2004

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EFFECTS OF FIN AND TUBE ALIGNMENT ON THE HEAT TRANSFER PERFORMANCE OF FINNED-TUBE HEAT EXCHANGERS WITH LARGE FIN PITCH

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ABSTRACT

The objective of this study is to provide experimental data that can be used in the optimal design of finned-tube heat exchangers with large fin pitch. In this study, six types of fined tubes were tested by varying airflow rate. Both the effects of number of tube row, fin pitch, and fin type and the influences of tube and fin alignment on the heat transfer performance are investigated. The air-side heat transfer coefficient is calculated from the measured data. Staggered fin and tube alignments improve heat transfer performance by 7% and 10%, respectively, as compared to the inlined fin alignment. By applying both the staggered fin and tube alignments, the heat transfer performance increases by approximately 20% as compared to the continuous flat plate finned-tube. The heat transfer performance decreases with a rise of tube number. For large fin pitch, the effects of fin pitch on the heat transfer performance are negligible.

1. INTRODUCTION

A heat exchanger is a device in which heat is transferred form a hot fluid to a cold fluid, and it is an essential unit in heat extraction and recovery system. With increasing emphasis on energy savings, extensive efforts are being made to enhance the heat transfer performance of a heat exchanger. These efforts are divided into two parts. One is focused on liquid or two-phase side heat transfer performance, and the other is focused on air-side heat transfer performance. However, the heat exchanger performance often can be limited by the air-side, because the air-side heat transfer coefficients are naturally very lower than the liquid or two-phase side. So, many active and passive methods are developed to improve the air-side energy performance and reduce heat exchanger volume and manufacturing costs.

Rich (1973, 1975) investigated the effects of fin pitch and number of tube row for staggered plate finned-tube heat exchanger. It was shown in his research that the heat transfer coefficient was essentially independent of fin pitch (1.23-8.7mm) at a given mass velocity. McQuiston (1978) developed heat and mass transfer for four rows staggered tube banks with flat-plate fins. Wang (2000) reviewed and summarized the most influential investigations of the pate finned-tube heat exchangers of plain fin geometry since 1971. Romero-Méndez et al. (2000) examined the influence of fin spacing of a single row finned-tube heat exchanger through flow visualization and numerical analysis. Halici et al. (2001) experimentally investigated the effects of number of tube row on heat, mass and momentum transfer of a finned-tube heat exchanger. They showed that the Colburn and friction factors decreased with an increase in the number of tube row. Recently, a heat transfer enhancement technique by using a winglet-type vortex generator was tested by Wang et al. (2002) and Torii et al. (2002).

As mentioned above, numerous studies on air-side heat transfer characteristics had been reported in the open literature, and most of them were focused on heat exchangers with small fin pitches from 3 to 1.5mm. In these studies, a heat exchanger surface has been changed from flat type to wavy, slit, and louver type to develop effective and compact heat exchangers in residential air conditioning systems, which have no frosting problem. However, flat plate finned-tube heat exchangers with large fin pitch are used in refrigerator and freezer due to frosting problems. Therefore, further study on the air-side heat transfer performance of finned-tube heat exchangers with large fin pitch are required.

The major objective of this study is to provide experimental data that can be used in the optimal design of an evaporator for a refrigerator and freezer. In this study, we tested an individual row of the heat exchanger coil to simplify analysis of the complicated coil. After we investigated the heat transfer performance of each row, we combined several pieces of heat exchanger by varying longitudinal fin space, fin and tube alignments. In this manner, the effects of longitudinal fin space and fin arrangement on the heat transfer performance were investigated.

Several types of finned-tube heat exchangers, such as a continuous flat plate finned-tube, a discrete flat plate finned-tube, and a spine finned-tube, are used as an evaporator coil of a refrigerator and freezer. Due to its durability and productivity, the continuous flat plate finned-tube is widely used. However, it has some structural defects compared to the discrete flat plate finned-tube. Therefore, the air-side heat transfer performance of the continuous flat plate finned-tube is compared with that of the discrete flat plate finned-tube.

2. EXPERIMENTAL SETUP AND TEST PROCEDURE

A schematic of the experimental setup for measurements of heat transfer characteristics is shown in Fig. 1. The test setup was installed in a psychrometric chamber to provide a pre-controlled ambient temperature. The psychrometric chamber was maintained at 2°C using an air-handling unit including a cooling coil, a heating coil, and a humidifier. For measurement convenience, the ethylene glycol-water mixture was utilized as a refrigerant inside the coil.

Generally, the air-side heat transfer coefficient is a function of only air mass velocity and shape of heat exchanger surface. Therefore, air inlet temperature, refrigerant inlet temperature, refrigerant mass flux, and humidity ratio were maintained constant in the experiments. Only airflow rate was varied from 0.8 CMM to 1.7 CMM. A constant temperature bath including a chiller and an electric heater controls the ethylene glycol-water mixture temperature. An open wind tunnel was placed inside of the psychrometric chamber. Air temperatures were measured by arithmetically averaging T-type thermocouple grid located before and after the test coil. The air temperature entering the test section was adjusted by using an electric resistance heater installed at the entrance of the flow chamber. The absolute humidity was measured using a chilled mirror dew point sensor. The humidity was maintained at a set point by adjusting power input to an ultrasonic humidifier using a PID controller.

The fin and tube surface temperatures were measured by using an infrared thermometer. The data was stored in a data logger at every three seconds. The refrigeration capacity of the test coil was determined by an air enthalpy method as well as a water-side capacity, which yielded a good agreement within 5%.

In this study, we tested an individual row of the heat exchanger coil to simplify analysis of a complicated coil. After investigating heat transfer characteristics of each row, two pieces of heat exchanger were combined. In this manner, the effects of tube space and fin arrangement on the heat transfer performance were investigated. As shown in Fig. 2, six types heat exchangers with a discrete flat plate fin (type 1, 3, 4), a continuous flat plate fin (type 5, 6), and a discrete wavy fin (type 2) were tested. Test conditions are specified in Table 1.

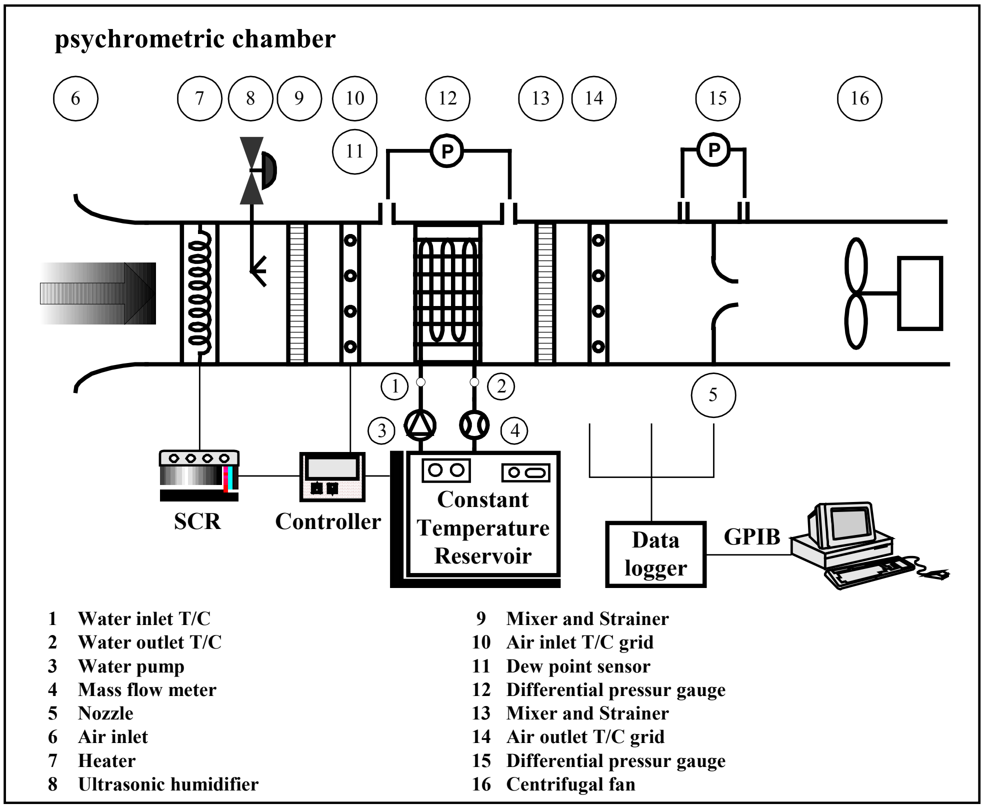

Figure 1. Schematic of the experimental setup

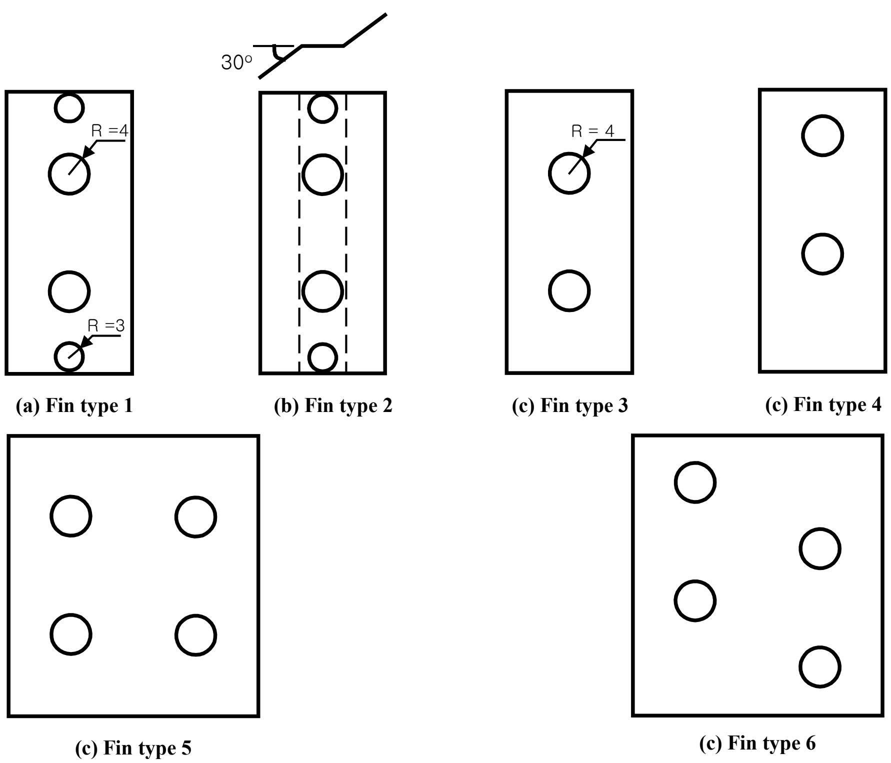

Figure 2. Fin configurations of the evaporator coil

Table 1. Test conditions

Parameter	Value
Inlet air temperature (°C)	3
Inlet air relative humidity (%)	60
Airflow rate (CMM)	0.8, 1.1, 1.4, 1.7
Refrigerant inlet temperature (°C)	33
Refrigerant mass flow rate (kg/hr)	150
Longitudinal tube space (mm)	27, 30, 33
Fin type	Discrete flat plate, Continuous flat plate, Wavy
Fin pitch (mm)	7.5, 10.0, 12.5, 15.0
Number of row	1, 2, 3, 4
Fin and tube alignment	In-lined, Staggered

3. RESULTS AND DISCUSSION

3.1 Heat Loss Test

The test section was made of transparent acrylic resin, which was not insulated from the ambient. So, there was a heat loss from the test section to the surroundings. The heat loss was measured by comparing power input to an electric resistance heater with an air-side heat capacity. The heat loss tests were conducted by varying power input and airflow rate. Conduction thermal resistance of the thick acrylic resin is very large compared to the convection thermal resistance of the inside and outside acrylic resin. Therefore, the heat loss can be a function of the temperature difference between the air temperature inside the test section and the surrounding temperature. Figure 3 shows the relationship between the temperature difference and the heat loss. By using this relationship, the air-side heating capacity and outlet air temperature were altered, which were used in determination of the air-side heat transfer coefficient.

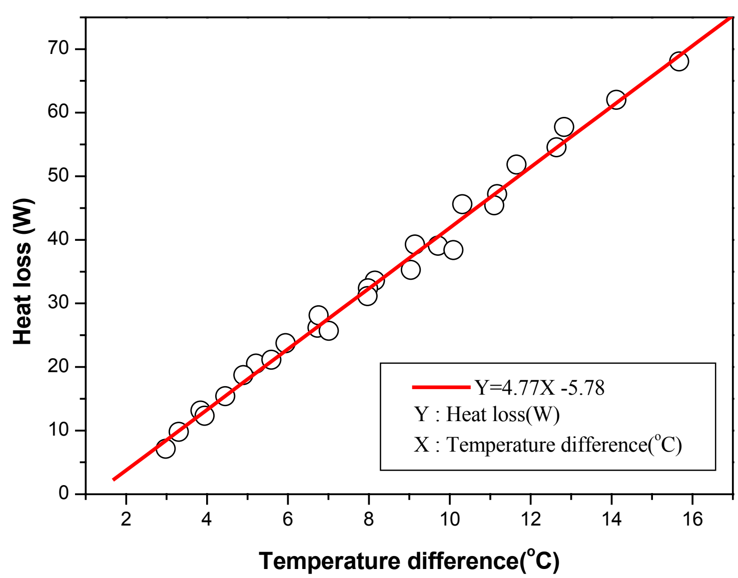

Figure 3. Relationship between temperature difference and heat loss

3.2 Effects of Number of Tube Rows on Heat Transfer Performance

Figure 4 shows the effects of number of tube row on the heat transfer performance of the continuous flat plate finned-tube with a fin pitch of 7.5 mm. Colburn *j*-factor decreases with an increase of number of tube row from 1 to 4 at a given fin pitch. Rich (1973) showed that the effects of number of tube row on the heat transfer performance was significant at low Reynolds numbers, while it become negligible at Reynolds number above 15,000. In this study, the decreasing rate of *j*-factor diminishes gradually with an increase of number of tube row at a fixed Reynolds number. This indicates that the flow fields become uniform at the rear portion of the heat exchanger with an increase of number of tube row. Even though the fin pitch increases from 7.5 to 15 mm (Figure 5), the effects of number of tube row on *j*-factor are consistent each other. This can be explained with the development of a boundary layer along with the distance from the inlet.

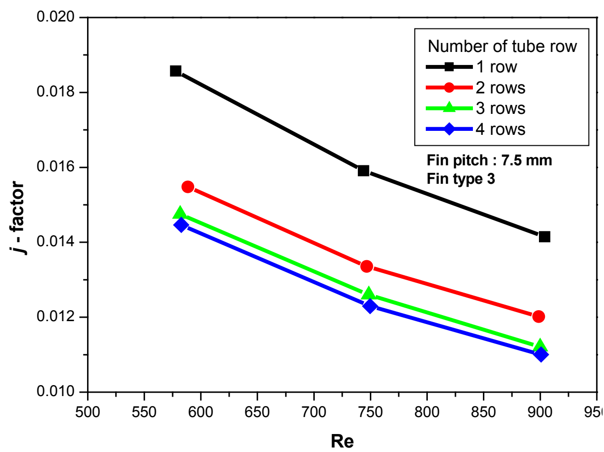

Figure 4. Variation of *j*-factor with number of tube row at a fin pitch of 7.5 mm

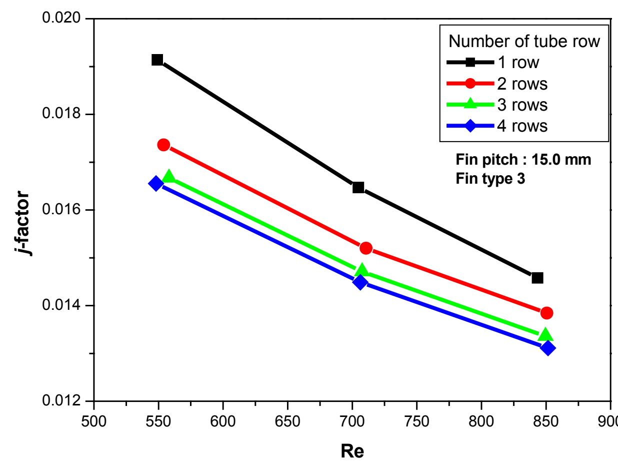

Figure 5. Variation of *j*-factor with number of tube row at a fin pitch of 15.0 mm

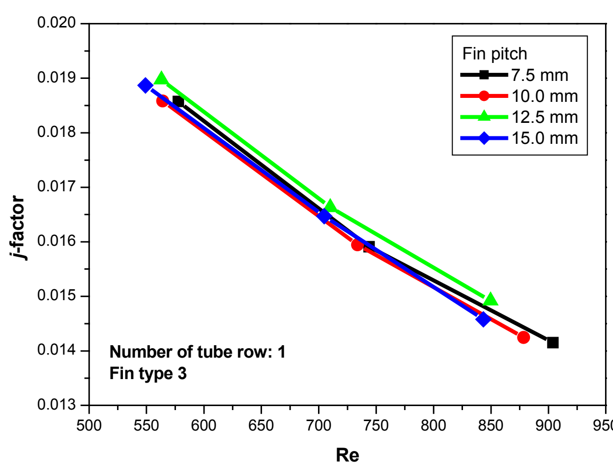

Figure 6. Effect of fin pitch on the heat transfer performance

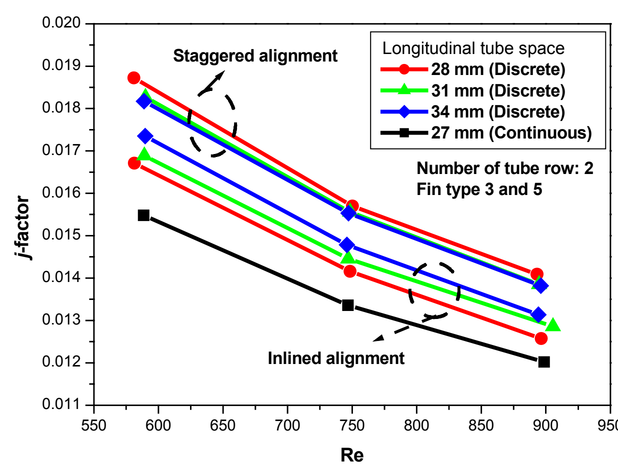

Figure 7. Effect of longitudinal tube space on the heat transfer performance

3.3 Effects of Fin Pitch and Longitudinal Tube Spacing on Heat Transfer Performance

Figure 6 shows the effects of fin pitch for fin type 3 on the heat transfer performance. Rich (1973) reported that *j*-factor for a flat plate finned-tube heat exchanger decreased with a decrease of fin pitch. However, the *j*-factor was independent of fin pitch when fin pitch was greater than 7.5 mm. This was identical to the data of Romero-Méndez et al. (2000). The boundary layer thickness for velocity and temperature is very important in the determination of heat and mass transfer coefficients over a flat plate. As the distance from the leading edge increases, the boundary layer thickness increases and the average heat transfer coefficient decreases. Based on the Blasius's similarity solution, the boundary layer thickness will be nearly 3.5 mm at the tailing edge of fin type 3. Because the boundary layer thickness is smaller than fin pitch at all cases, the heat transfer performance remains constant with a variation of fin pitch. Figure 7 represents the effects of longitudinal tube space. The heat transfer performance increases with a rise of longitudinal tube space in the inlined fin alignment, while this trend is reversed in the staggered fin alignment. The highest heat transfer region exists at the leading edge of the plate and frontal area of the tube. On the contrary, the heat transfer coefficient becomes smaller behind the tube, which is called the wake region. For the inlined fin alignment, the area of successive row that affected by the wake region decreases and the heat transfer performance increases with a rise of longitudinal tube space. For the staggered fin alignment, the thermally fully developed region occurs at the fin of successive row with a rise of longitudinal tube space. Therefore, the heat transfer performance decreases with an increase of longitudinal tube space in the staggered fin alignment.

3.4 Effects of Fin Type and Fin and Tube Alignment on Heat Transfer Performance

The design of an evaporator coil type in a refrigerator and freezer is dependent on defrosting methods. In this study, only fin type 1 and 2 have defrosting coil. The other fin types can be applicable when glass heaters or sheath heaters are used in the defrosting process. Figure 8 shows the effects of fin shape on the heat transfer performance. Because fin type 1 and 2 have a defrosting coil, maximum air velocity is greater than that of fin type 3. Therefore, Reynolds number becomes larger at the same airflow rate. From the results of fin type 1 and 2, we can see that a wavy shape fin improves the heat transfer performance by 13%. Although the defrosting coil increases heat transfer capacity by increasing heat transfer area and promoting turbulent intensity, these effects are not as large as those obtained by increasing air velocity. Figure 9 shows the effects of fin alignment on the heat transfer performance. By applying the staggered fin alignment, the heat transfer performance increases by 7% as compared with the inlined alignment. Moreover, the heat transfer performance improves by 15% as compared with the continuous flat plate fin. As shown in Figure 10, even though *j*-factor decreases with an increase of tube row from 2 to 4, the effects of staggered fin alignment are not reduced. Figure 11 shows the effects of tube alignment on heat transfer performance. Staggered tube alignment is more effective to improve the heat transfer performance of the continuous flat plate finned-tube. By making tube alignment staggered, the heat transfer performance increases by 10%. Besides, by making both fin and tube alignment staggered, the heat transfer performance increases by 20%.

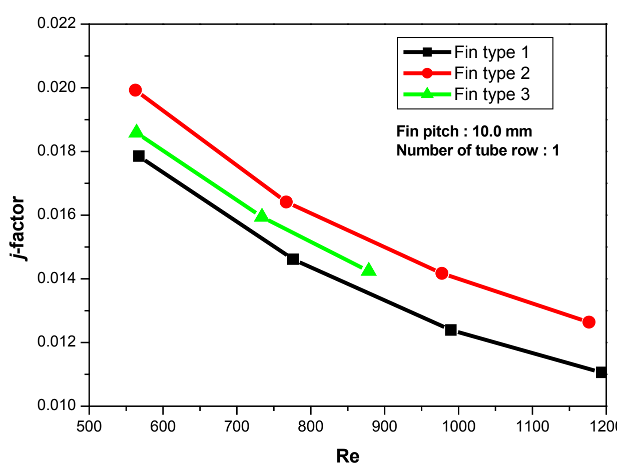

Figure 8. Effect of fin shape on the heat transfer performance

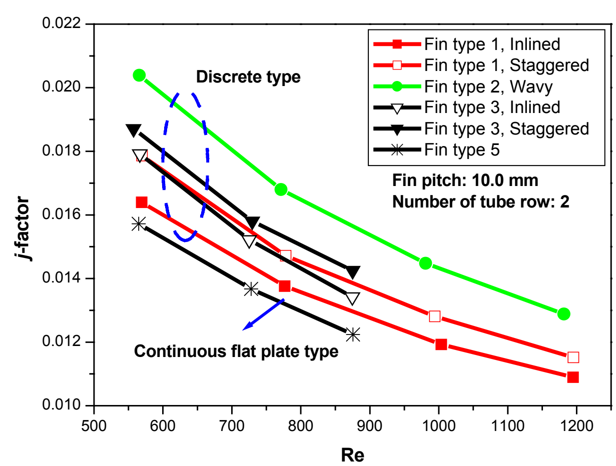

Figure 9. Effect of fin alignment on heat transfer performance

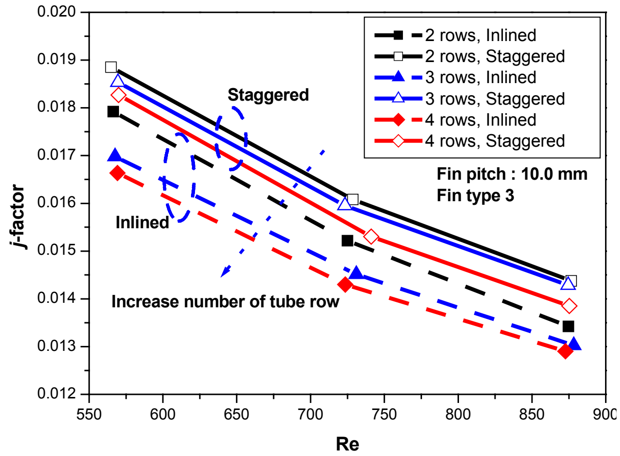

Figure 10. Effect of fin alignment on the heat transfer performance with an increase of tube row

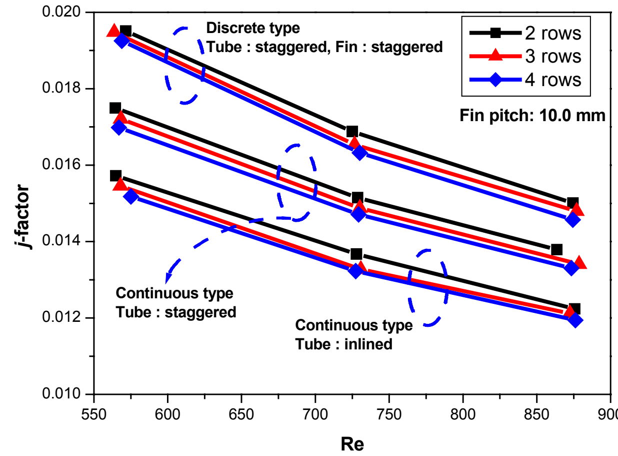

Figure 11. Effect of tube alignment on the heat transfer performance

4. CONCLUSIONS

The performance of finned-tube heat exchangers was experimentally investigated with a variation of heat exchanger geometries. The Colburn *j*-factor decreases with an increase of tube row, and the fin pitch shows negligible influence on the heat transfer performance at large fin pitches. The wavy shape discrete fin can improve the heat transfer performance by approximately 13%. The staggered fin alignment shows 7% higher heat transfer coefficients than the in-lined alignment, which effects are maintained regardless of number of tube row. The continuous flat plate finned-tube shows lower heat transfer performance than the discrete flat plate finned-tube. Applying staggered fin and tube alignments can improve the heat transfer performance by 20% as compared to the continuous flat plate finned-tube.

NOMENCLATURE

j	Colburn factor	
L	Longitudinal tube space	(mm)
Nu	Nusselt number	
Re	Reynolds number	

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ACKNOWLEDGEMENT

This work was supported by the Samsung Electronics Co. Ltd..