, respectively. In this study, the mathematical average of the heat rate is used, i.e.,

$$Q_{avg} = 0.5(Q_a + Q_w). \quad (4.3)$$

The average heat transfer rate is related to the rate equation given in the following (enthalpy based potential),

$$Q_{avg} = U_{o,w} A_o F \Delta i_m. \quad (4.4)$$

Where F is the correction factor of unmixed/unmixed configuration.

The log-mean enthalpy potential Δi_m is (Threlkeld [18])

$$\Delta i_m = \frac{(i_{a,in} - i_{r,out}) - (i_{a,out} - i_{r,in})}{\ln \left(\frac{i_{a,in} - i_{r,out}}{i_{a,out} - i_{r,in}} \right)} \quad (4.5)$$

Myers [19] derived the enthalpy-based overall heat transfer coefficient ($U_{o,w}$) to individual resistance as

$$\frac{1}{U_{o,w}} = \frac{b'_r A_o}{h_i A_{p,i}} + \frac{b'_p x_p A_o}{k_p A_{p,m}} + \frac{1}{h_{o,w} \left(\frac{A_{p,o}}{b'_{w,p} A_o} + \frac{A_f \eta_{f,wet}}{b'_{w,m} A_o} \right)} \quad (4.6)$$

where

$$h_{o,w} = \frac{1}{\frac{C_{p,a}}{b'_{w,m} h_{c,o}} + \frac{y_w}{k_w}} \quad (4.7)$$

Note that the ratio of water film thickness and thermal conductivity of water (y_w / k_w) is very small compared to other terms [20] and it is neglected in this study.

The tube side heat transfer coefficient can be calculated from Gnielinski correlation [11] as

$$h_i = \frac{(f_i / 2)(\text{Re}_{Di} - 1000)\text{Pr}}{1.07 + 12.7\sqrt{f_i / 2}(\text{Pr}^{2/3} - 1)} \left(\frac{k_i}{d_i} \right) \quad (4.8)$$

where

$$f_i = \frac{1}{(1.58 \ln \text{Re}_{Di} - 3.28)^2} \quad (4.9)$$

The four quantities in Equation 4.7 can be estimated follow the method of Wang et al. [20] base on the enthalpy-temperature ratios. b'_r and b'_p , they can be calculated as

$$b'_r = \frac{i_{s,p,i,m} - i_{r,m}}{T_{p,i,m} - T_{r,m}} \quad (4.10)$$

$$b'_p = \frac{i_{s,p,o,m} - i_{s,p,i,m}}{T_{p,o,m} - T_{p,i,m}} \quad (4.11)$$

The quantity $b'_{w,p}$ is the slope of saturated enthalpy curve evaluated at the outer mean water film temperature at the base surface and can be approximated at the slope of saturated enthalpy curve evaluated at the base surface temperature of tube [20]. However, the quantity $b'_{w,m}$, which defines as the slope of saturated enthalpy curve evaluated at the outer mean water film temperature at the fin surface, can not be calculated directly.

Consequently, a trial and error procedure of iteration is needed (Wang et al. [20]) Detailed procedures are in the following:

1. Assume a value of $T_{w,m}$ and calculate the quantity $b'_{w,m}$
2. Calculate $h_{o,w}$ from Equation 4.6
3. Calculate the quantity $i_{s,w,m}$ by this following relation

$$i_{s,w,m} = i - \frac{C_{p,a} h_{o,w} \eta_{f,wet}}{b'_{w,m} h_{c,o}} \times \left(1 - U_{o,w} A_o \left[\frac{b'_r}{h_i A_{p,i}} + \frac{x_p b'_p}{k_p A_{p,m}} \right] \right) (i - i_{r,m}). \quad (4.15)$$

4. Determine the new $T_{w,m}$ at $i_{s,w,m}$ and repeat the procedure again until the tolerance is met.

The wet fin efficiency is calculated as (Wang et al [20]):

$$\eta_{f,wet} = \frac{2r_i}{M_T(r_o^2 - r_i^2)} \times \left[\frac{K_1(M_T r_i) I_1(M_T r_o) - K_1(M_T r_o) I_1(M_T r_i)}{K_1(M_T r_o) I_0(M_T r_i) + K_0(M_T r_i) I_1(M_T r_o)} \right], \quad (4.16)$$

where

$$M_T = \sqrt{\frac{2h_{o,w}}{k_f f_t}} = \sqrt{\frac{2h_{c,o}}{k_f f_t}} \sqrt{\frac{b'_w}{C_{p,a}}}. \quad (4.17)$$

In this research work, the sensible heat transfer coefficient ($h_{c,o}$) and the pressure drop of air stream across tube bank are presented in terms of the Colburn factor (j) and the friction factor (f) factors,

$$j = \frac{h_{c,o}}{G_{max} C_{p,a}} \text{Pr}^{2/3}, \quad (4.18)$$

$$f = \frac{A_{mn}}{A_o} \frac{\rho_i}{\rho_m} \left[\frac{2\rho_i \Delta P}{G_c^2} - \left(1 + \sigma^2 \right) \left(\frac{\rho_i}{\rho_o} - 1 \right) \right]. \quad (4.19)$$

4.3 Results and Discussion

4.3.1 Staggered Arrangement

4.3.1.1 Pressure Drop

The associated pressure drops for all the test samples are shown in Figures 4.1-4.4. In Figure 4.1, the influence of tube diameter is examined. As seen in the figure, the pressure drops for wet condition is only slightly higher than that of dry condition. This is because the fin spacing in this figure is comparatively large (3.85 mm). In that respect, the condensate can easily drain without adhering to the interspacing of fins, thereby

giving only a slight increase of pressure drops of the wet surface relative to dry conditions. However, one can find a considerable influence of tube size on the total pressure drop. For the same frontal velocity of 1.5 m/s, the associated pressure drop for $d_o = 27.2$ mm is roughly 2.5 times higher than that of $d_o = 17.3$ mm.

The effect of fin height on the total pressure drops is shown in Figure 4.2. The pressure drops increase with fin height because of more fin surface is provided. The effective surface area of $f_h = 0.015$ m is roughly 30% higher than that of $f_h = 0.01$ m and the corresponding increase of pressure drop is also around 30~40% which indicate a linear relationship of the fin height and total pressure drop. Conversely, one can go back to Figure 4.1 where the effective surface area increase caused by the tube size is less than 10% because the surface area is dominated by secondary surface (fins). However, the pressure drops is greatly increased with the tube size. The excess pressure drop are attributed to (1) large form drag of the large tube; and (2) ineffective flow separation zone behind the tube which may increase its influence to further downstream and results in more pressure drops associated with it. This phenomenon is analogous to the continuous fin pattern reported by Wang et al. [21].

The effect of fin spacing on the pressure drops is shown in Figure 4.4. Notice that there is very small difference between dry and wet conditions since the tube diameter and fin height is relatively small which in term helps to drain the water condensate. Again smaller fin spacing increases the effective surface area and correspondingly higher pressure drops. In essence, the pressure drop increases with tube diameter (d_o), fin height (f_h) and decrease with it for a smaller fin spacing (f_s). Among them, the influence of tube diameter is most pronounced. The effect of tube arrangement is also found in Figure 4.3-4.4. Higher transverse pitch of tube bank (S_t) gives rise to lower pressure drop. Again this is also attributed to the increase of surface area. The relevant influences of geometric parameters on the friction characteristics are correlated in terms of friction factor, and is given as,

$$f = 17.02 \text{Re}_D^{-0.5636} \left(\frac{d_o}{S_t} \right)^{0.3956} \left(\frac{f_h}{f_s} \right)^{-0.3728} \left(\frac{S_t}{S_l} \right)^{1.2804} \left(\frac{d_o}{d_f} \right)^{-0.1738} \quad (4.20)$$

From Figure 4.5, one can see the proposed friction factor correlation can predict 75% of the experimental data within $\pm 15\%$ accuracy.

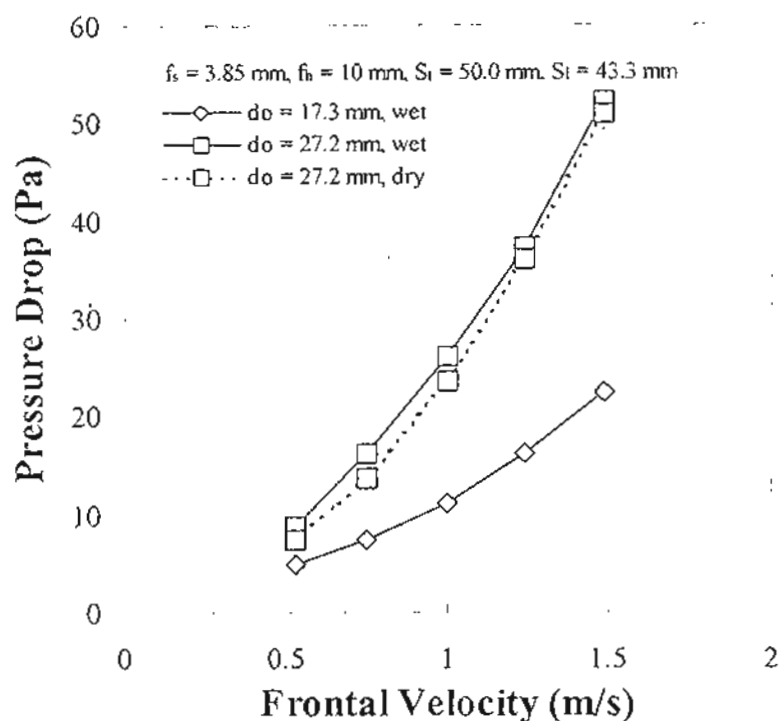


Figure 4.1 Comparisons of the pressure drop in dry and wet condition in case of different tube diameter.

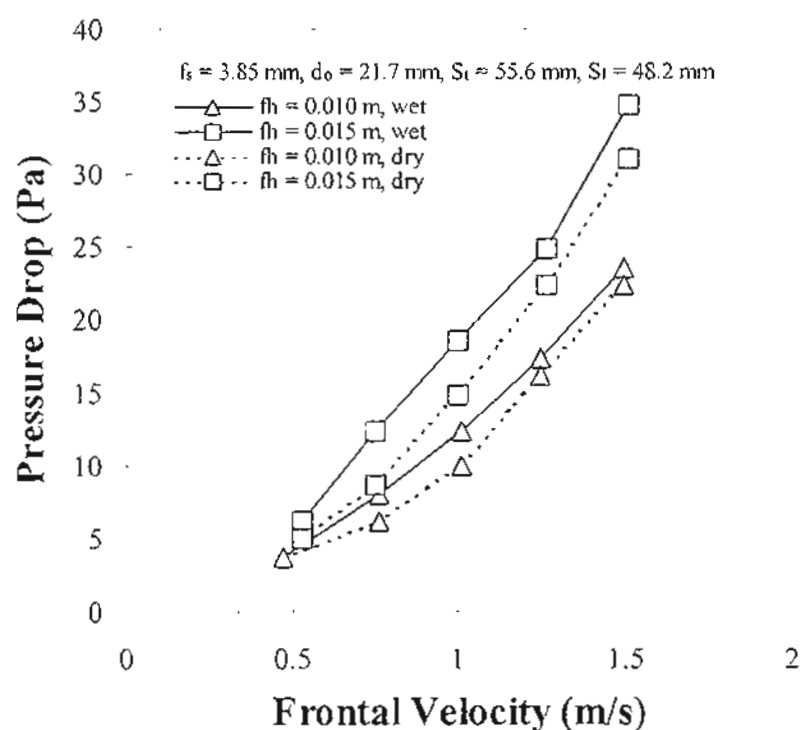


Figure 4.2 Comparisons of the pressure drop in dry and wet condition in case of different fin height.

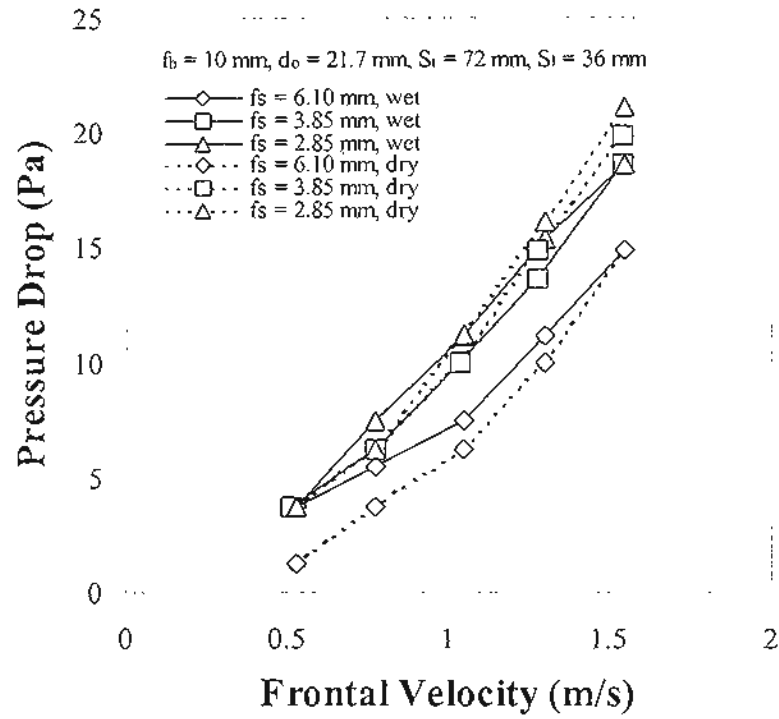


Figure 4.3 Comparisons of the pressure drop in dry and wet condition in case of different fin spacing at $S_i = 72$ mm and $S_t = 36$ mm.

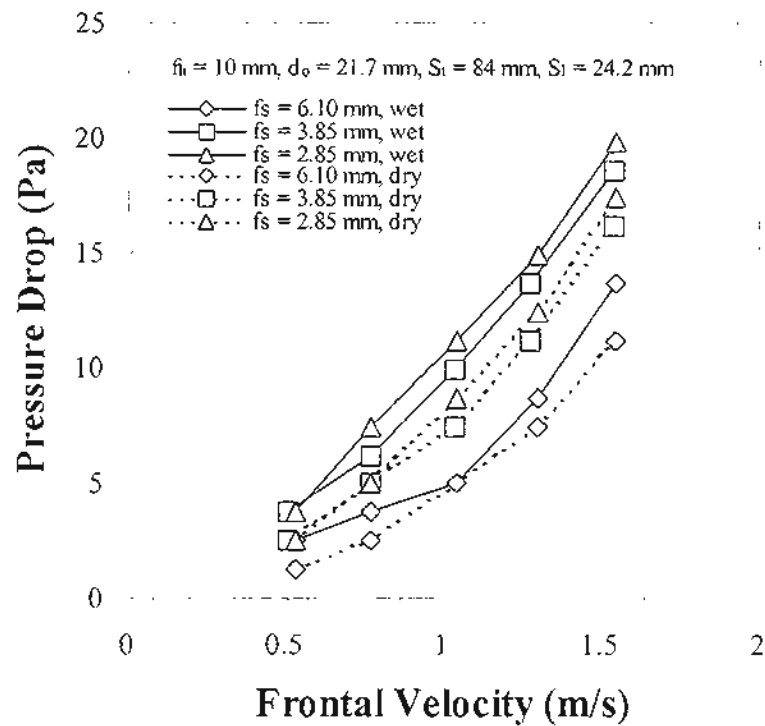


Figure 4.4 Comparisons of the pressure drop in dry and wet condition in case of different fin spacing at $S_i = 84$ mm and $S_t = 24.2$ mm.

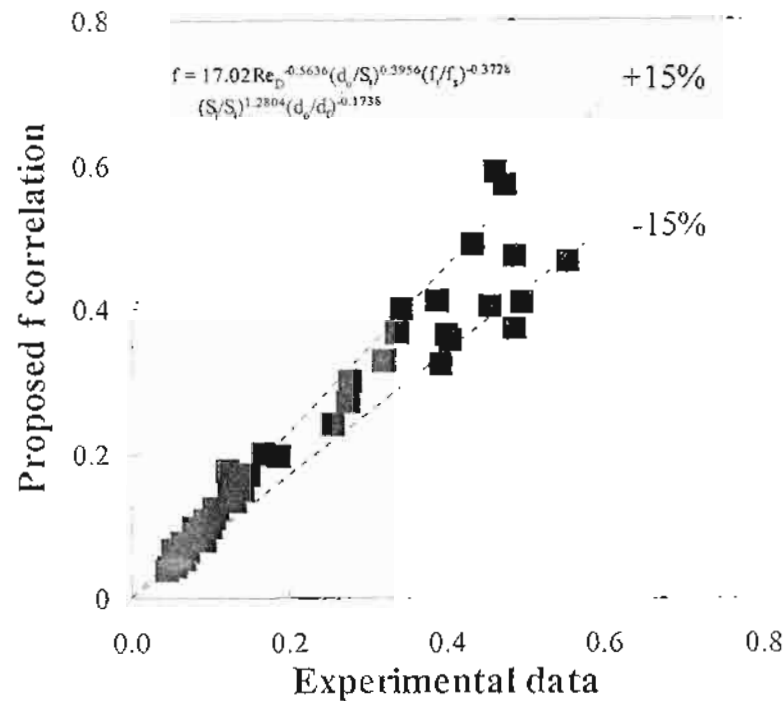


Figure 4.5 Comparison of the frictional data with the proposed correlation.

4.3.1.2 Sensible heat transfer coefficient

The related heat transfer coefficients vs. frontal velocity for all the test samples are shown in Figures 4.6-4.10. For comparison purpose, the relevant heat transfer coefficient in dry condition is also shown in the figure. It is found that the heat transfer coefficient of wet surface is slightly lower than that of dry surface. Actually, there are many studies showing the comparison of the heat transfer coefficients between wet and dry surface heat exchanger. Some studies indicated that the heat transfer coefficient is augmented in wet surface conditions, such as Myers [19], Elmahdy [22] and Eckels and Rabas [23] who reported results for the continuous plate finned tube. These investigators argued that the presence of water condensate may roughen the heat transfer surface, leading to higher heat transfer coefficient. However, some of the studies showed a drop of heat transfer coefficient of wet surface, such as the wavy finned tube heat exchanger by Mirth and Ramadhyani [24] who reported about 17-50% decreasing of heat transfer coefficient of wet surface. One possible cause of the degradation is due to water film resistance and condensate blocking. Moreover Wang et al. [20] shows the decreasing of the Colburn j factor of plate finned tube heat exchangers when the Reynolds number lower than 2,000. However, at the higher Reynolds number, the j factor of wet surface is

slightly higher than that of dry surface. The present results are generally in agreement with the trend of Wang et al. [20].

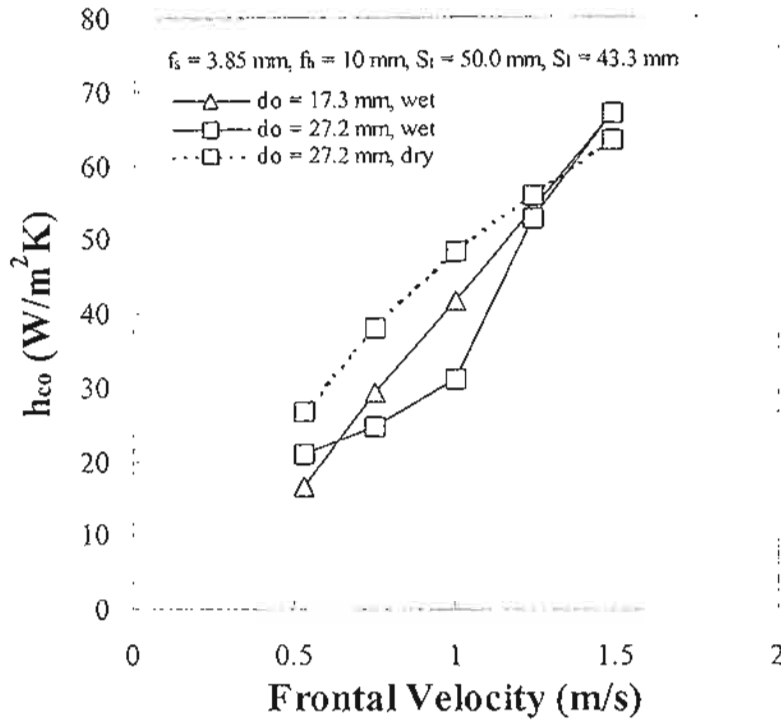


Figure 4.6 Comparisons of the heat transfer coefficient in dry and wet condition in case of different tube diameter.

The effect of tube diameter on the heat transfer performance is shown in Figure 4.6, it is found that the larger tube diameter ($d_o = 27.2$ mm) has a lower heat transfer coefficient than that of the small one ($d_o = 17.3$ mm). This phenomenon is attributed to the ineffective area behind the tube increases with the tube diameter. Wang et al. [15] performed flow visualizations via dye injection technique for fin-and-tube heat exchangers having inline arrangement. Their visual results unveil a very huge flow circulation behind the tube row. Consequently this huge recirculation not only contributes to the decrease of heat transfer coefficient but also to the rise of pressure drop as shown in the preceding discussions. In addition, the huge recirculation may also block the subsequent tube row and degrade the heat transfer performance hereafter.

Figure 4.7 shows the effect of fin height on the heat transfer coefficient. As seen in the figure, the influence of the fin height is negligible in wet condition. For the fully dry case, the previous chapter reports the 15 mm fin height gives lower heat transfer

coefficient than that of 10 mm. This result comes from the airflow by pass effect. Actually the airflow is prone to flowing the portion where the flow resistance is small. In case of $f_h = 15$ mm, the airflow resistance across the finned tube portion is larger than that of $f_h = 10$ mm. Therefore, part of the directed airflow just bypass the tube row without effective contribution to the heat transfer, hence showing a lower heat transfer coefficient. However, in case of wet surface, the condensate of water vapor covering the surface of heat exchanger become dominant and it increases the airflow resistance around tube. Therefore the effect of fin height is comparatively reduced.

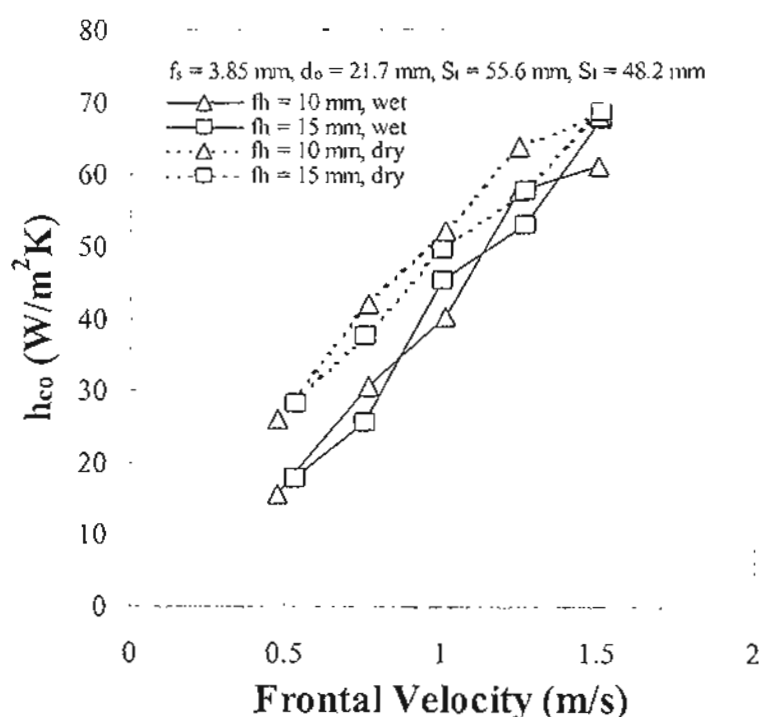


Figure 4.7 Comparisons of the heat transfer coefficient in dry and wet condition in case of different fin height.

The effect of fin spacing shown in Figures 4.8 and 4.9. As seen in the figure, the effect of fin spacing is small. However, small fin spacing tends to have lower heat transfer coefficient, and this is particularly pronounced in Figure 4.9 where $S_i = 84$ mm and $S_o = 24.2$ mm. This result may be related to the water condensate effect. The presence of water condensate increases the air flow resistances into the heat exchanger, thereby causing more airflow to bypass. This phenomenon becomes more significant with $S_i = 84$ mm because more bypass airflow. In that regard, one can see a moderately

decrease of heat transfer coefficient at smaller fin spacing. The reports of McQuiston [25,26] also show the decreasing of sensible heat transfer coefficient when the fin density is higher than 10 fins per inch.

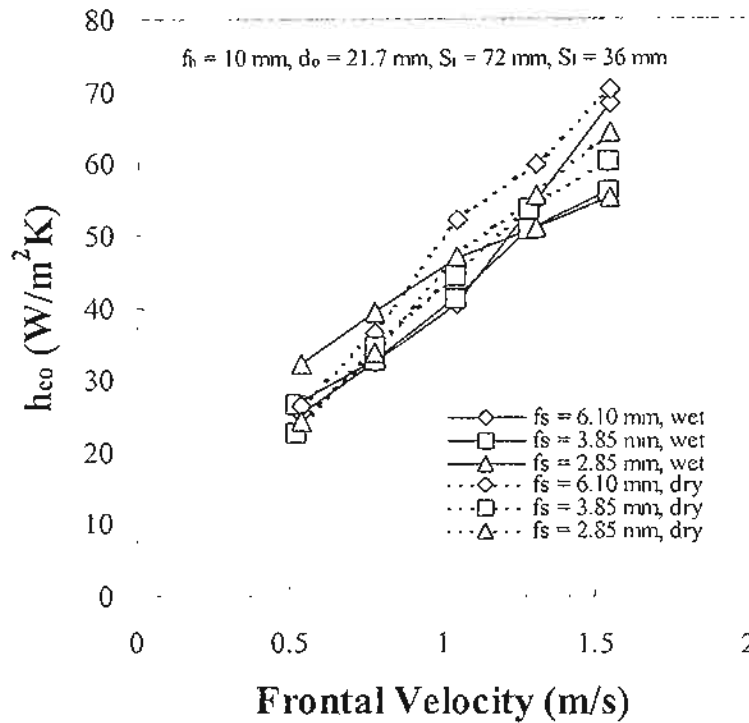


Figure 4.8 Comparisons of the heat transfer coefficient in dry and wet condition in case of different fin spacing at $S_t = 72$ mm and $S_f = 36$ mm.

Figure 4.10 shows the effect of tube arrangement on the sensible heat transfer coefficient. It is found that higher transverse tube pitch tends to have lower heat transfer coefficient and again this result is attributed to the airflow bypass effect. In case of $S_t = 84$ mm and $S_f = 24.2$ mm, more airflow is prone to flowing across the space between adjacent tube. Therefore, the amount of air stream contributed to the heat transfer is decreased when compared to that of $S_t = 72$ mm and $S_f = 36$ mm.

The associated effect of geometric parameters on the heat transfer performance are correlated in terms of the Colburn j factor, and is given as,

$$j = 0.0208 \text{Re}_D^m \left(\frac{d_o}{S_t} \right)^{-2.5950} \left(\frac{f_t}{f_s} \right)^{0.7905} \left(\frac{S_t}{S_f} \right)^{0.2391} \left(\frac{d_o}{d_f} \right)^{0.2761} \quad (4.21)$$

where

$$m = -0.2871 + 0.5322 \left(\frac{d_o}{S_t} \right) - 1.2856 \left(\frac{f_t}{f_s} \right) + 0.1845 \left(\frac{S_t}{S_i} \right). \quad (4.22)$$

In Figure 4.11, one can see the proposed j correlation can predict 95% of the experimental data within $\pm 15\%$ accuracy.

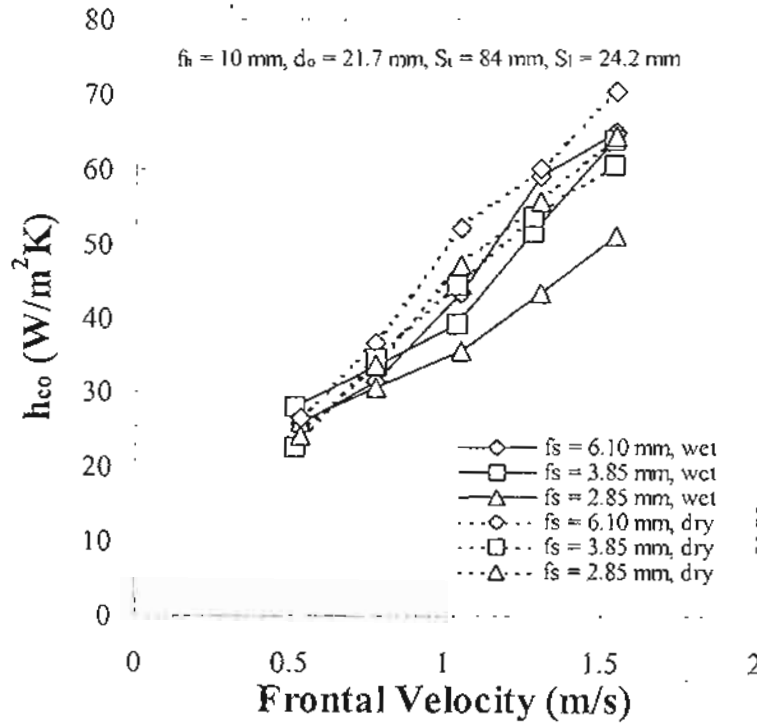


Figure 4.9 Comparisons of the heat transfer coefficient in dry and wet condition in case of different fin spacing at $S_t = 84$ mm and $S_i = 24.2$ mm.

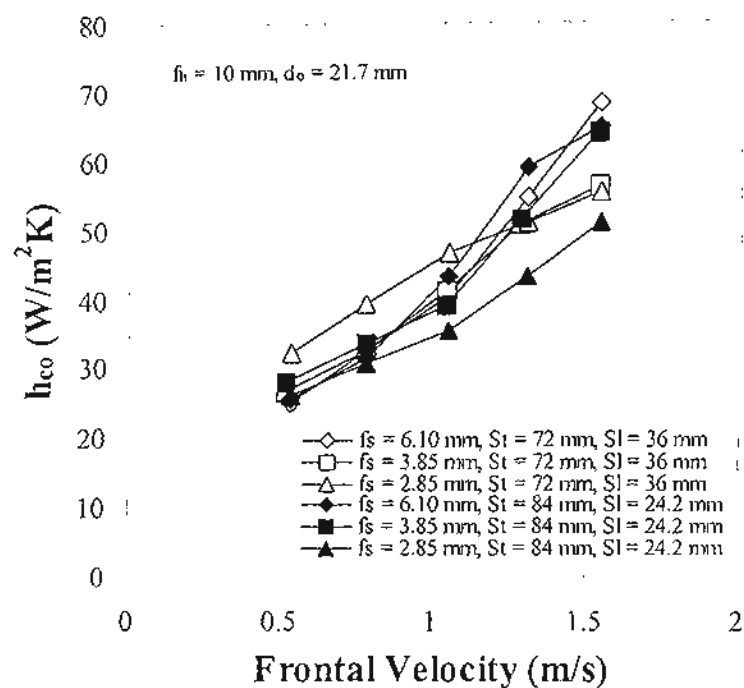


Figure 4.10 Comparisons of the heat transfer coefficient in wet condition in case of different tube arrangement.

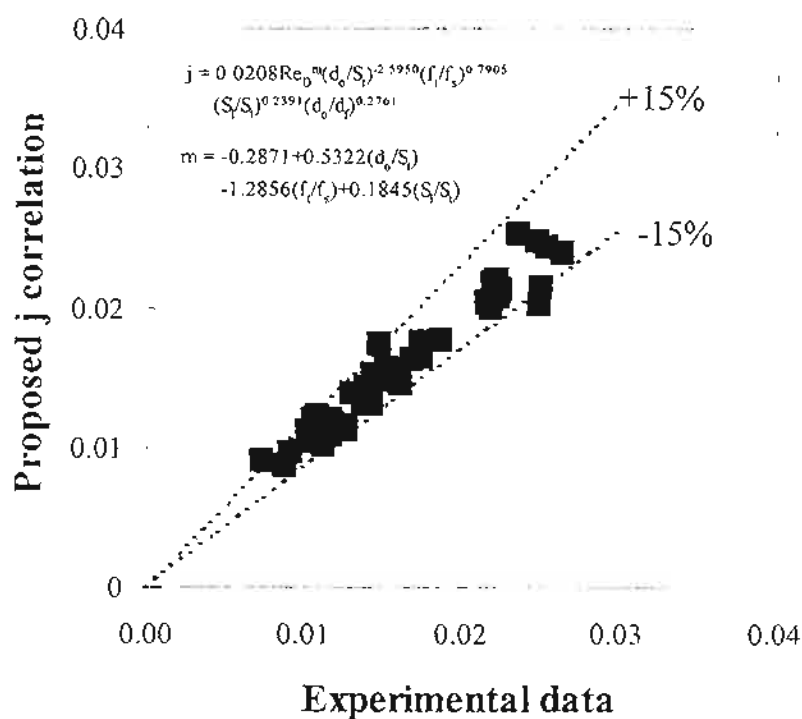


Figure 4.11 Comparison of the heat transfer data with the proposed correlation.

4.3.2 Inline Arrangement

4.3.2.1 Pressure Drop

Figures 4.12-4.15 show the air stream pressure drops in the cross flow heat exchanger. It is found that the air stream pressure drop increases with the frontal velocity of air. Moreover, when compare with the result obtained from previous chapter in the case of dry surface heat exchanger, the pressure drop is slightly higher than that of dry surface. This is because only small amount of water vapor is condensed on the heat exchanger surface.

Figures 4.12-4.14 also show the pressure drop increases with tube diameter (d_o) and fin height (f_h). However, it decreases with the increasing of fin spacing (f_s). These phenomena come from the increasing of surface area resulted in higher the airflow resistance. The effect of tube arrangement is also found in Figure 4.15. Higher transverse pitch of tube bank (S_t) gives lower pressure drop.

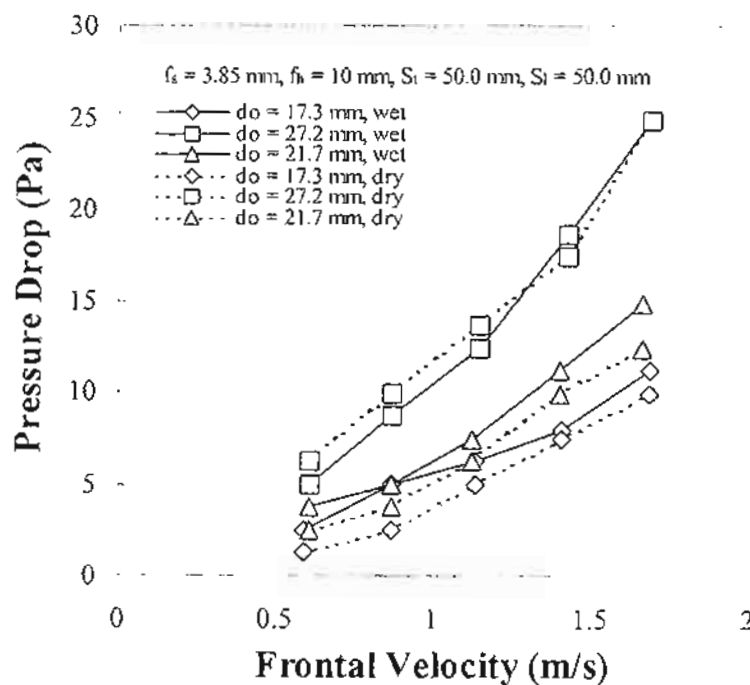


Figure 4.12 Effect of tube diameter on the pressure drop.

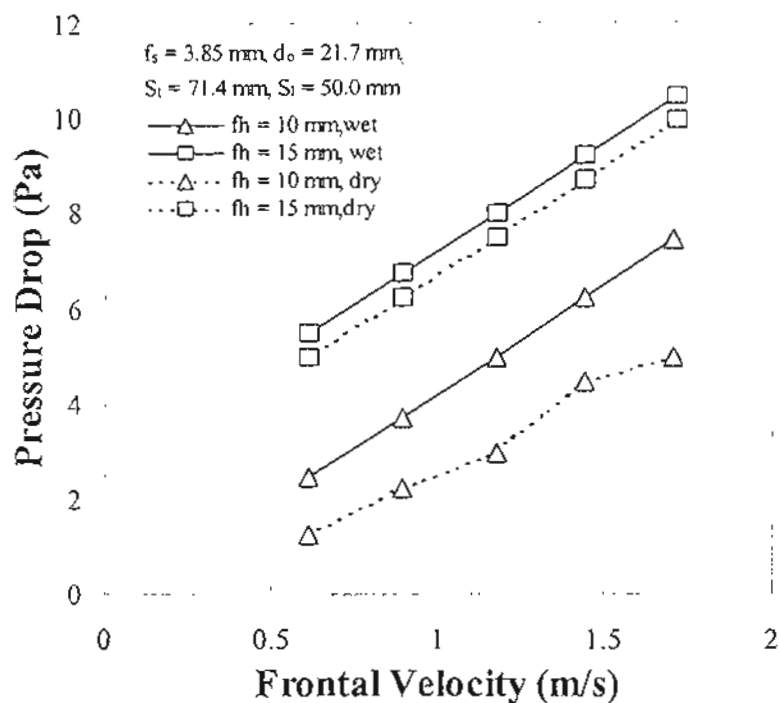


Figure 4.13 Effect of fin height on the pressure drop.

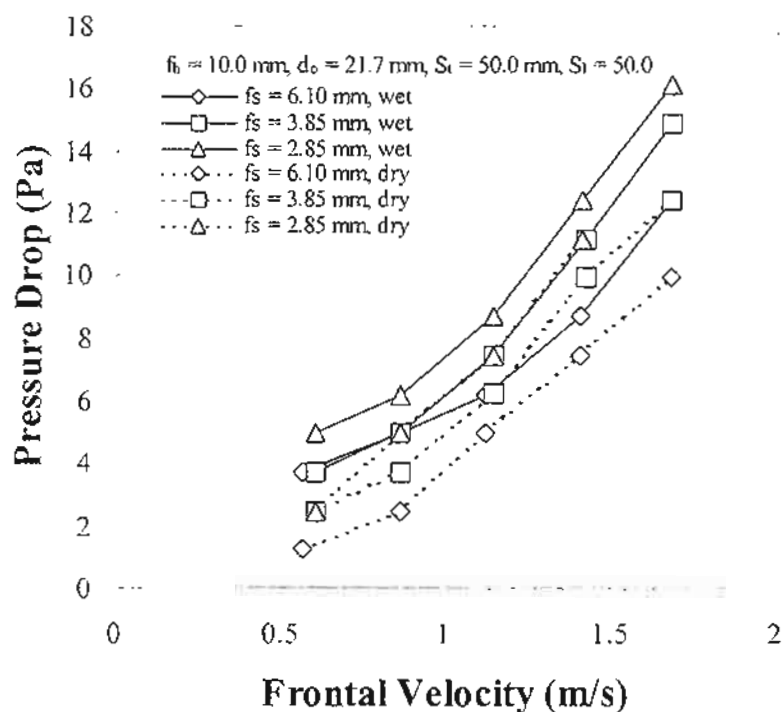


Figure 4.14 Effect of fin spacing on the pressure drop.

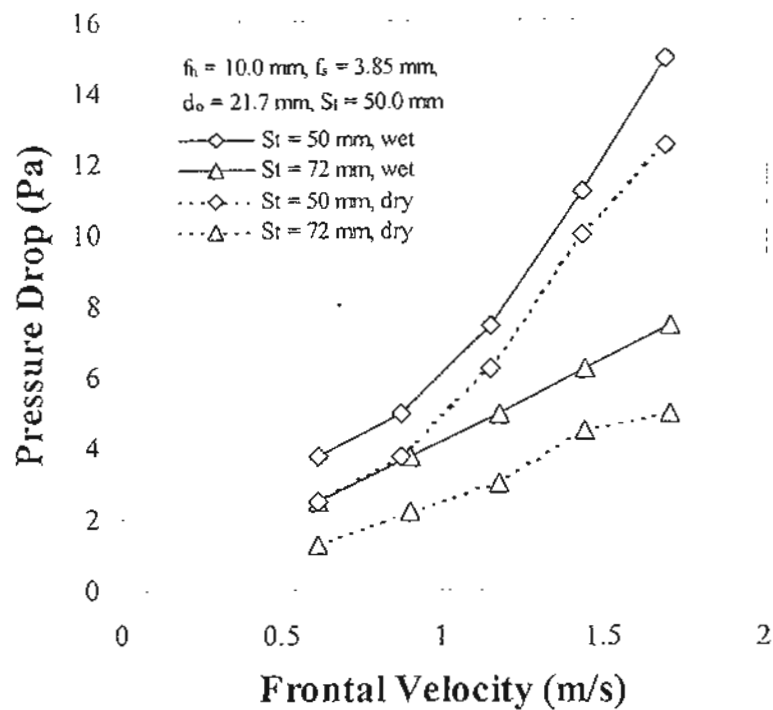


Figure 4.15 Effect of transverse pitch on the pressure drop.

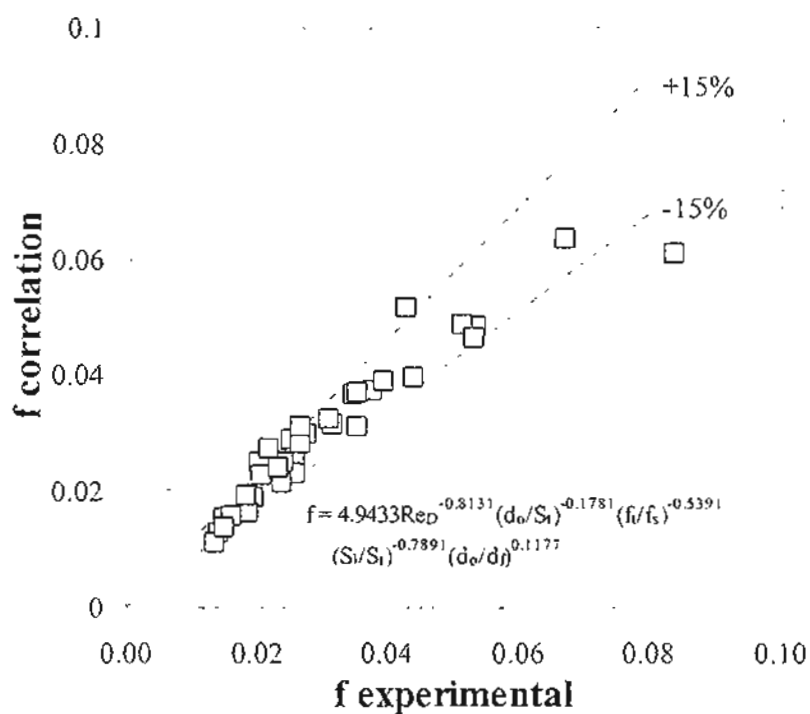


Figure 4.16 The comparison of f factor from experiment and correlation.

In this research, the correlation for predicting the air stream pressure drop including the effect of various quantities is also developed and the model is in a form of

$$f = 4.9433 \text{Re}_D^{-0.8131} \left(\frac{d_o}{S_t} \right)^{-0.1781} \left(\frac{f_t}{f_s} \right)^{-0.5391} \left(\frac{S_t}{S_l} \right)^{-0.7891} \left(\frac{d_o}{d_f} \right)^{0.1177} \quad (4.23)$$

From Figure 4.16, it is found that the f model can predict about 82.3% of the experimental data within $\pm 15\%$ accuracy.

4.3.2.2 Sensible heat transfer coefficient

Figure 4.17 shows the effect of tube diameter on the sensible heat transfer coefficient at various frontal velocities of air stream. The fin spacing (3.85 mm), fin thickness (0.4 mm), and the fin height (10 mm) are taken for this comparison. The transverse and the longitudinal tube pitches are 50 mm. As expected, the heat transfer coefficient rises with the frontal velocity. However, it is interesting to note that the heat transfer coefficient increases with the reduction of tube diameter. This phenomenon is attributed to the ineffective area behind the tube which increases with the tube diameter especially, the inline arrangement. Wang et al. [15] performed flow visualizations via dye injection technique for fin-and-tube heat exchangers having inline arrangement. Their visual results unveil a very large flow circulation behind the tube row. Consequently this large recirculation does not only contribute to the decrease of heat transfer coefficient but also to the rise of pressure drop. In addition, the large recirculation may also block the subsequent tube row and degrades the heat transfer performance hereafter.

Figure 4.19 shows the effect of fin height on the airside performance for inline arrangement. In this comparison, the associated fin heights are 10 and 15 mm and the fin spacing and the tube diameter are 3.85 mm and 21.7 mm with the transverse and the longitudinal pitches are 71.4 and 50 mm, respectively. As seen in the figure, the influence of fin height shows tremendous influence on the heat transfer performance. The heat transfer coefficients drop drastically with the increase of fin height. This is probably due to the airflow bypass effect. Actually the airflow is prone to flowing the portion where the flow resistance is small. In case of $f_h = 15$ mm, the airflow resistance around fin tube is larger than that for $f_h = 10$ mm. Therefore, part of the directed airflow just bypasses the tube row without effective contribution to the heat transfer and lower heat transfer coefficient is obtained.

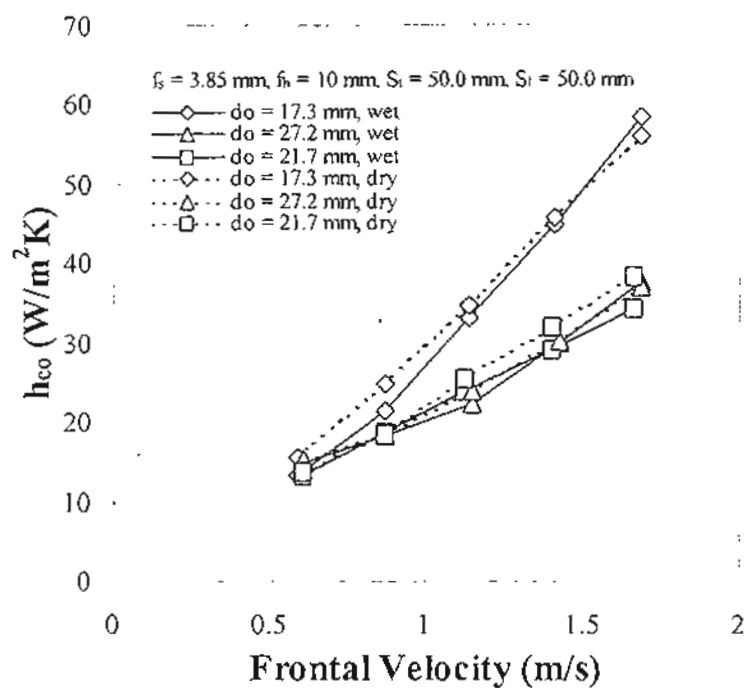


Figure 4.17 Effect of tube diameter on the sensible heat transfer coefficient.

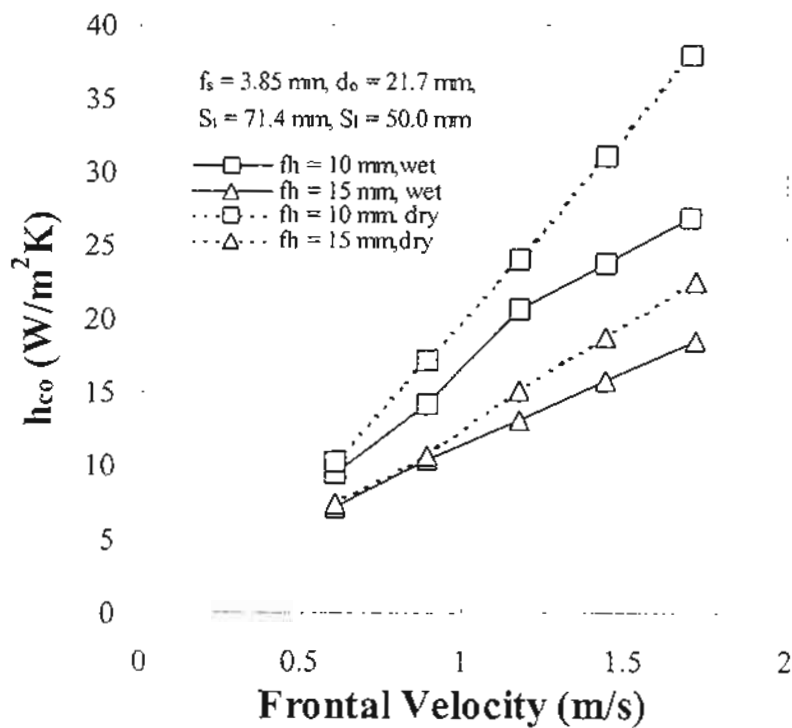


Figure 4.18 Effect of fin height on the sensible heat transfer coefficient.

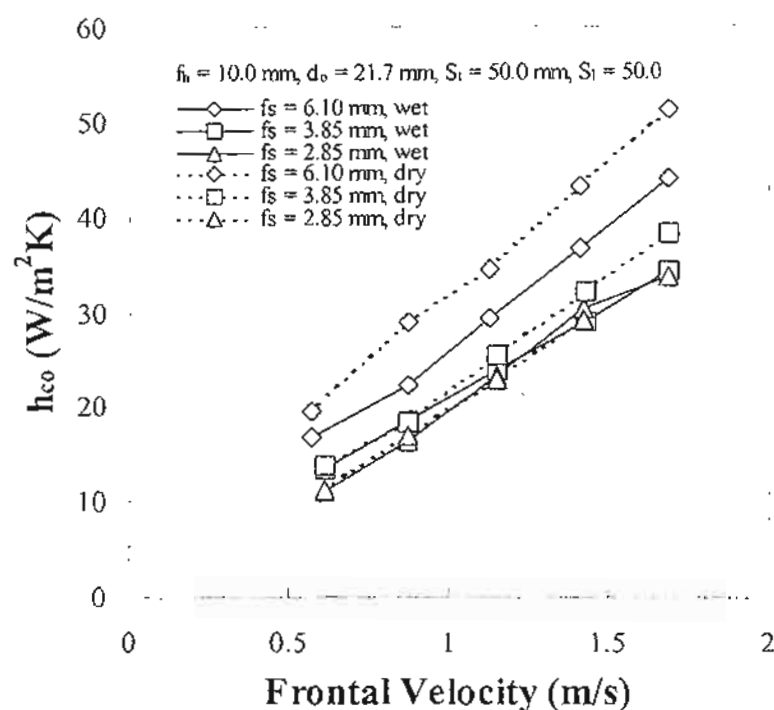


Figure 4.19 Effect of fin spacing on the sensible heat transfer coefficient.

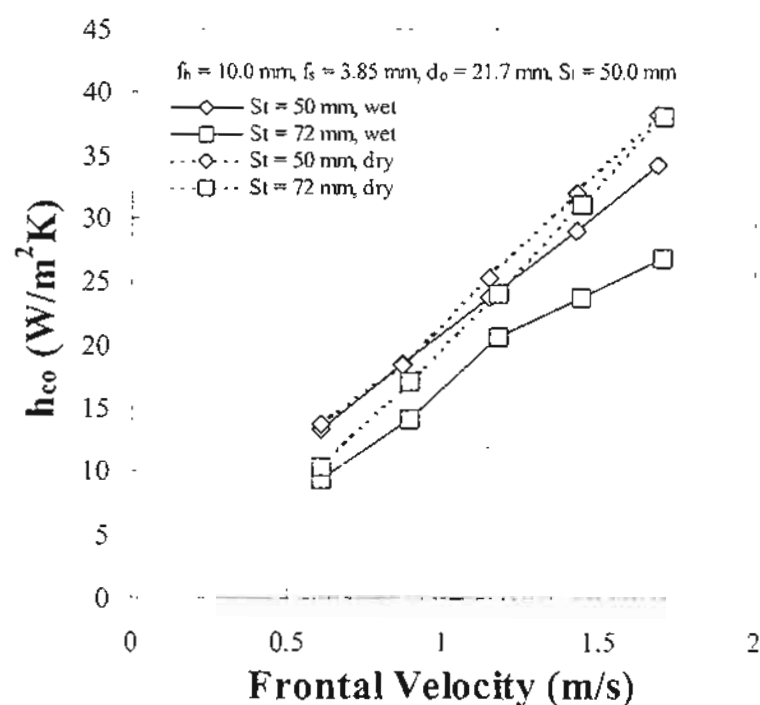


Figure 4.20 Effect of transverse pitch on the sensible heat transfer coefficient.

The effect of the fin spacing on the airside performance is shown in Figure 4.19. It was found that the increase of fin spacing gives a rise to the heat transfer coefficient. An explanation of this phenomenon is the same as that in the previous case which concludes that the result comes from the airflow bypass effect. The result of airflow bypass effect is also shown in Figure 4.20. It can be seen that the high transverse tube pitch ($S_t = 71.4$ mm) gives lower heat transfer coefficient than that of the low value ($S_t = 50$ mm).

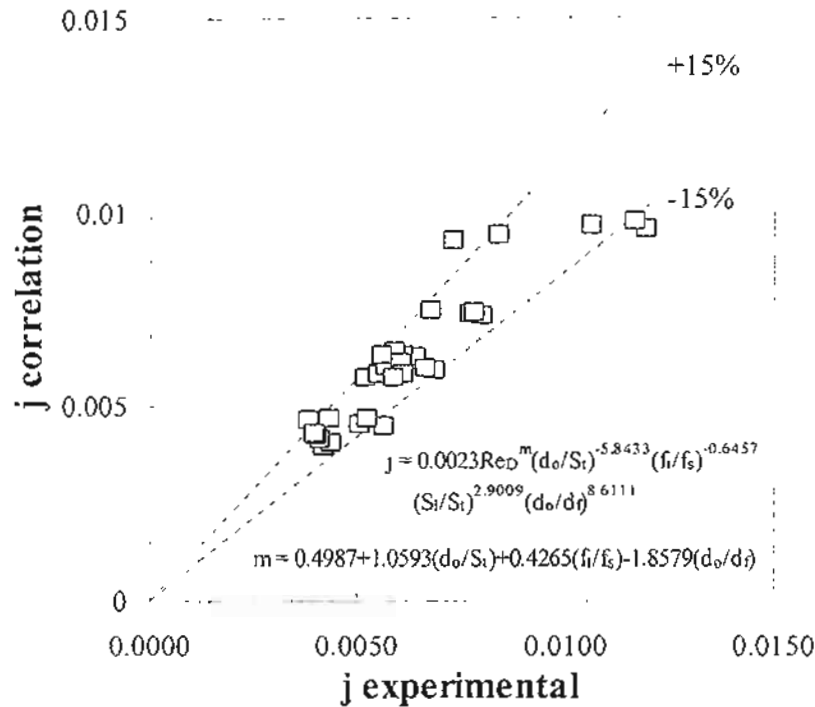


Figure 4.21 The comparison of j factor from experiment and correlation.

Comparison of the heat transfer coefficient under dehumidifying process with that of non-dehumidifying condition is also shown in Figures 4.17-4.20. The heat transfer phenomena of the wet surface heat exchanger are close to those of the dry surface. However, the heat transfer coefficient of the wet surface is lower than that of dry surface. The explanation is the same as the staggered arrangement that the present results are generally in agreement with the trend of Wang et al. [20].

In this research, the correlation for predicting the Colburn j factor including the effect of various quantities is also developed and the model is

$$j = 0.0023 \text{Re}_D^m \left(\frac{d_o}{S_t} \right)^{-5.8433} \left(\frac{f_t}{f_s} \right)^{-0.6457} \left(\frac{S_t}{S_l} \right)^{2.9009} \left(\frac{d_o}{d_f} \right)^{8.6111}, \quad (4.24)$$

where

$$m = 0.4987 + 1.0593 \left(\frac{d_o}{S_t} \right) + 0.4265 \left(\frac{f_t}{f_s} \right) - 1.8579 \left(\frac{d_o}{d_f} \right). \quad (4.25)$$

It is found that the j model can predict about 85.7% of the experimental data within $\pm 15\%$ accuracy. The comparison is also shown in Figure 4.21.

4.4 Conclusion

This study experimentally investigated the airside performance of a cross flow heat exchangers having crimped spiral configurations under the dehumidification. The effects of tube diameter, fin spacing, transverse tube pitch are examined. Based on the experimental observations, the following results are concluded:

8. The pressure drop of wet surface heat exchanger increases with the mass flow rate of air and the result is slightly higher or close to that of dry surface because only water condensate can be easily drained in the present comparatively large fin spacing and individual finned configuration.
9. The heat transfer coefficient of wet surface is slightly lower than that of dry surface.
10. The effect of tube diameter on the airside performance is significant. Larger tube diameter not only gives rise to lower heat transfer coefficient but also contribute significantly to the increase of pressure drops. This phenomenon is applicable in both dry and wet condition.
11. For wet surface, the influence of fin height is negligible whereas there is a small effect in dry surface.
12. The effect of fin spacing on the heat transfer performance is rather small. However, the increasing of fin spacing tends to have a lower heat transfer coefficient.
13. The tube arrangement plays an importance role on the heat transfer coefficient, the lower transverse pitch gives the higher heat transfer coefficient.
14. Airside performance in the present study is presented in terms of f and the j factor. The proposed correlations in case of staggered arrangement can predict 75% and 95% of experimental data within $\pm 15\%$. In case of inline arrangement, the proposed correlations can predict the j and the f factors can estimated about 85.7% and 82.3% of experimental data within $\pm 15\%$ accuracy.

CHAPTER 5

RESEARCH OUTPUT

The output of this research work can be classified into two groups. The first group is the advantage of this work that can help the designer who design the cross flow heat exchanger using crimped spiral finned tube producing from Thai industry.

The second output of this work is the publication. The researchers published the research result as follows:

The international journal

1. Nuntaphan A., Kiatsiriroat T., Wang C.C., Airside Performance at Low Reynolds Number of Cross Flow Heat Exchanger Using Crimped Spiral Fins, Int. Communications in Heat and Mass Transfer, Accepted.
2. Nuntaphan A., Kiatsiriroat T., Wang C.C., Heat Transfer and Friction Characteristics of Crimped Spiral Finned Heat Exchangers With Dehumidification, Applied Thermal Engineering, Submitted.

The national journal

1. Nuntaphan A., Kiatsiriroat T., Thermal Behavior of Crimped Spiral Fin Tube Bank Under Dehumidifying Process: A Case Study of Inline Arrangement, Songklanakarin J. of Sci. and Tech., Accepted

Conference

1. Nuntaphan A., Kiatsiriroat T., Heat Transfer Characteristic of Cross Flow Heat Exchanger Using Crimped Spiral Fins a Case Study of Staggered Arrangement, 17th Conference on Mechanical Engineering Network of Thailand, Prachinburi, 2003
2. Nuntaphan A., Kiatsiriroat T., Air-side Heat Transfer Coefficient of Thermosyphon Heat Pipe with Crimped Spiral Fins: a Case Study of Staggered Arrangement, 2nd Conference on Heat and Mass Transfer in Thermal Equipment, Chiang Mai, 2003

The detail of each paper is shown in the appendix.

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APPENDIX

PUBLICATIONS

1. Nuntaphan A., Kiatsiriroat T., Wang C.C., Airside Performance at Low Reynolds Number of Cross Flow Heat Exchanger Using Crimped Spiral Fins, Int. Communications in Heat and Mass Transfer, Accepted.
2. Nuntaphan A., Kiatsiriroat T., Wang C.C., Heat Transfer and Friction Characteristics of Crimped Spiral Finned Heat Exchangers With Dehumidification, Applied Thermal Engineering, Submitted.
3. Nuntaphan A., Kiatsiriroat T., Thermal Behavior of Crimped Spiral Fin Tube Bank Under Dehumidifying Process: A Case Study of Inline Arrangement, Songklanakarin J. of Sci. and Tech., Accepted
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January 15, 2004

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Subj: Ms. #CJ03/2732 — "Air-Side Performance at Low Reynolds Number of Cross Flow Heat Exchanger Using Crimped Spiral Fins" by A. Nuntaphan, T. Kiatsiriroat, and C.C. Wang

Dear Dr. Nuntaphan:

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AIR-SIDE PERFORMANCE AT LOW REYNOLDS NUMBER OF CROSS FLOW HEAT EXCHANGER USING CRIMPED SPIRAL FINS

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ABSTRACT

A total of 23 cross flow heat exchangers having crimped spiral configurations is studied. The effect of tube diameter, fin spacing, transverse tube pitch, and tube arrangements are examined. For the inline arrangement, the pressure drop increases with the rise of tube diameter but the associated heat transfer coefficient decreases with it. The increase of fin height also gives rise to considerable increase of pressure drop and decrease of heat transfer coefficients for the inline arrangement. However, for the staggered arrangement, the effect of the fin height on the pressure drop is much smaller than that of the inline arrangement due to the major contribution to the total pressure drops from the blockage of the airflow from staggered arrangement. Effect of the fin spacing on the airside performance is strongly related to the transverse tube pitch for both inline and staggered arrangements. Correlations of the present crimped spiral fins in both staggered and inline arrangements are developed. The proposed correlations give fairly good predictive ability against the present test data.

Introduction

There are many types of heat exchanger applicable to the recovery of waste heat such as shell-and-tube, plate, and cross-flow heat exchangers. Among them, the cross flow type is especially suitable for recovering heat from high the temperature air. Normally, the liquid always flows inside the tube whereas the gas flows across the tube bank. Because the dominant resistance occurs on the airside, it is a common practice to employ fins (such as

circular fins) on the tube bundle to increase the gas side performance. For circular fin geometries, there are many empirical correlations available in the literature such as the correlations of Briggs and Young [1], Robinson and Briggs [2], Rabas et al. [3] and ESDU [4] for evaluating the air side heat transfer coefficient and pressure drop of the staggered tube bank.

In case of inline arrangement, despite of its comparatively low heat transfer performance, its lower pressure drop and high reliability (easy to maintain and clean) are very attractive in very severe environment. For the heat transfer performance, Schmidt [5] recommended a correlation for predicting the heat transfer coefficient of the inline arrangement.

However, circular fin implemented in typically industrial application is usually in the form of crimped spiral fin as shown in Fig. 1. Unfortunately, there is no airside data reported in the literature. In this regard, it is the objective of this study to present relevant airside data of the crimped spiral fins. Especially in case of low Reynolds number of air stream which is the condition of the cross flow heat exchanger recovering heat from the flue gas of small package boiler which is many in Thailand. The relevant important geometric parameters such as tube diameter, fin spacing, fin height, and tube spacing influencing the airside performance are also investigated. Moreover the empirical correlations capable of evaluating the airside performance are also developed.

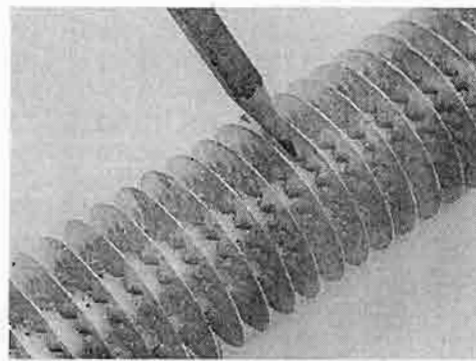


FIG. 1

Schematic of the crimped spiral fin geometry

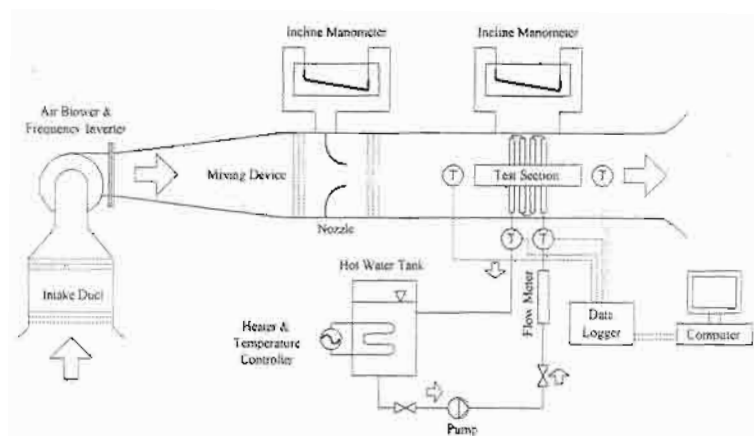


FIG. 2

Schematic diagram of the experimental setup