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Experimental Investigation on Effect of Fin Shape on the Thermal-Hydraulic Performance of Compact Fin-and-Tube Heat Exchangers

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Abstract. The aim of this paper is to investigate the effect of fin shapes on the performance of compact finned flat tube heat exchangers. Three types of fin shapes namely plain, wavy, and rectangular grooved fins attached to three by three arrays of flat tube banks were considered. Moreover, the tubes were deployed in in-line and staggered arrangements. In addition to the fin shapes, the air velocity and the tube inclination angles were varied and the thermal-hydraulic performance was analysed. On the other hand, the temperatures at the tube surfaces were kept constant to produce constant heat flux throughout the study. The results showed that as flowrate increases, the heat transfer increases, however, the friction factor decreases. Staggered arrangement produces higher heat transfer and friction factor than inline fin. Moreover, the rectangular fin is the best in terms of high heat transfer however the drawback of high friction factor leads the fin to have the least efficiency of all. On the other hand, plain fin had the least heat transfer performance however the highest efficiency was achieved. Therefore, plain fin should be used when efficiency is prioritized and rectangular fin when high heat transfer is desired.

1. Introduction

Compact fin-and-tube heat exchangers are used in many engineering applications such as power plants, transport, heating systems and air conditioning [1, 2]. The performance and efficiency of such types of heat exchangers are represented through the ratio of heat transfer over pressure drop (friction factor). Most process and power industries commonly use plate fin-and-tube heat exchangers for a wide temperature ranges. This is due to their compactness, close temperature approach. Moreover, they are also easy to inspect and clean [3]. There are different shapes of fins attached to the tube banks. For instance, in some applications metal plates are clamped as one used in conventional plate heat exchangers. The main types of fins used in compact heat exchangers include plane, wavy and rectangular fins.

The wavy fin and tube arrangement is famous plate fin and tube heat exchanger where it is used in applications that are constricted by high resistance to thermal on the air side of the heat exchanger [4]. The wavy fin surface is popular because it can increase the length of the airflow in the heat exchanger by improving or enhancing the air flow mixing and increasing heat transfer performance [5]. Furthermore, wavy fin and tube heat exchangers are cheap, reliable and easier to install or conduct maintenance [6]. The waviness of the fin enhances heat transfer because wavy fin has more surface area due to corrugations and it also promotes turbulence at bigger flows [7]. Similarly, the herringbone wavy fin corrugation also increases the contact area and promotes air flow mixing [4].

Most heat exchanger designs involve forced airflow through the exchanger, and the designs vary depending on specific requirements. These heat exchangers typically use fin-and-tube geometries and modified fin patterns to attain an enhancement of the air-side heat transfer, because the major limitation on the performance arises from the air-side heat transfer resistance. While conventional fin-and-tube heat exchangers have planar fins and round tubes, many contemporary designs have been developed with serpentine fins and flat tubes. To authors best knowledge, the performance differences between flat-tube heat exchangers with different fin shapes are not fully understood, because there have been much fewer investigations of the flat-tube geometries.

Thus, the main objective of the current study is to investigate the flow and heat transfer behaviours of flat tube heat exchanger with various fin shapes.

2. Methodology

2.1. Material Preparation

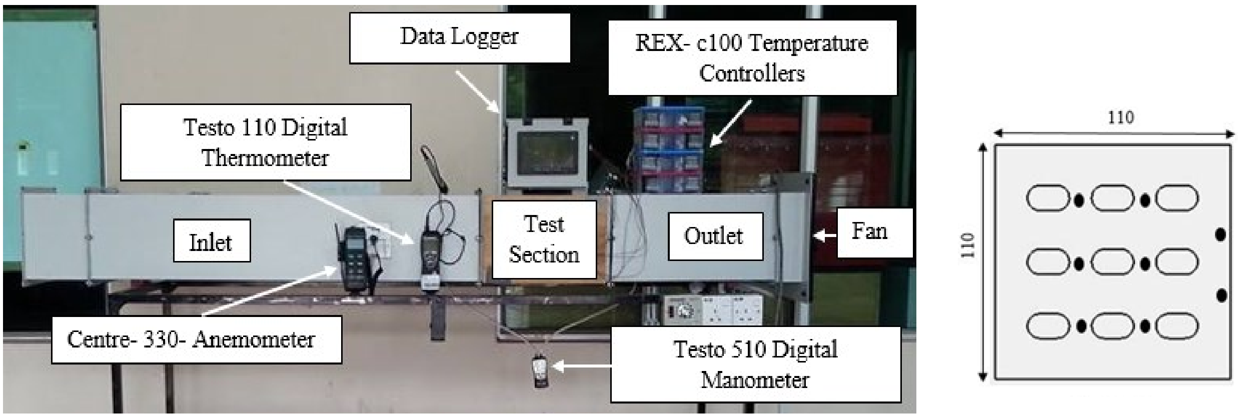
The specification of the test section available at the Faculty of Mechanical Engineering, Universiti Malaysia Pahang had to be taken in to consideration during designing of the fins. For the wavy fins, the parameters of the wavy angle and height were obtained from the works of Jang and Chen [8] where their study shows that the wavy angle of 162.1˚ and height of 1.5 mm are suitable for enhanced heat transfer. As for the rectangular grooved fin, there was no previous study made on the shape. Therefore, the parameters were set to be 1cm x 1cm x 1cm. After finalising on the fin parameters selection, the selected parameters were converted to Solidworks for ease of manufacturing. For each fin type, a total of 10 aluminum fins were used with constant fin spacing.

2.2. Experimental Setup and Procedure

Figure 1 shows the complete experimental setup a typical example of the location of temperature sensors for inline plain fin. The working fluid was air and the experiments were conducted in an open wind tunnel to obtain the thermal-hydraulic performance of the fin and flat tube heat exchanger. The system consists of inlet section, a flow straightener, a test section, and measuring devices for air velocity, air temperature and air pressure difference, temperature controller, a suction fan, and frequency inverter. The system operates as the suction fan allows the air to be sucked into the test rig where it flows over the fin and tube heat exchanger. The tubes were heated using electrical heating rod. The test section of the duct was designed to ensure that test sample could be changed easily. This arrangement operated in a cross-flow where the coolant air passes over the heat exchanger and the tube was heated by a hot electrical rod.

The wind tunnel is a square duct of 26 cm each side. A straightener was put at inlet to laminarize the flow to evade any distortion of air velocity. The distance between the straightener and the test setions inlet was 1.2m where the distance enables the air to be fully developedbefore the air enterd the test section. The wind tunnel walls were holed at the bottom side (before and after test section) to install pitot tubes which will be connected to digital manometer. The top side was holed to install digital thermometer which measures the temperature of air flow before it reaches the test section. Then, the air leaves the extension section and flows through the fin-and-tube HEs and then to the suction fan where it is discharged to the surrounding. The wind tunnel is fixed on top the stand made of steel which is 50 cm height to minimize the noise and vibration during the operation.

Axial fan model EPM is used as the suction fan driven by a 50 W AC motor. They fan speed was varied using frequency inverter however the air velocity could not be measured. Therefore, the air velocity was measured using Digital Anemometer where frequency inverter was set when the Anemometer measures the air velocity desired. Digital Anemometer also has the function of measuring the inlet temperature of air. Mainly, Testo 110 Digital Thermometer was used to measure the inlet air temperature. REX C-100 temperature controllers were used to maintain the surface temperature of the flat tubes where the heating rods deliver constant heat. K-type thermocouples were installed to every temperature controllers to closely monitor the flat tube surface temperature. The air pressure at the inlet and the outlet of the test section were measured using the digital differential pressure manometer of model TESTO 510. Data logger was used to collect the temperature distribution shown across the fins (Figure 1b). T- type sensors were used and placed at 8 points where 6 of them measure the temperature distribution in between the flat tubes and the other 2 sensors will read the oulet temperature. The position of sensors in the test section is as shown in Figure 1b and they were fixed using high-temperature resistant plaster. The black spot marks the position of the sensors placement. Other geometric parameter of the flat tube such as tube wall thickness, tube spacing, fin spacing, etc used in the experiment are presented in Table 1.



(a) **(b)**

Figure 1: (a) Complete experimental setup with measuring devices and (b) The location of temperature sensors in inline plain fin

Table 1: Geometric details of the test samples

Geometric parameter	Unit	Dimension (mm)
Tube wall thickness	δ	1
Transverse tube pitch	P_t	35
Longitudinal tube pitch	P_l	35
Tube length	L	220
Tube high outside diameter	D_{ho}	11.59
Tube wide outside diameter	D_{wo}	19.50
Fin thickness	t	0.6

2.3. Data Reduction

It is difficult to measure the wall temperature of the heat exchanger directly to acquire the convection heat transfer coefficient of the air side. Therefore, the experimental data was reduced and the thermal properties of the air was determined by the average values of the air at the inlet and outlet of the test section [9]. The main purpose of data reduction is to determine the heat transfer coefficient of the air side and pressure drop through the heat exchanger by calculating the nusselt number Nu, friction factor f and Colburn factor j of the flat tube heat exchanger from the experimental temperature data results which were recorded during each test run.

The heat transfer rate ($\mathrm{Q_a}$) of the air side which used to determine the heat transfer coefficient is computed by the following equation [10]:

$$Q_a = Q_{conv} = \dot{m}\, C_p(\Delta T) \tag{1}$$

where $\dot{m}$ is the mass flow rate; ΔT is the temperature difference between the outlet and the inlet air temperature and Cp is the specific heat capacity. Due to the small thermal conductivity of air which is lower than 0.23 (w/m.˚c) and the experiments were conducted at the room temperature, the heat transfer from the tube wall by conduction and radiation were neglected. Therefore, the energy balance can be expressed as following:

$$Q_{elec} = Q_{conv} = v \times I \tag{2}$$

where v and I are voltage and current. The forced convection heat transfer coefficient (h) expressed in terms of the heat transfer rate, total surface area and log mean temperature difference.

$$h = \frac{Q_{conv}}{A_t\, LMTD} \tag{3}$$

where A_t is the total heat transfer area and it is equal to the sum of the tubes and the fins surface area and LMTD is the log mean temperature difference. Thereafter, to evaluate the level of the heat transfer by convection in the air side Nu number is adopted in this study and it can be written as follows:

$$Nu = \frac{hD_h}{k} \tag{4}$$

The Colburn factor (j) and friction factor (f) of finned and unfinned flat tube have been calculated according to the following equations[11]:

$$j = \frac{h}{\rho v_{max} C_p} Pr^{\frac{2}{3}} \tag{5}$$

$$f = \frac{D}{L} \frac{2\Delta P}{\rho\, V_{max}{}^2} \tag{6}$$

where Δp is the pressure difference between the inlet and the outlet. The dimensionless number such as Nu number gives partial indication of the overall performance because sometimes the benefit gained from the heat transfer enhancement are not great enough to overcome the increase in friction losses. Therefore, thermal performance coefficient JF was calculated to compare the air side performance. Therefore, the performance evaluation criteria of the finned and flat tube heat exchanger is defined as [12]:

$$PEC = JF = \frac{j}{f} \tag{7}$$

3. Results and Discussion

3.1. Heat Transfer Performance

The heat transfer performance of the heat exchanger was evaluated in terms of dimensionless parameter Nusselt number. The main purpose was to test the effects of geometric (tube arrangement, fin shape and tube inclination angle) and flow parameters on the heat transfer performance of compact flat tube-and-fin heat exchanger.

3.1.1. Effects of flowrate.
Figure 2 shows variation of Nusselt number for plain fins against Reynolds number for inline configuration. The presented result shows that as the flow rate increases so does the Nusselt number. Moreover, it shows that the Nusselt number is directly proportional to the flowrate. This means the increase in the air velocity yields increment in the convection heat transfer of the heat exchanger. Results with similar trend has been seen from the simulation work by Kumar et al.[13]. The increasing trend is seen due to the better mixing of flow.

3.1.2. Effect of Tube Arrangement.
The comparison between inline and staggered configuration with different fin shapes and Reynolds number range was also analysed. This is to understand the effect of inline and staggered configurations on the heat transfer performance of the heat exchanger. Figure 3 shows the difference between Nusselt number variation of inline and staggered configurations of rectangular fin heat exchanger. The figure shows that the heat transfer performance of staggered configuration is higher than the inline configuration by about 7.5%. Similar trend was obtained for wavy and plain finned heat exchangers. This is due to the reason that there exist distortion to the velocity and the temperature profiles of the air flow which enhances the heat transfer properties. Another way of describing the reason for enhancement in staggered is due to the increase in the intensity in turbulence.

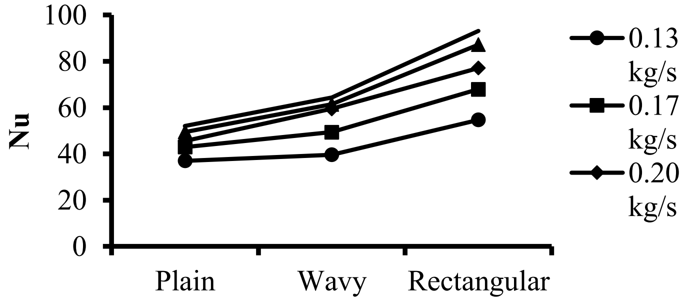


Figure 2: Variation of Nusselt number against different fin at different flowrate

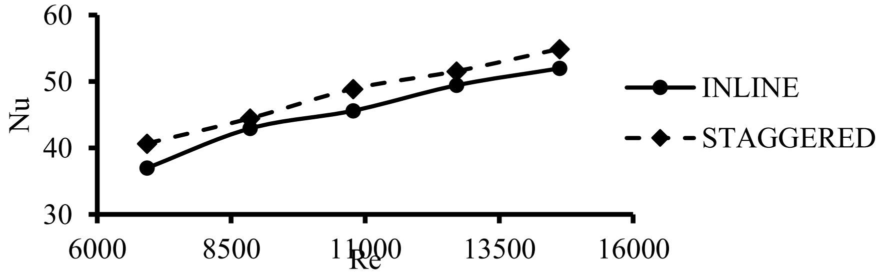


Figure 3: Nusselt number against Reynolds number for inline and staggered plain fin

3.1.3. Effect of Different Fin Shapes.
The performance of heat transfer affected by different fins shapes are presented in Figures 4a and 4b for inline and staggered arrangements respectively. The results show that the performance increases in plain, wavy, and rectangular order. Considering plain fin as reference, wavy fin shows 30% of increased performance and rectangular fin produces 69% higher performance than plain fin for inline. Rectangular fin produces the highest heat transfer performances due to the interruption

done by the staggered surfaces to the flow and temperature boundary layers along the flow orientation. Similar arguments were observed in the works of Sanaye et al. [14].

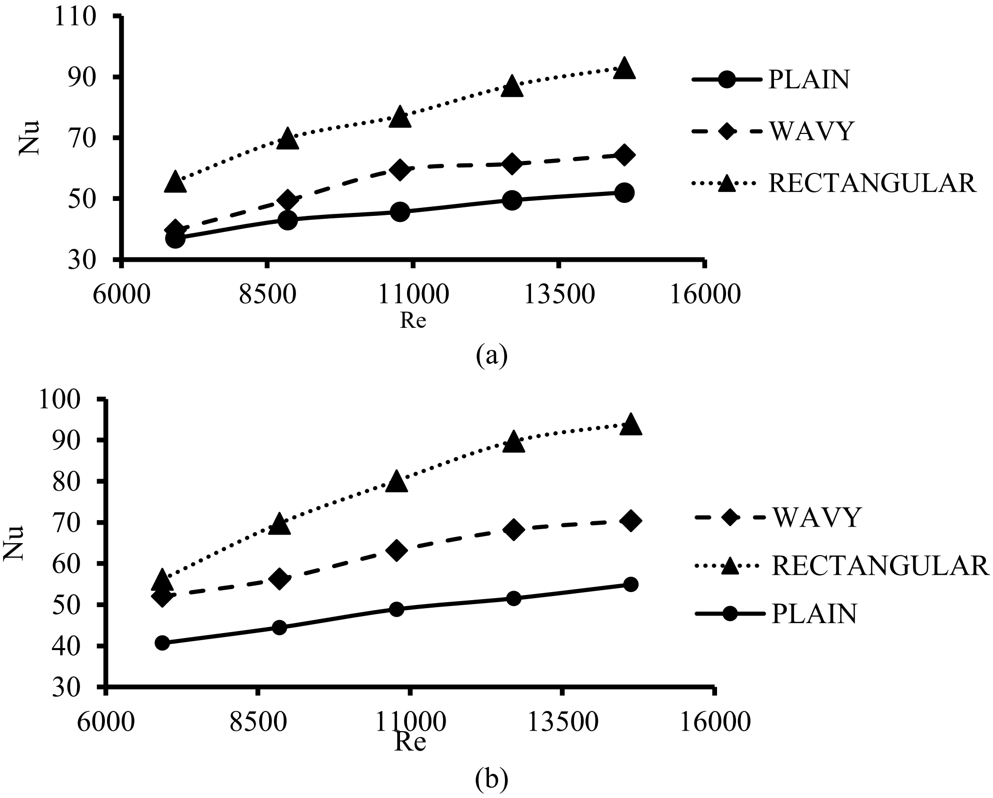


Figure 4: Variation of Nu against Reynolds number for plain, wavy, and rectangular fins for (a) inline and (b) tube arrangements

3.2. *Hydraulic Performance*

The hydraulic performance is presented in terms of friction factor, f. The main purpose is to test the effects of geometric (tube arrangement, fin shape and tube inclination angle) and flow (Reynolds number) parameters on the hydraulic performance of compact flat tube-and-fin heat exchanger.

3.2.1. Effects of air flow velocity.

Figure 5 shows variation of friction factor against Reynolds number for inline wavy fin. As the air flow velocity increases the friction factor decreases. This relationship is proved through equations where friction factor is inversely proportional to Reynolds number. As the air velocity increases the boundary layer thickness of air flow over the fin decreases and this produces decreasing trend of viscous force where viscous force is proportional to the friction factor.

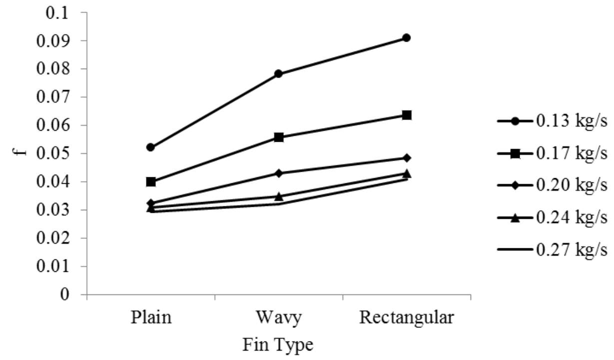


Figure 5: Friction factor against different fin at different flowrate

3.2.2. Effect of Tube Arrangement.
Figure 6 shows the comparison between inline and staggered tube arrangement against friction factor. The result shows that staggered arrangement has higher friction factor about 16.7% compared to inline tube arrangement. The results obtained for wavy and rectangular fins shows the same trend. This is due to the higher intensity in turbulence where the temperature and velocity profiles are being distorted significantly for staggered arrangement.

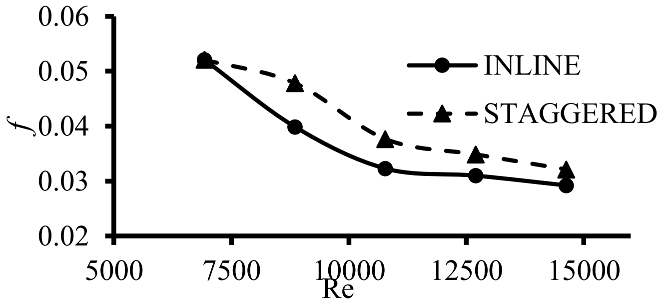


Figure 6: Comparison of friction factor between inline and staggered arrangement for plain fin

3.2.3. Effect of Different Fin Shapes
Figure 7 shows variation of friction factor against Reynolds number for different fins. The result shows that rectangular fin has the highest friction factor about 100% higher than plain fin meanwhile friction factor of wavy fin is higher by about 49.8% than plain fin. This is due the interruption that wavy and rectangular fin geometry does to the flow of air. Higher turbulence intensity leads to high friction factor.

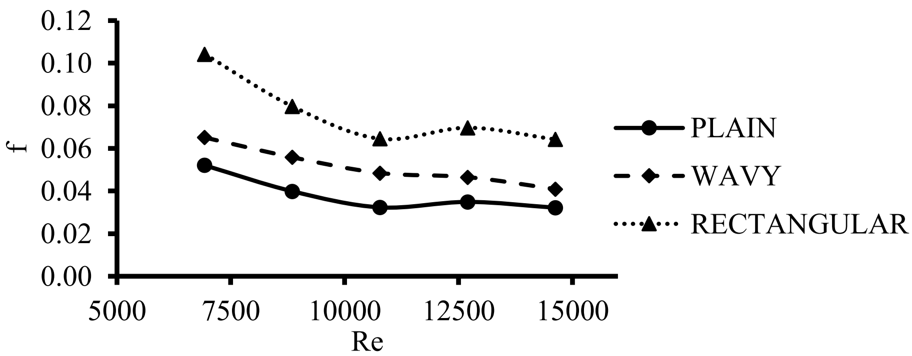


Figure 7: Friction factor against Reynolds number for plain, wavy, and rectangular fins with inline tube arrangement

3.3. Performance Evaluation Criteria
Figure 8 shows that plain fin has the highest efficiency of performance where even though plain fin has low heat transfer performance, it also has the lowest friction factor, therefore higher efficiency is obtained. On the other hand, wavy and rectangular fins have higher heat transfer performance; however, the drawback of high friction factor affects the efficiency of the fins. The usage of fin type differs according to the application where some prefer efficiency which plain fin excels. Some other applications prefer high heat transfer performance where wavy and rectangular fin excels.

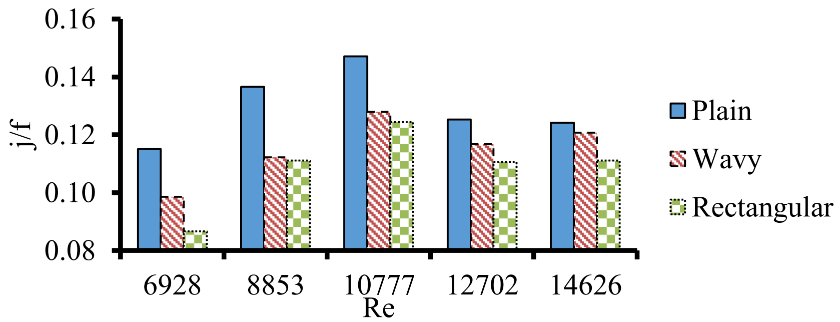


Figure 8: Efficiency of different fins against Re

4. Conclusion
From this study, it was found that the rectangular fin has the highest heat transfer performance compared to wavy and plain fin where wavy fin is higher than plain fin. Rectangular fin produces the highest heat transfer performances due to the interruption done by the staggered surfaces to the flow and temperature boundary layers along the flow orientation. The hydraulic performance has the similar trend where rectangular has the highest pressure drop compared to wavy and plain fin. This is due the interruption that wavy and rectangular fin geometry does to the flow of air. The pressure drop of plain fin is low compared to complex design of wavy and rectangular fin. Wavy and rectangular fin despite having higher heat transfer performance, they have a greater drawback in higher pressure drop. On the other hand, rectangular fin is suited for any application which prioritises thermal performance over hydraulic performance or efficiency.

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