Characteristics of Heat Transfer in Parallel Plates with Installed Transversal Twisted Tapes

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Abstract

This research has studied the characteristics of heat transfer for turbulent flow within parallel plates with installed transverse twisted tapes having varying twist ratios. The height of the twisted tape (w) equals to 6 millimeters, where the ratio of the gap between the twisted tapes over its height (P/w) equals to 10 millimeters with modifications made to the ratio of twisting (y/w) equals to 1.56, 3.13, 6.25, 12.5 and 25 installed at the lower wall of the plate. The experiment is to be done under the state of constant heat flux within the parallel plates with a ratio of the width over height (Aspect Ratio, AR) that equals to 3.75. The experiment is done when the Reynolds number ranges from 9,000 to 24,000. To consider the values of heat transfer as presented with the Nusselt number (Nu), and pressure loss in a form of friction factor (f), it is found that the parallel plates installed with twisted tapes resulted in heat transfer value and friction factor of 1.21-1.56 times and 1.79-2.49 times higher accordingly, when compared to plane tubes. The ratio of twisting though, equals to 1.56 at Reynolds number of 9,000 which resulted in highest value of heat transfer efficiency (η) that equals to 1.24.

Keywords: Channel flow, heat transfer enhancement, transversal twisted tapes.

1. Introduction

A heat exchanger is a device designed to allow heat transfer from one fluid (liquid or gas) to another fluid through a tube wall without the fluids mixing. Therefore, heat transfer rate improvement is necessary to take into consideration as it directly results in a smaller heat exchanger size. In-tube convective heat transfer modifications of heat exchangers are divided into two groups: active and passive methods. The active method requires additional power activation from an external source to energize heat transfer rate. In contrast, an external power is not necessarily required for enhancing heat transfer rate in the passive method; only surface or geometric modification by introduction of some devices in the flow channel will be needed [1-2]. Both of heat transfer enhancements have been widely employed in several heat transfer systems as follows: nuclear reactors, chemical reactors, and process heat exchangers. Swirl flow, generally applied in a variety of engineering applications, is one of the most well-known techniques to boost heat transfer rate in the passive group. Swirl flow generators may be found in several forms, for example: twisted tape, helical screw tape, propeller, axial and radial guide vanes, snail entry, etc. [3-7]. Turbulence flow near the tube wall created by swirl flow generator results in a thinner boundary layer and a longer residence time of the fluid in the tube. Higher turbulent intensity of the fluid adjacent to the tube wall as a result of the swirl flow generator is responsible for an excellent fluid mixing and efficient redevelopment of the thermal/hydrodynamic boundary layer.

Most of studies have paid attention to heat transfer enhancements in circular tubes while case with rectangular channels are quite limited. Eiamsa-ard et al. [8] investigated the heat transfer and pressure loss characteristics in a circular tube with short-length twisted tape inserted. They have found that those with short-length twisted tape inserted, has a great effect on enhancing heat transfer rate, while the pressure drop is surged. Krishna et al. [9] experimentally studied the heat transfer characteristics of a circular tube equipped with straight full twist under the turbulent flow regions. The results showed that the heat transfer coefficient straightly increased with the Reynolds number, but decreased with large spacer distance. Eiamsa-ard [10] has studied the thermal and fluid flow characteristics in the turbulent flow with multiple twisted tapes. Their results have revealed that, the decreases of free-spacing ratio and twisting ratio resulted in the increase in Nusselt number values, friction factor values and thermal enhancement index. Insertion of twisted tapes have been extensively studied in heat exchanger systems to reduce the thermal boundary layer thickness that results in enhancing the heat transfer performance. As the above has mentioned, most researches mainly focused on heat transfer enhancement and fluid friction behaviors in the circular tubes equipped with various styles of twisted tapes rather than for the rectangular channels. The advantages of twisted tapes have lighten up the motivation for us to develop a new design for rectangular channels.

Therefore, the goal of the present investigation is to study heat transfer and flow friction characteristics of air heated in a rectangular channel equipped with multiple twisted tapes. The effects of tape feature and arrangement on Nusselt number, friction factor and thermal enhancement index characteristics in the turbulent channel flows are examined for Reynolds number ranging from 9,000 to 24,000. Five twisting ratios: y/w=1.56, 3.13, 6.25, 12.5 and 25, were performed in this experiment. Importantly, in order to gain confidence in the received results, the heat transfer and friction factor in the case of plain channel have been verified by comparing with previous correlations.

To date, twisted tapes can perform in compact heat exchangers because of its low cost, ease of maintenance and manufacture. However, at some operating conditions, the huge friction losses within the heat exchangers are generated, causing an elevation in pumping power.

2. Data reduction theory

To consider the efficiency of heat transfer of parallel plates with installed transversal twisted tapes, it is required to also consider the energy needed to propel the fan and heat transfer of the passing-by air. The energy that the fan requires to propel the air and lead its way through the parallel plates can be considered as:

$$\dot{W}_{in} = \dot{V}\Delta P \tag{1}$$

where $\dot{W}_{in}$ is the pumping power (W), $\dot{V}$ is the volume flow rate (m^3/s) and ΔP is the pressure drop across the test section (N/m^2 or Pa).

The friction factor (f) across the test section is computed using the following equation:

$$f = \frac{\Delta P}{\left(\frac{L}{D_h}\right)\left(\rho_b \frac{V_i^2}{2}\right)} \tag{2}$$

where L is the length of the test channel (m), D_h is the Hydraulic diameter (m), ρ_b is the density of bulk air temperature (kg/m^3) and V_i is the inlet air velocity (m/s).

Energy balance equation of constant heat flux condition

$$Q = \dot{m}C_p\left(T_o - T_i\right) = hA_s\left(T_w - T_b\right) \tag{3}$$

and

$$T_b = \left(T_i + T_o\right)/2 \tag{4}$$

where $\dot{m}$ is the mass flow rate (kg/s), C_p is the specific heat capacity of (J/kg K), T_i and T_o are the inlet and outlet air temperature (K), respectively, h is the heat transfer coefficient (W/m^2 K), A_s is the area of absorber surface (m^2), T_w and T_b are the wall and mean bulk air temperatures (K), respectively.

where

$$T_w = \sum_{j=0}^{n}\sum_{i=0}^{m}\left(T_{i,j}\right) \tag{5}$$

where i and j are both the positions that locate pixel in x and y axis accordingly.

The convective heat transfer coefficient was then used to obtain the Nusselt number, as follows:

$$Nu = \frac{hD_h}{k} \tag{6}$$

Where k is the thermal conductivity of air (W/m K).

and an equation used for the division of fluid flow within the channels will consider the Reynolds number that can be calculated from the following equation:

$$\mathrm{Re} = \frac{VD_h}{\nu} \tag{7}$$

Where ν is the kinematics viscosity of air (m^2/s).

The value of heat transfer efficiency (η), considering the loss of constant pumping power between the smooth channels and the channels installed with transverse twisted tapes, can have its equation written in a form of an energy equation as follows:

$$\left(\dot{V}\Delta P\right)_0 = \left(\dot{V}\Delta P\right) \tag{8}$$

The term on the left side of "Eq. (6)" is a power used for the flow of air within empty channels, while the term on the right side of the equation is a power used for the flow of air within channels installed with transverse twisted tapes.

and the relationship between friction factor and Reynolds number can be expressed as:

$$\left(f\,\mathrm{Re}^3\right)_0 = \left(f\,\mathrm{Re}^3\right) \tag{9}$$

Also,

$$\mathrm{Re}_0 = \mathrm{Re}\left(f/f_0\right)^{1/3} \tag{10}$$

Consideration of heat transfer efficiency has been defined from the proportion of the convection coefficient value of test channels that have been installed with transversal twisted tapes (h) and the convection coefficient value of smooth channels (h_0) or defined from the proportion of the Nusselt number value in test channels installed with transversal twisted tapes (Nu) and the Nusselt number value of smooth channels (Nu_0) as suggested by Webb [11].

$$\eta = \frac{h}{h_0} = \left(\frac{Nu}{Nu_0}\right)\left(\frac{f}{f_0}\right)^{-1/3} \tag{11}$$

where h_0 is the convective heat transfer coefficient for the smooth channel (W/m^2 K), and Nu_0 and f_0 are the Nusselt number and friction factor of smooth channel, respectively.

3. Multiple twisted tapes

Details of a rectangular channel equipped with transversal twisted tapes are presented in "Fig. 1". The test channel was characterized by the channel height (*H*) of 40 mm, and the overall length of the test section (*L*) of 600 mm with the channel width (*W*) of 150 mm (*AR*=3.75). Multiple twisted tapes were made of PLA plastics with the tape width (*w*) of 6 mm, tape thickness (*t*) of 2 mm and tape length (*l*) of 150 mm. The dimension of single twisted tapes were the same as those of the multiple twisted tapes. Transverse twisted tapes (TTTs) were prepared with five different twist ratios, *y*/*w*=1.56, 3.13, 6.25, 12.5 and 25 ("Fig. 2"), were performed in this experiment. Multiple twisted tapes were placed along the whole length of a smooth heater plate.

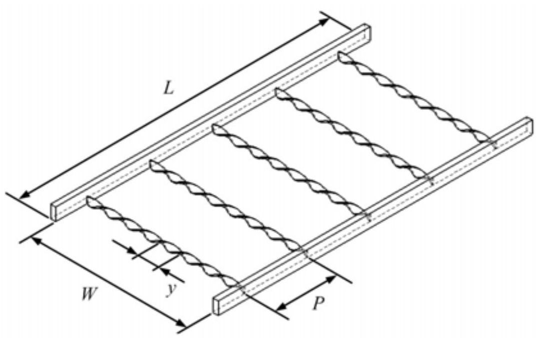

Fig. 1. Configuration of the channel fitted with transversal twist tapes.

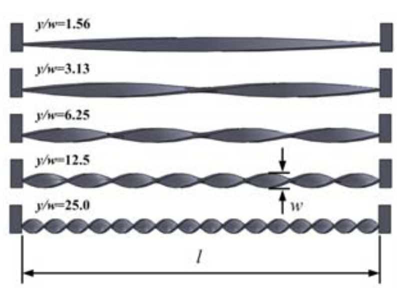

Fig. 2. Geometry parameters of the TTTs installed.

4. Experimental apparatus

4.1 Experimental setup

The experimental apparatus consisted of an entrance channel, a test section, an air supply system (high pressure blower) and a heating preparation. A schematic diagram of the experimental apparatus with the basic components are presented in "Fig. 3". The tube shaped entrance channel, 2000 mm long, was made as an integral part of the test section to avoid any flow disturbances against and to get a fully developed flow in the test section as well. The entrance channel of the experimental setup was made as per suggested by Ower and Pankhurst [12] to avoid separation and stratification of the fluid flow.

4.2 Procedure

Air, as the tested fluid (type: gas) in both the heat transfer and pressure loss experiments, was aimed and directed into the systems by a 3-hp high pressure blower. The operating speed of the blower was alternated by using an inverter to provide desired air flow rates. The air flow rate was measured by using an orifice meter which was built according to the ASME standard [13] and was calibrated with a hot-wire anemometer. The pressure drop across the orifice meter was measured using a pressure differential transmitter (PDT).

The experiments were conducted for the Reynolds number ranging from 9,000 to 24,000. The AC power supply was the source of power for the plate-type heater, used for heating the lower wall of the test section. All details of the experiment setup are presented in Table 1.

Table. 1. Dimensionless quantities and other parameters used in experiments of test channel.

Test section	
Height of test channel, *H* (mm)	40
Width of test channel, *W* (mm)	150
Length of test channel, *L* (mm)	600
Aspect ratio, *AR*	3.75
Transversal twisted tapes	
Material	PLA Plastics
Thickness of tapes, *t* (mm)	2
Height of tapes, *w* (mm)	6
Width of tapes, *l* (mm)	150
Free-spacing length, *P* (mm)	60
Length of twisting, *y* (mm)	9.37, 18.75, 37.5, 75 and 150
Test condition	
Working fluid	Air
Reynolds number, Re	9,000 to 24,000
Prandtl number, Pr	0.7

The tube wall (channel surface) temperatures, bulk air temperatures, the pressure drop across the test section and the air flow rate velocity were measured and recorded to evaluate and analyze the Nusselt number and friction factor of the heated tube. Measuring channel surface temperature with Thermochromic liquid crystal is an indirect exposure method. It involves the characteristics of changing Thermochromic liquid crystal's color continuously, according to temperature variation. With a digital camera that captures high resolution images along with an image analyzing computer program, it enables the distribution of temperature to be measured conveniently and accurately. The colors of the images were captured as *RGB* images and were changed into *HSV*, where the relationship between Hue values (*H*) in the *HSV* system and the temperature (*H*=*H*(*T*)) were further used to predict the surface temperature. The average surface temperature can be calculated with "Eqs. (4)–(5)" and the position used in the calculation to find the surface temperature is shown in "Fig. 4".

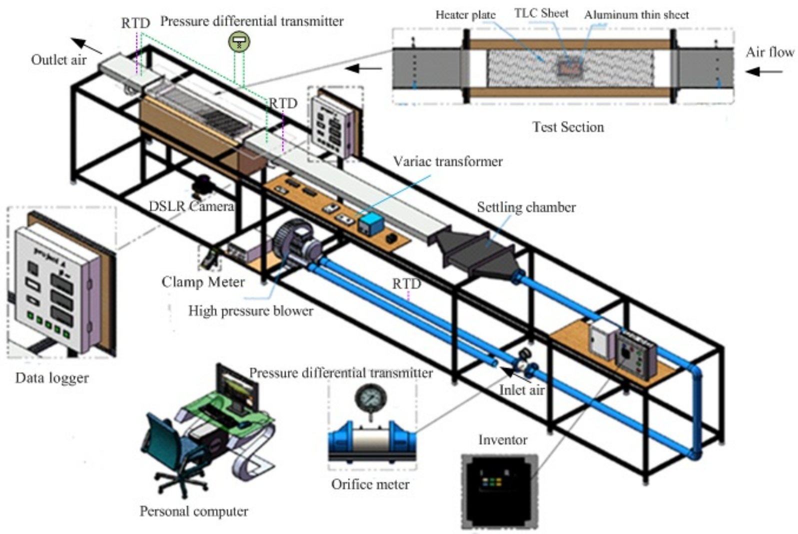

Fig. 3. Schematic diagram of the experimental facility.

The bulk air temperatures were measured with the help of RTDs. Data logger was used to record the experiment data of the test section. The pressure drop across the test section was measured with a pressure differential transmitter, and the heat transfer experiment was performed under a constant heat flux condition. In contrast, the pressure drop test was conducted under an isothermal condition without turning on the heater.

The uncertainties in the experimental measurements were determined by using the method introduced by Kline and McClintock [14]. The maximum uncertainties of non-dimensional parameters were ±5% for Reynolds number, ±6% for Nusselt number and ±4% for friction factor.

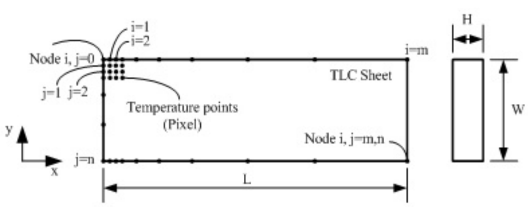

Fig. 4. Location of TLC points on absorber plate and outlet temperature of test section.

5. Results and discussion

5.1 Results from smooth channel tube verification

Before conducting the experiment, this study has examined the results of heat transfer and pressure drop within test channels without the installation of transversal twisted tapes and was then compared to the reliable correlation basics, where the results of heat transfer in the form of Nusselt number were compared to Dittus-Boelter's correlation. On the other hand, the pressure drop in the term of friction factor values were compared to the correlation of Blasius [15].

Conditions of correlation for smooth channels by Dittus-Boelter; for use in Re>7,000 and 0.5<Pr<120 will be described as the following equation:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{n} \qquad (12)$$

Where n=0.4 is for heating and n=0.3 is for cooling.

Therefore, the equation of correlation that has been used in the finding of the Nusselt number of smooth channels can be modified as the following equation:

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \qquad (13)$$

and Blasius's correlation can be calculated from

$$f = 0.085\,\mathrm{Re}^{-0.25} \quad \text{for Re<20,000} \qquad (14)$$

$$f = 0.184\,\mathrm{Re}_D^{-0.2} \quad \text{for Re}\geq 20{,}000 \qquad (15)$$

From “Fig. 5” and “Fig. 6”, it is found that the data obtained from the experiment has a tendency to be similar to the basic correlation values in every section of the experimental conditions and the Nusselt number tolerance values along with friction factor values were at ±6% and ±4% accordingly, once compared to the correlation of Dittus-Boelter and Blasius.

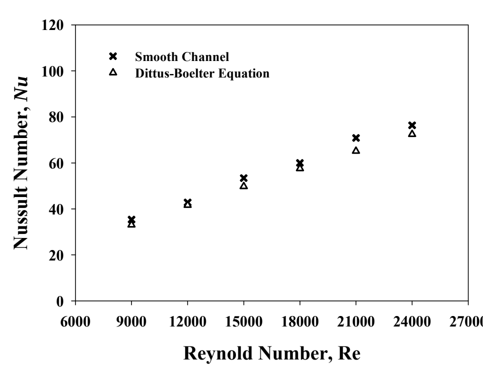

Fig. 5. Comparison of experimental and estimated values of Nusselt number for smooth channel.

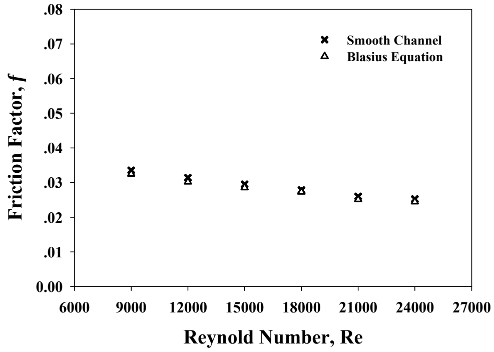

Fig. 6. Comparison of experimental and estimated values of friction factor for smooth channel.

5.2 Nusselt number distribution

The distribution of Nusselt number values for this research study has been displayed as the Reynolds number of 24,000 where the transversal twisted tapes with the ratio of twist adjusted to (*y*/*w*) equals to 1.56, 3.13, 6.25, 12.5 and 25 as displayed in "Figs. 7(a)-7(f)". In case of typical baffles installed, refer to "Fig. 7(a)".

From "Fig. 7(a)" which refers to the installation of typical baffles (TBs), it can be observed that the recirculation zone has been formed at the area between the baffles. Recirculation highly affects heat transfer as it increases the mixing of fluids between the surface that allows heat transfer with air at the core region. However, with installed transversal twisted tapes in all experimental conditions, as displayed in "Fig. 7(b)" into "Fig. 7(f)", it can be observed that there is no recirculation zone formed in the area as stated earlier and therefore resulting in the area between the twisted tapes having a drop in heat transfer efficiency. On the contrary, heat transfer efficiency increased in the area the twisted tapes are installed due to air and its entrance that refracted with the surface and the back side of the twisted tapes, resulting in higher heat transfer in the latter area (in "Fig. 8").

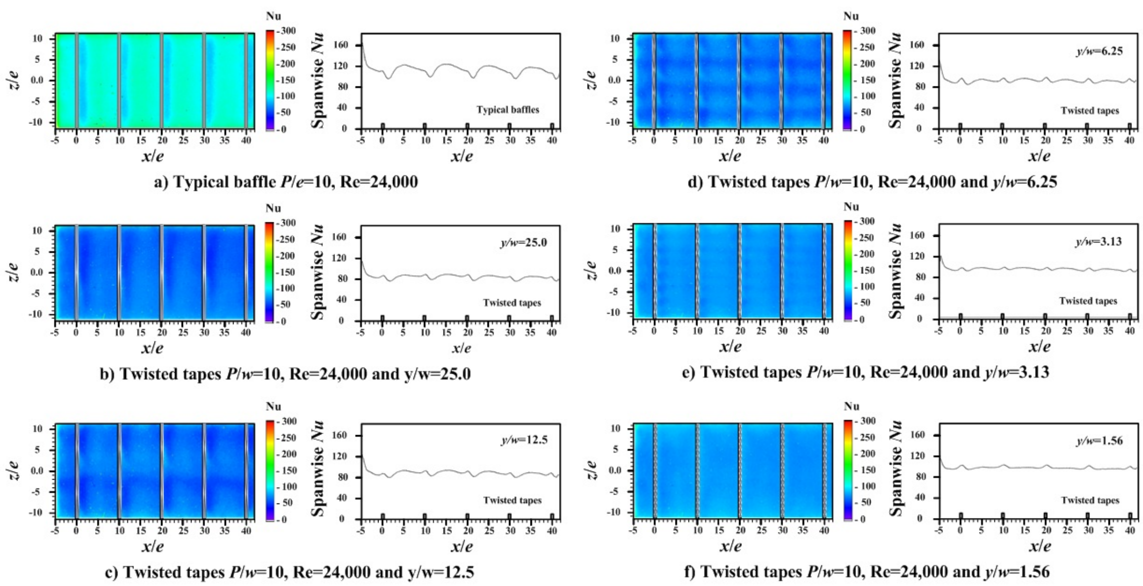

Fig. 7. Nusselt number distribution and spanwise average Nusselt number on the heat transfer surface: (a) typical baffles, (b-f) transverse twisted tapes (*y*=1.56*w*, 3.13*w*, 6.25*w*, 12.5*w* and 25.0*w*).

5.3 Effects of twisting ratio

"Fig. 9" and "Fig. 10" has shown the increase in Nusselt number when the Reynolds number increased due to higher turbulent intensity. In each every experimental condition, all values are higher than that of smooth channels.

In similar experimental conditions at the twisting ratio of *y*/*w*=1.56 gave off the highest heat transfer values and reached the peak with Reynolds number of 9,000, where the average Nusselt number in the case where twisted tapes are installed at twisting ratio of (*y*/*w*) equals to 1.56, 3.13, 6.25, 12.5 and 25 gave off 1.47, 1.42, 1.37, 1.32 and 1.26 times accordingly, once compared to the smooth channels.

The research study has also found the decrease in *y*/*w* value affecting the space where air can go through the opening of the twisted tapes, increasing in values. The air passing by the twisted tapes stroke, refracted, and impinged the surface where heat transfer is to happen. Therefore, heat transfer in the latter area increased. However, in the installation of the twisted tapes, the recirculation zone or eddy motion was absent, resulting in the decrease in heat transfer at the area between the twisted tapes. In which the above relates to the previous research article by Wayo et al. [16] and Wai Peng et al. [17]

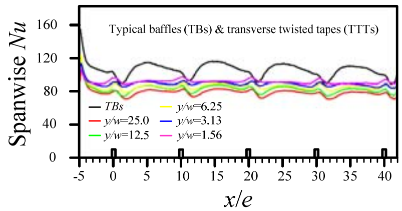

Fig. 8. Spanwise average Nusselt number on the heat transfer surface wall.

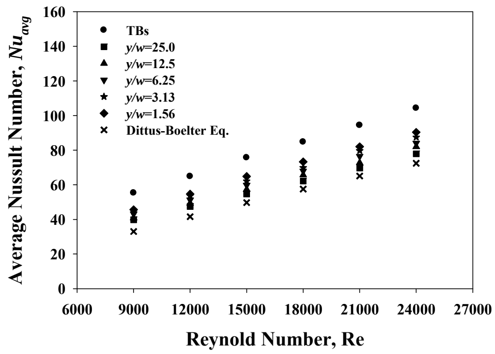

Fig. 9. Effect of twisting ratio values on Nusselt number.

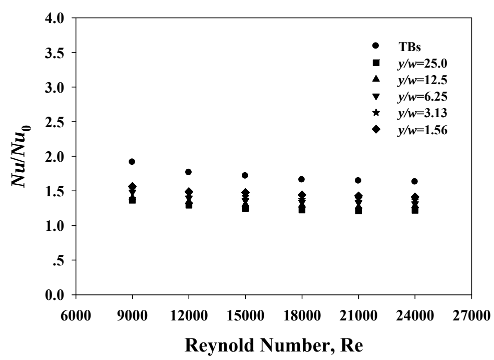

Fig. 10. Effect of twisting ratio values on Nusselt number ratio.

"Fig. 11" and "Fig. 12" has shown the trends of decreasing friction factor when the Reynolds number increased. It can as well be observed that when the twisting ratio or the Pitch distance increased, more air could pass through the twisted tapes, which resulted in a decrease in momentum loss and pressure drop within the system.

Friction factor values for the installation of twisted tapes were in section 1.79 to 2.49 times which are more than smooth channels once compared. Installation of twisted tapes into the system to create turbulence has influenced the increase in friction in the case of smooth channels due to the decrease in dynamics pressure of air, viscosity of fluids in areas where the channel surface is higher, and force from the action of fluids- caused by the penetration of fluids. Apart from the latter that has been stated, pressure and inertia at the boundary layer may be the causes.

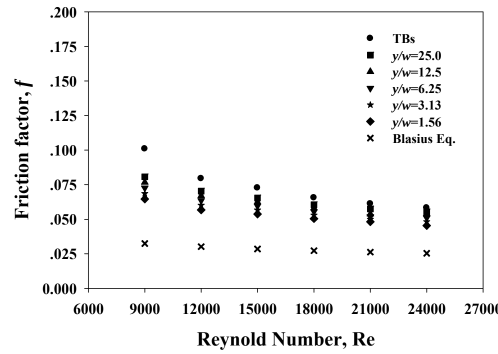

Fig. 11. Effect of twisting ratio values on friction factor.

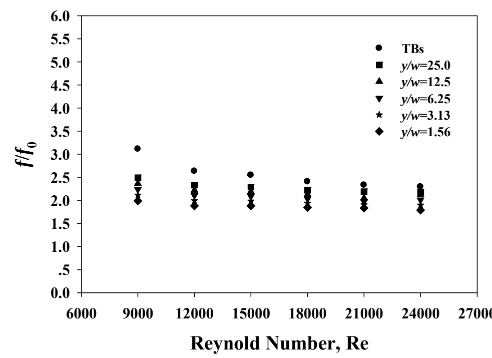

Fig. 12. Effect of twisting ratio values on friction factor ratio.

5.3 Thermal enhancement index

Thermal enhancement index in this research is shown in "Fig. 13". The results of the research study has shown that thermal enhancement index decreased when Reynolds number increased, and therefore has achieved the highest value at the Pitch value or twisting ratio (y/w) of 1.56 with Reynolds number of 9,000 which equals to 1.24.

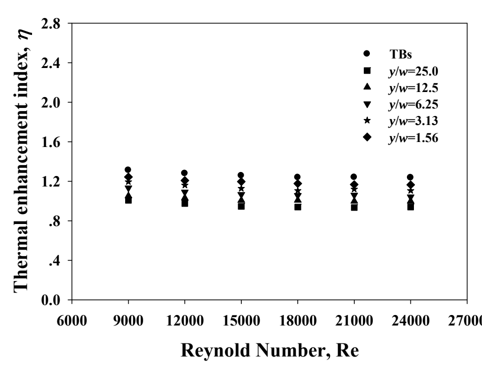

Fig. 13. Variation of thermal enhancement index with Reynolds number.

Thermal enhancement index of the heat exchange equipment installed with twisted tapes has trends of decreasing values following the system's energy loss. The best Reynolds number section suitable for this study is between 9,000 to 15,000. Thermal enhancement index for the case where twisted tapes are installed, had values from 0.93 to 1.24 once it has been compared to similar constant pumping power.

6. Conclusion

This article aims to present the results of heat transfer and friction of heat exchangers within a rectangular channel installed with twisted tapes, with twisting ratio (y/w) of 1.56, 3.13, 6.25, 12.5 and 25 in Reynolds number section 9,000 to 24,000. The main results of the experiment done with installed twisted tapes can be summarized as follows:

- With twisted tapes installed, heat transfer in the area where the twisted tapes are located increased and as well enabled the decrease in pressure loss within the system more than cases with typical baffles installed. Nevertheless, at the area between the twisted tapes, heat transfer is observed to be lower than that of the typical baffles, due to the absence of recirculation zone at the areas between the twisted tapes. Therefore, it led to a decrease in heat transfer value.
- The average Nusselt number was between section 1.26 to 1.56 times in the case with twisted tapes installed, once compared to smooth channels.
- Thermal enhancement index was observed to be around 0.93 to 1.24 in the rectangular channel with multiple transverse twisted tapes installed.
- Installed twisted tapes at the twisting ratio that (y/w) equals to 1.56 at Reynolds number of 9,000 gave off thermal enhancement index value of 1.24 which is the highest value compared to other conditions in the experiment.

7. Acknowledgement

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