

**Experimental Study on the Heat Transfer Characteristics of a Spiral Fin-Tube Heat Exchanger
with Staggered Tube Alignment**

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ABSTRACT

A spiral fin-tube heat exchanger is expected to have better heat transfer performance than a flat plate fin-tube heat exchanger [1]. The objective of this study is to investigate the heat transfer characteristics of the spiral fin-tube heat exchanger with staggered tube alignment varying fin spacing and the number of tube rows under non-frosting conditions. The heat transfer performance of the fin-tube heat exchanger was inversely proportional to fin spacing [2]. The heat transfer performance is inversely proportional to the number of tube rows but the effect of the number of tube rows become negligible as the number increased. The developed heat transfer correlation shows good agreement with measured data with a mean deviation of 4.27%.

NOMENCLATURE

A	: surface area (m ²)
d	: tube diameter (mm)
D _f	: dimensionless fin spacing
F _s	: fin spacing (mm)
h	: heat transfer coefficient (w/m ² · k)
h _f	: fin height (mm)
k	: thermal conductivity (w/m · k)
L	: characteristic length (mm)
N	: the number of tube rows
Nu	: Nusselt number
r	: radius (mm)
Re	: Reynolds number
U	: overall heat transfer coefficient (w/m ² · k)
η _o	: surface effectiveness

Subscripts

a : air

do	: outer tube diameter
dh	: hydraulic diameter
i	: inner
loss	: loss
md	: modified
o	: outer
r	: refrigerant

INTRODUCTION

Heat exchanger is the one of the most important component in the air conditioners and refrigerators. In a fin-tube heat exchanger for refrigeration and air conditioning system, nearly over 70% of total heat transfer rate is determined by the fin, so the fin could be regarded as a very important factor. To make more compact and better heat exchangers, studies on various heat exchangers were performed experimentally and numerically.

Wang et al. [3] studied the heat transfer and friction characteristics of plain fin-and-tube heat exchangers using experiment. They studied heat exchangers having small tube diameters (7.3, 8.28, 10.0 mm) and small fin spacing from 1.21 to 2.31 mm with staggered tube alignment. They proved that the effect of fin spacing is independent on j factor for $n > 4$ $Re > 2000$ and the effect of the number of tube rows on the heat transfer performance became very small at very high Reynolds number.

Jacobi and Sommers [4] studied vortex generator enhancement for multi-rows plain-fin-tube heat exchangers with fin spacing of 3.6 mm. They tested experimentally heat exchangers with one winglet vortex generators, three winglet vortex generators and a general type. They proved the heat transfer performance of the heat exchanger attaching winglet

vortex generators was superior to the baseline heat exchanger over the entire range of Reynolds number from 200 to 900.

Yun et al. [5] studied the heat transfer characteristics of a spiral fin-tube heat exchanger varying fin spacing from 5 to 9 mm, fin height with 6, 8, 9 mm, the number of tube rows from 1 to 6, and tube alignments for inline and staggered. They proved that the heat transfer coefficient is dependent on fin spacing but independent on fin height. The Nusselt number of the staggered tube alignment has 10% higher than that of the inline tube alignment. They also proposed a correlation to predict heat transfer coefficient yielding the deviation of 4%.

Kim et al. [2] studied flat plate fin-tube heat exchangers having large fin spacing (7.5~15.0 mm) with the number of tube rows from 1 to 4. They showed that the heat transfer coefficient increased with the increase of the fin spacing because boundary layer interruption between the fins was delayed. The heat transfer coefficient was inversely proportional to the number of tube rows. They proposed a new correlation for the heat transfer performance, which could be applied to heat exchanger with large fin spacing.

Lee et al. [1] experimentally investigated the heat transfer performance of circular fin-tube heat exchangers varying fin spacing (5~12.5 mm), the number of tube rows (1~5 rows) with inline tube alignment. The heat transfer performance of the circular fin-tube heat exchanger showed better than that of flat plate fin-tube heat exchangers. From their results, the study of the circular fin-tube heat exchanger was regarded as having better heat transfer performance than flat plate fin-tube heat exchangers. Therefore, the objective of this study is to investigate the air-side heat transfer characteristics of the circular fin-tube heat exchanger with large fin spacing and staggered tube alignment. In addition, the heat transfer performance of circular fin-tube heat exchangers was studied varying fin spacing and the number of tube rows.

EXPERIMENTAL SETUP AND TEST PROCEDURE

Figure 1 shows the schematic of the experimental set-up. The test equipment consisted of an air circulation section and a refrigerant circulation section. Air circulation section included a centrifugal fan, a nozzle, thermocouple bundles and dew point sensor. The centrifugal fan controlled the air flow rate using an inverter controller. The air flow rate was calculated with the differential pressure gauge. The inlet and outlet temperatures of the test section were measured by averaging the values in thermocouple grids. The test section was set in a 760 mm × 320 mm × 100 mm

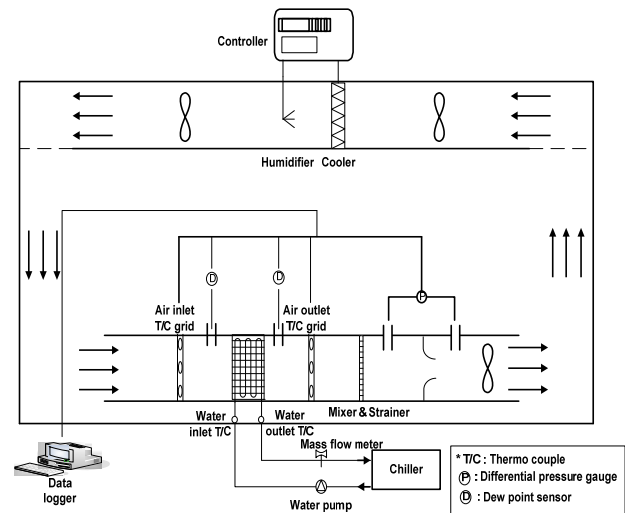


Figure1 The schematics of the experimental set-up

transparent acryl box. An insulation form was attached around the test section to reduce the heat loss. The refrigerant section was composed of a chiller which controlled the water temperature and, mass flow rate, inlet and outlet thermocouples, and ethylene glycol-water mixture (60% : 40%) used as the refrigerant. The thermocouples were attached to inlet and outlet pipes of the heat exchanger. The mass flow rate was maintained at 155 kg/h by using a magnetic gear pump. The temperature of the refrigerant was kept at 33°C during the test. The experimental set-up was put in the climate chamber, which was able to control the air temperature and humidity. All measured data were collected in a data logger with every ten second. Detail of the test conditions is shown in Table 1.

Figure 2 shows the configurations of the tested fin-tube heat exchangers. Twenty heat exchangers were tested by varying fin spacing from 5 to 12.5 mm and the number of tube rows from 1 to 5. The spiral fin height (diameter) was 24.45 mm and fin thickness was 0.2 mm. Tube outer diameter was 8 mm and thickness was 0.5 mm. Table 2 shows the specification of the spiral heat exchanger.

The heat transfer rate of the air-side was calculated by considering the heat loss between the test section and surroundings. Heat loss was calculated by comparing electric power input with the air-side heat transfer rate. The overall heat transfer coefficient calculated using the LMTD method. And the heat transfer coefficient of the refrigerant side was calculated using the correlation suggested by Dittus and Boelter [6]. Air-side heat transfer coefficient was calculated from equation (1). The Nusselt number was calculated by equation (2).

Table1 Specifications of the Spiral Fin-Tube Heat Exchanger

Design Condition	Component		Spec.
	Tube-Side	Column	2
		Rows	1/2/3/4/5
		Material	Al
		Transverse/Longitudinal Spacing (mm)	30.5/35.0
		d _i /d _o (mm)	7/8
	Fin-Side	Length (mm)	260
		Arrangement	Staggered
		Spacing (mm)	5/7.5/10/12.5
		Type	Spiral
		Thickness (mm)	0.25
		Height (mm)	24.45

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi Lk} + \frac{1}{h_o A_o \eta_o} \quad (1)$$

$$Nu = \frac{h_o d_o}{k} \quad (2)$$

RESULTS AND DISCUSSION

The effect of the fin spacing

Figure 3 shows the effect of the fin spacing on the Nusselt number with the variation of the Reynolds number for N=1. The Nusselt number increased linearly with the fin spacing because the velocity boundary and thermal boundary layers interruption was occurred quicker in small fin spacing, resulting in the decrease of the heat transfer performance. These results showed good agreement with Yun et al. [5]. The Nusselt number over fin spacing of 10.0 mm showed similar because of no boundary layer interruption. These trends showed in studies of Kim et al. [2]. They mentioned that the heat transfer performance of the fin-tube heat exchanger showed similar, as the boundary layer between the fins was not interrupted. Unfortunately, any visualization studies of these effects could not be found in literature. Figure 4 shows the effect of the fin spacing on the heat transfer rate with the variation of the air flow rate

Table 2 Experimental Conditions

	Parameter	Condition
	Temperature (°C)	3 ± 0.5
Air-Side	Relative Humidity (%)	90
	Air Flow Rate (m ³ /min)	0.8/1.1/1.3/1.8
	Temperature (°C)	33 ± 0.5
Refrigerant-Side	Mass Flow Rate (kg/h)	155
	Mixing Ratio (%)	60:40

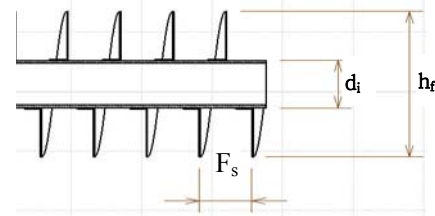
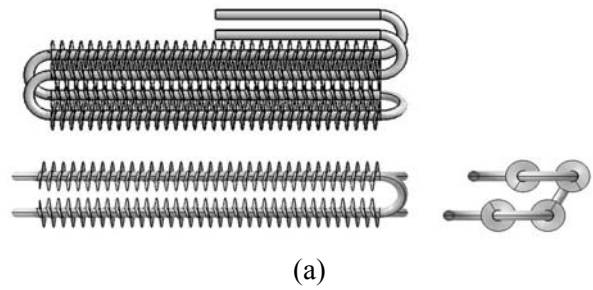


Figure 2 Configurations of the spiral fin-tube heat exchanger: (a) Schematic and (b) Cross-section

for N=3. The heat transfer rate increased with the decrease of the fin spacing because of the increase of the heat transfer area as shown in Figure 5. Figure 5 shows the heat transfer area with the variation of the number of tube rows at fin spacing from 5 to 12.5 mm.

The effect of the number of tube rows

Figure 6 shows the Nusselt number with the variation of the number of tube rows. The Nusselt number decreased with the increase of the number of tube rows at the same fin spacing (10 mm). This result showed a good agreement with Yun et al. [5] and Lee et al. [1]. They mentioned that the heat transfer performance decreased with the increase of the number of tube rows in the staggered alignment.

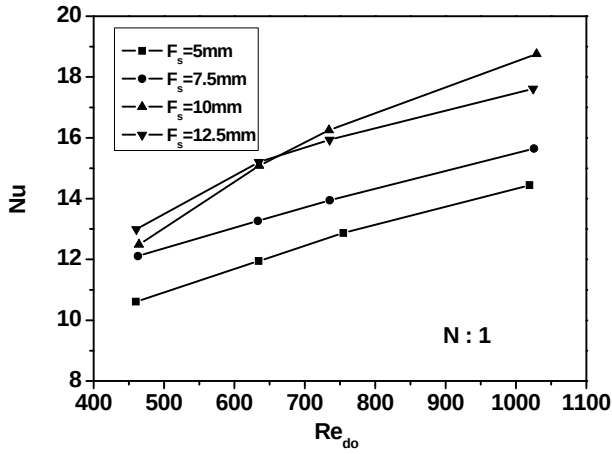


Figure 3 Effect of the fin spacing on the Nusselt number with the variation of the Reynolds number

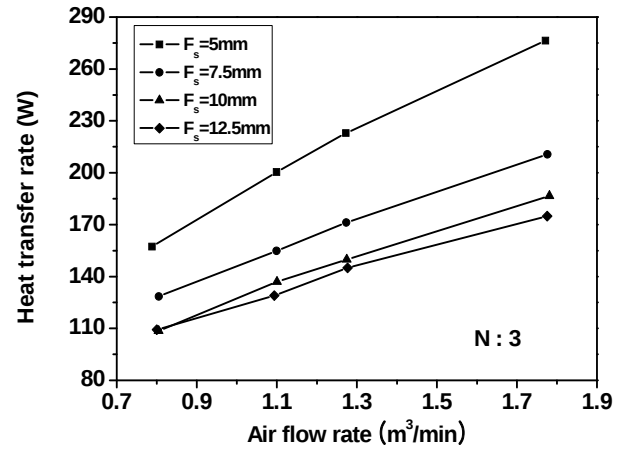


Figure 4 Effect of the fin spacing on the heat transfer rate with the variation of the air flow rate for N=3

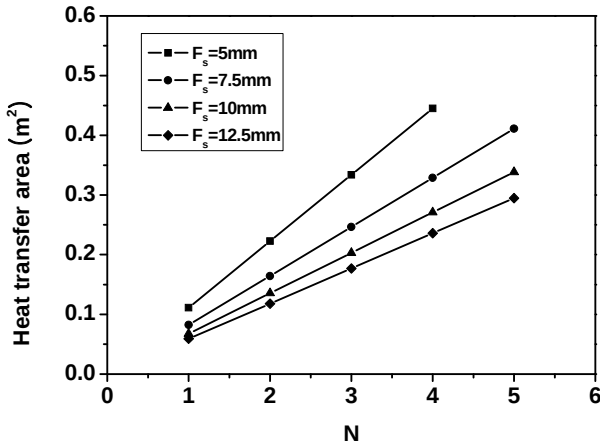


Figure 5 Effect of the fin spacing on the heat transfer area with variation of the number of tube rows

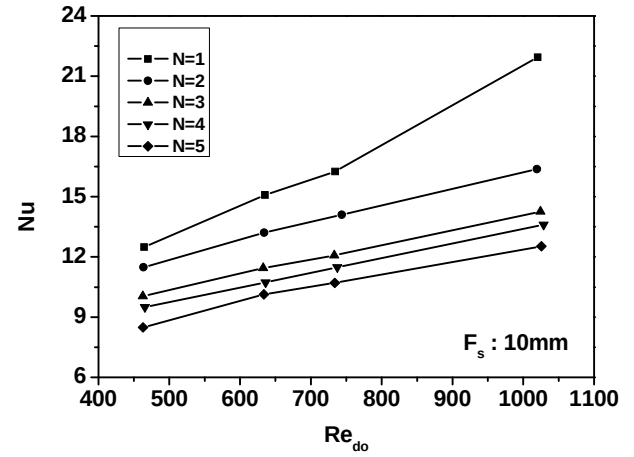


Figure 6 Effect of the number of tube rows on the Nusselt number with the variation of the Reynolds number at fin spacing of 10 mm.

The difference of the heat transfer rate from 1 to 2 rows, 2 to 3 rows, and 3 to 4 rows were 11.4%, 12.0%, and 5.9%, respectively. The effect of the number of tube rows on the heat transfer performance decreased because the wake region behind the first tube row in the staggered tube alignment was suppressed to deeper rows by blockage effect, resulting in the increase of the turbulence as mentioned in Mon and Gross [7]. Moreover, these effects were activated by horseshoe vortex in deeper rows as mentioned in Baker [8]. Figure 7 shows the effect of the number of tube rows on the heat transfer rate with the variation of the number of tube rows at fin spacing of 10mm. The heat transfer rate increased with the increase of the number of tube rows because of the increase of the heat transfer area as shown in Figure 5.

Correlation development and validation

Based on the experimental data, the Nu_{dh}

was correlated as a function of dimensionless fin spacing, Reynolds number of hydraulic diameter and the number of tube rows. A total of 80 data points were used for development of the correlation using multiple regression method suggested by Wang et al. [9]. The predicted Nu_{dh} was consistent with the measured with a mean deviation of 4.27%. The correlation was validated within $400 < Re_{dh} < 1300$, $1 \leq N \leq 5$, $5 \leq F_s \leq 12.5$ mm.

$$Nu_{dh} = 3.4131 \frac{Re_{dh}^{0.472} D_f^{0.492}}{N^{0.1808}} \quad (3)$$

Figure 8 compares the predicted Nu_{dh} with the result of Kim et al. [2] and Yun et al. [5]. The predicted correlation was matched with a mean deviation of 5.15% and an average deviation of -2.20%.

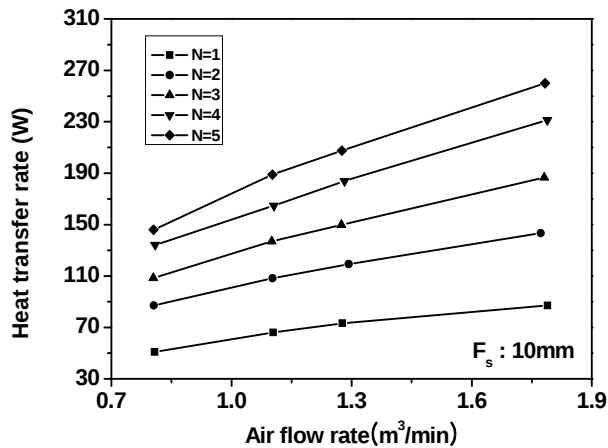


Figure 7 Effect of the number of tube rows on the heat transfer rate with the variation of the air flow rate at fin spacing of 10mm

The correlation proposed by Yun et al. [5] showed similar trends with the measured data with a mean deviation of 8.95% and an average deviation of -0.6% for staggered tube alignment and $N > 3$ although their test samples had small fin spacing. It is thought that the same type of the fin type and many rows of staggered tube alignment minimized the effect the fin spacing [3].

The correlation proposed by Kim et al. [2] under-predicted the measured data with a mean deviation of 42.07% and an average deviation of -42.07% for staggered tube alignment because the correlation was developed in different fin type (flat plate fin) and thicker fin thickness [10] despite of similar fin spacing.

CONCLUSIONS

The heat transfer performance was inversely proportional to fin spacing under fin spacing of 7.5 mm for $Re_{do} < 1000$ and $N = 1$ due to the boundary layer interruption. However, when fin spacing was over 10 mm, the heat transfer performance was independent on the fin spacing. In staggered alignment, the effect of the number of tube rows decreased with the increase of the number of tube rows due to the blockage effect. A new correlation for the Nu_{dh} in spiral fin-tube heat exchangers with large fin spacing and staggered tube alignment were developed.

ACKNOWLEDGEMENTS

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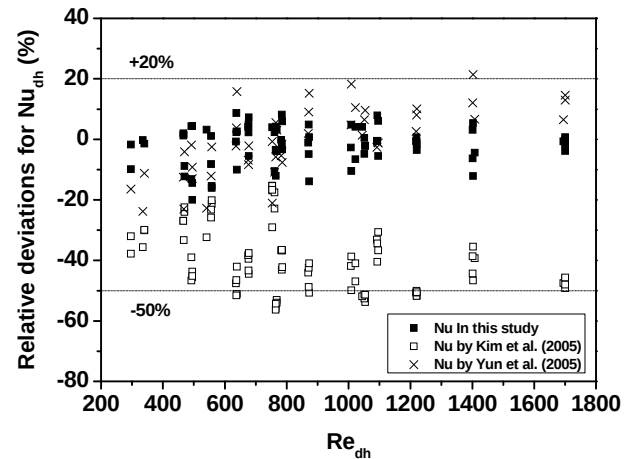


Figure 8 Comparisons of the predicted Nu with the other studies

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