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Enhancing Heat Transfer in Tube Heat Exchanger by Inserting Discrete Twisting Tapes with Different Positions

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ABSTRACT

Enhancement of heat transfer in the tube heat exchanger is studied experimentally by using discrete twisted tapes. Three different positions were selected for inserting turbulators along tube section (horizontal position by $\alpha = 0^0$, inclined position by $\alpha = 45^{\ 0}$ and vertical position by $\alpha = 90^0$). The space between turbulators was fixed by distributing 5 pieces of these turbulators with pitch ratio PR = (0.44). Also, the factor of constant heat flux was applied as a boundary condition around the tube test section for all experiments of this investigation, while the flow rates were selected as a variable factor (Reynolds number values vary from 5000 to 15000). The results show that using discrete twisted tapes enhances the heat transfer rate by about 60.7-103.7 % compared with plane tube case. Also, inserting turbulators with inclined position offers maximum heat transfer rate by 103.7%.

Keywords: tube heat exchanger, turbulators, twisted tapes.

تحسين انتقال الحرارة في مبادل حراري انبوبي بحشر اشرطة ملتوية منفصلة بوضعيات مختلفة

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الخلاصة

تحسين انتقال الحرارة في مبادل حراري انبوبي دُرس عملياً باستخدام أشرطة ملتوية منفصلة. أُختيرت ثلاث وضعيات مختلفة لإدخال ادوات الحشر على طول انبوب الاختبار هي (الوضعية الافقية $\alpha = 0^0$ ، الوضعية المائلة $\alpha = 45^0$ والوضعية العمودية $\alpha = 90^0$). المسافة بين قطع ادوات الحشر ثُبتت بتوزيع 5 قطع من هذه المضطربات بنسبة PR=(0.44). بالإضافة الى ذلك، تم تطبيق فيض حراري ثابت كشرط حدي لكل التجارب في هذا التحقيق، بينما اختيرت معدلات تدفق الجريان كعامل متغير (قيم عدد رينولدز تتراوح بين 5000 الى 15000). أظهرت النتائج بأن استخدام الاشرطة الملتوية المنفصلة تُحسن معدل انتقال الحرارة بنسبة حوالي من 60.7 الى 103.7 % مقارنة مع حالة الانبوب المستوي. اضافة الى ذلك، حشر المضطربات بالوضعية المائلة يعرض اعظم معدل لانتقال الحرارة بنسبة 103.7%.

الكلمات الرئيسية: مبادل حراري انبوبي، مضطربات، اشرطة ملتوية.

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1. INTRODUCTION

Heat exchangers are considered one of the different instruments that convert energy, **Elias, et al., 2014.** Therefore, many methods appeared to enhance their performance. Using augmentations which are called passive technique is considered the simplest method to do this purpose. This technique is based on inserting turbulators such as nozzles, twisted tape, rings…etc. inside heat exchanger to increase turbulent fluctuations which cause the reducing thickness of thermal boundary layer near the tube wall, hence, achieving high heat transfer, **Promvonge, et al., 2015.** But, this technique has a limitation that is inserting turbulators leads to increasing pressure drop. Therefore, researchers tried to solve this problem through choosing of turbulators. Twisted tapes turbulators are commonly used in heat exchangers for enhancing heat transfer due to its simplicity in both manufacturing and installation; moreover, their performance is considered stable. Therefore, researchers put their efforts to manipulate using twisted tapes in different ways that ensure extra enhancement in its performance. For example, using continuous corrugated twisted tapes with different twisted ratios offers increasing in heat transfer efficiency between 18% and 52 %, **Patil, et al., 2015.** Making twisted tapes from different materials such as aluminum or M.S. material, and investigated their effect on heat transfer improvement, **Shinde, et al., 2015.** Inserting different numbers of twisted tapes (single, double, triple, and quadruple), **Chokphoemphun, et al., 2015.** or inserting multiple sizes of twisted tapes (small size of twisted tapes inserted with larger tapes), **Piriyarungrod, et al., 2018.** or inserting double perforated twisted tapes, **Bhuiya, et al., 2016.** were demonstrated as a useful way for enhancing the heat transfer coefficient. Moreover, making holes in twisted tapes with different shapes like square holes and variable perforation width ratios gives extra enhancement in heat transfer rate, **Suri, et al., 2017.** Other ways of manipulating using twisted tapes to enhance heat transfer performance are inserting different types of baffles like alternate twisted-baffles and twisted cross-baffles, **Nanana, et al., 2017**, inserting multiple widths of cross hollow twisted tape **He, et al., 2018,** and inserting quadruple elements of twisted tapes in regular space, **Samruaisin**, **et al., 2018**.

The main aim of present work is to study the effect of inserting discrete twisted tapes in three positions for enhancing heat transfer in the tube heat exchanger.

2. APPARATUS DESCRIPTION

The experiments were performed by using the rig which is shown schematically in **Fig.1** and photographically in **Fig.2**. The experimental rig is divided into two parts: test tube section and other facilities. The dimensions of the test tube section which is made from aluminum are (L =1350 mm, r_i = 45 mm and t= 5 mm). The boundary condition of heat flux (50 W/m^2) around the test tube section was achieved by using an electrical wire with the help of clamp meter and variac transformer. Eighteen thermocouples type K were fixed along the tube to measure the temperature distribution along the tube wall, and the position of distributing these thermocouples is shown in **Table 1**. Moreover, two thermocouples were used at the inlet as well as two at the outlet of the test tube in order to measure the bulk air temperature. All thermocouples were connected to a selector switch that is connected to a digital thermometer. The outer surface of the test tube was insulated by rubber and gypsum layers. These layers were used to reduce the heat losses as possible. Two pieces of Teflon material were fixed at the ends of the tube section in order to reduce the losses from ends, as well as to connect manometer to measure pressure drop along the test tube section. The discrete twisted tapes which were used as augmentations are shown with all details in **Fig. 3**. Five pieces of these tapes were used in different positions (vertical $\alpha=90^0$, inclined $\alpha=45^0$ and horizontal $\alpha=0^0$), while the distance between pieces were kept constant (PR= 0.44) as shown in **Fig.4a** and **Fig.4b**. The facilities which were used in performing experiments were mentioned in **Table 2** with their purpose.

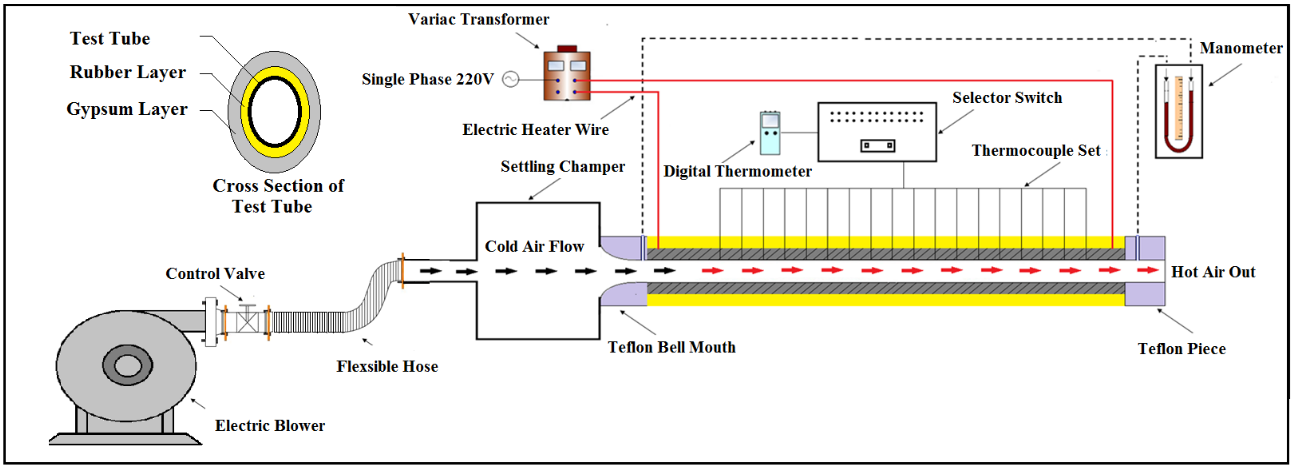

Figure 1. Schematic diagram of the using rig.

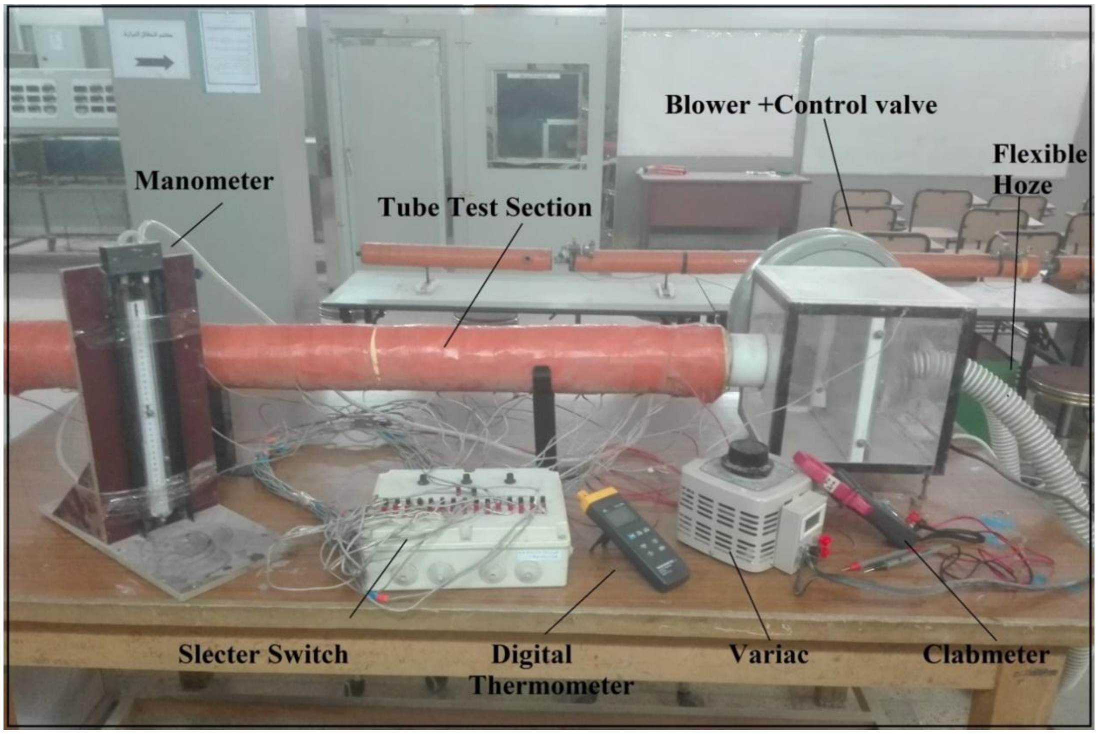

Figure 2. Photo of the used rig.

Table 1. Positions of Thermocouples**.**

No. of Thermocouple	Position (cm)	No. of Thermocouple	Position (cm)
1	1	10	29
2	2	11	38
3	3	12	47
4	5	13	57
5	7	14	67
6	9	15	77
7	13	16	90
8	17	17	104
9	23	18	118

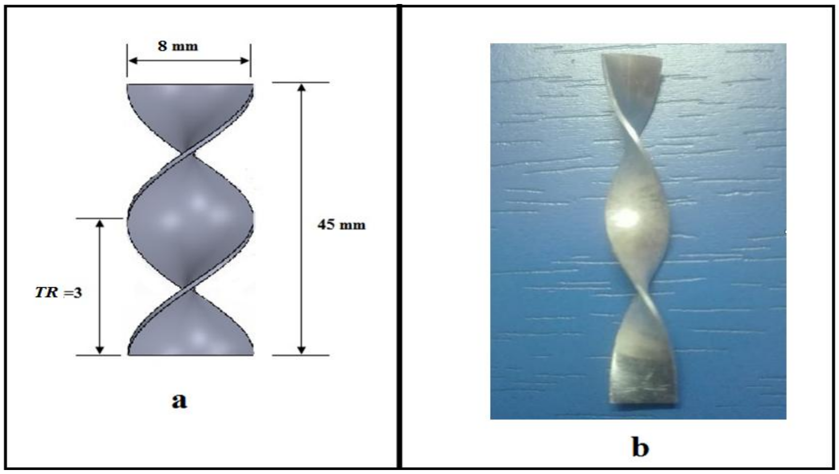

Figure 3. Schematic diagram and photo of discrete twisted tape.

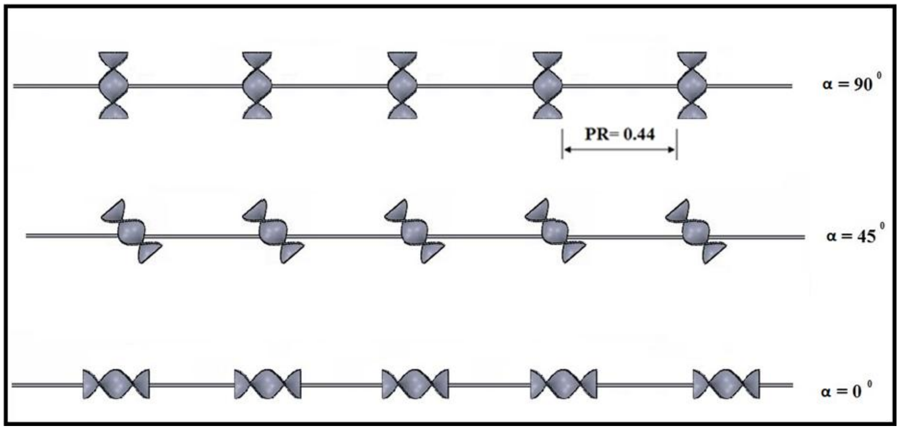

Figure 4a. Schematic diagram of distributing discrete twisted tapes.

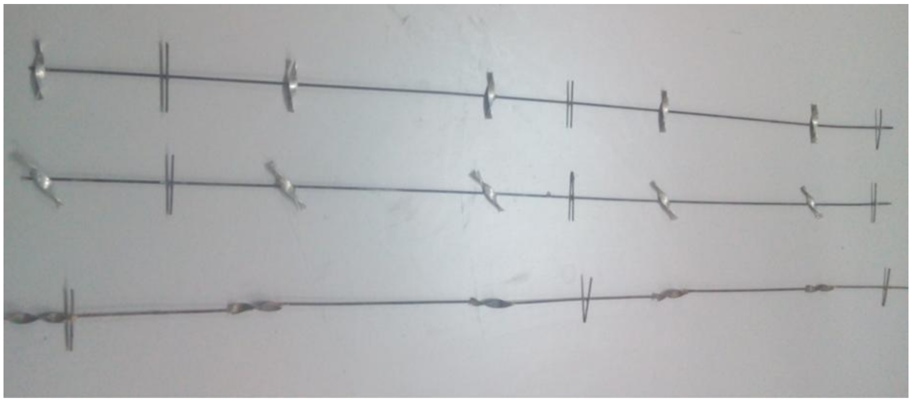

Figure 4b. Photo of distributing discrete twisted tapes.

Table 2. Facilities types and their purpose.

Facility type	Purpose of using
Electrical Blower	Providing airflow through the test section
Control Valve	Specifying an air flow rate
Variac Transformer with Clamp meter	Specifying suitable heat flux value
Manometer	Specifying the pressure drop
Selector Switch & Digital Thermometer	Measuring temperature distribution along the test section and other temperatures.

3. EXPERIMENTAL PROCEDURE

First of all, the electric blower is switched on and air flow is adjusted by the control valve. After that, the variac is adjusted at (100 volts) to provide the required constant heat flux (50 W/m^2). All the initial values of air velocity, pressure drop and temperatures of the system are measured. Then, the system is left to work for about 3 hours to achieve a steady state condition. During this time, the temperature distribution is measured every 15 minutes for checking. Finally, after reaching the steady state condition, the values of air velocity, pressure drop and temperatures of the system are measured again. The above procedure is repeated with three positions of inserting turbulators (vertical $\alpha=90^0$, inclined $\alpha=45^0$ and horizontal $\alpha=0^0$) and six values of Reynolds number (5000, 7000, 9000, 11000, 13000 and 15000).

4. ERROR ANALYSIS

The following equations were used to calculate the uncertainties of Nusselt number as well as friction factor to find the reliability of facilities which were used in experiments **Kline, 1953**.

$$\left(\frac{E_{Nu}}{Nu}\right)^2 = \left[\left(\frac{E_I}{I}\right)^2 + \left(\frac{E_v}{V}\right)^2 + \left(\frac{E_D}{D}\right)^2 + \left(\frac{E_{A_s}}{A_s}\right)^2 + \left(\frac{E_{\Delta T_s}}{\Delta T_s}\right)^2 + \left(\frac{E_{\Delta T_{oi}}}{\Delta T_{oi}}\right)^2\right] \quad (1)$$

E_{Nu} = 0.0372.

$$\left(\frac{E_f}{f}\right)^2 = \left[\left(\frac{E_{\Delta p}}{\Delta p}\right)^2 + \left(\frac{E_D}{D}\right)^2 + \left(\frac{E_{Re}}{Re}\right)^2\right] \tag{2}$$

E_f = 0.34.

5. DATA ANALYSIS

The following equations have been selected from, **Hasan, 2014 and Promvonge and Eiamsa-ard, 2007** for analyzing the experimental data. Also, a MATLAB program was used to do the calculations. The properties of working fluid were calculated depending on the bulk mean temperature:

Eq. (3) is used for calculating friction factor (f):

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\frac{\rho U^2}{2}\right)} \tag{3}$$

Reynolds number is calculated as follows:

$$Re = \frac{U*D}{\nu_a} \tag{4}$$

The actual heat input to the test section was determined as follows:

$$Q = \frac{Q_{net} + Q_{conv.}}{2} \tag{5}$$

where:

$$Q\ net = Input\ Voltage\ (V)\ x\ Input\ Current\ (I) - Q\ losses \tag{6}$$

Heat losses are calculated as follows:

$$Q_{losses} = \frac{\Delta T_L}{R_{th}} \tag{7}$$

where ΔT is the difference between the outer and inner lagging surface temperatures, and R_{th} is thermal resistance of insulation. And

$$Q_{conv} = m.Cp\ (T_{b,out} - T_{b,in}) \tag{8}$$

Also, the convection heat transfer from the test section can be written as follows:

$$Q = \bar{h}A_s(\bar{T}_{wall} - T_{bulk}) \tag{9}$$

Therefore, the heat transfer coefficient can be written as the following expression:

$$\bar{h} = \frac{Q}{A_s(\bar{T}_{wall} - T_{bulk})} \tag{10}$$

where: A_s equal to πDL , and T_{bulk} can be calculated from the following equation:

$$T_{bulk} = \left(T_{b,out} + T_{b,in}\right)/2 \tag{11}$$

The equation of average Nusselt number ($\overline{Nu}_d$) can be written as follows:

$$\overline{Nu}_d = \frac{\bar{h}.D}{K_a} \tag{12}$$

The thermal performance after inserting turbulators can be calculated from the following equation, **Promvonge, and Eiamsa-ard, 2007**:

$$Thermal\ Performance = \frac{\frac{\overline{Nu}_t}{\overline{Nu}_p}}{\left(\frac{f_t}{f_p}\right)^{\frac{1}{3}}} \tag{13}$$

6. RESULTS AND DISCUSSION

A comparison between the present experimental work and (Dittus–Boelter, Blasius) correlations **Incropera, et al., 2006** was performed for verification. This comparison was displayed in **Figs.5** and **6**, and it can be seen from these figures that the deviation value did not exceed 4.76 % for Nusselt number case and 3.7 % for friction factor case. Therefore, it can be demonstrated that the present experimental work is passable.

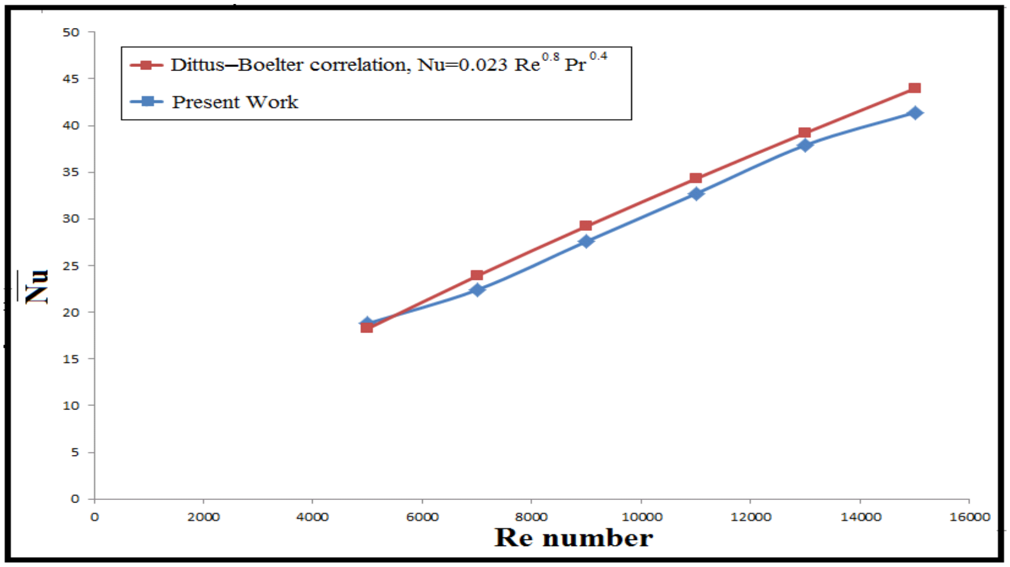

Figure 5. Validation of present work (Nu).

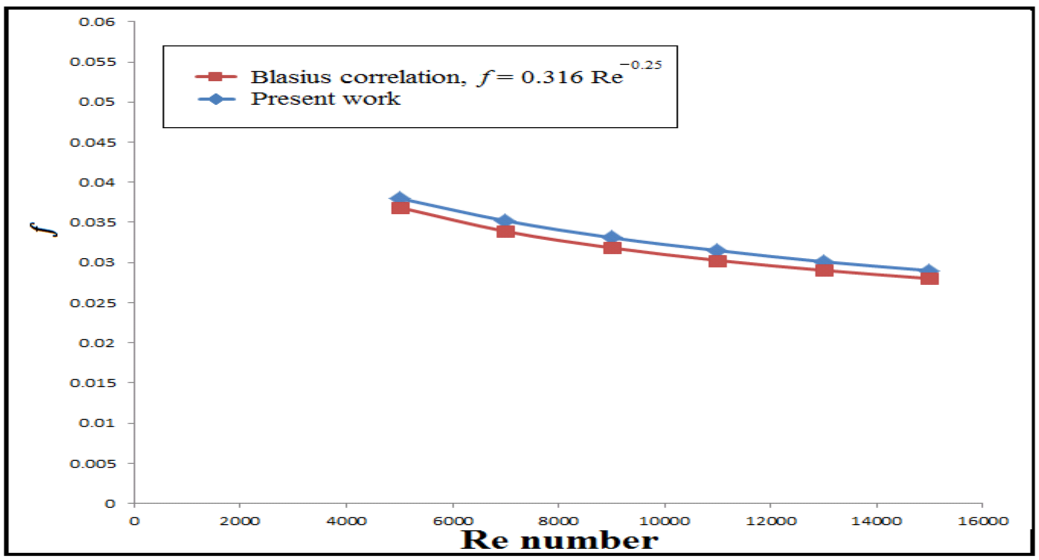

Figure 6. Validation of present work (*f*).

The relationship between ($\overline{Nu}$) & (Re) for the three different positions was displayed in **Fig.7**. It can be seen from this figure that the average Nusselt number values rise by inserting turbulators by (82.4 % for vertical position, 103.7 % for inclined position and 60.7 % for horizontal position) with respect to plain tube case. This behavior can be attributed to growing turbulence flow in tube due to inserting turbulators leading to minimizing the thermal boundary thickness at the wall, hence, enhancing convection **Chokphoemphun, et al., 2015**. Moreover, inserting turbulators in inclined position offers maximum thermal improving compared with other positions, and this result can be attributed to that the inclined position directs the flow towards the tube wall making the intensity of turbulence flow stronger, thus, improving heat transfer better.

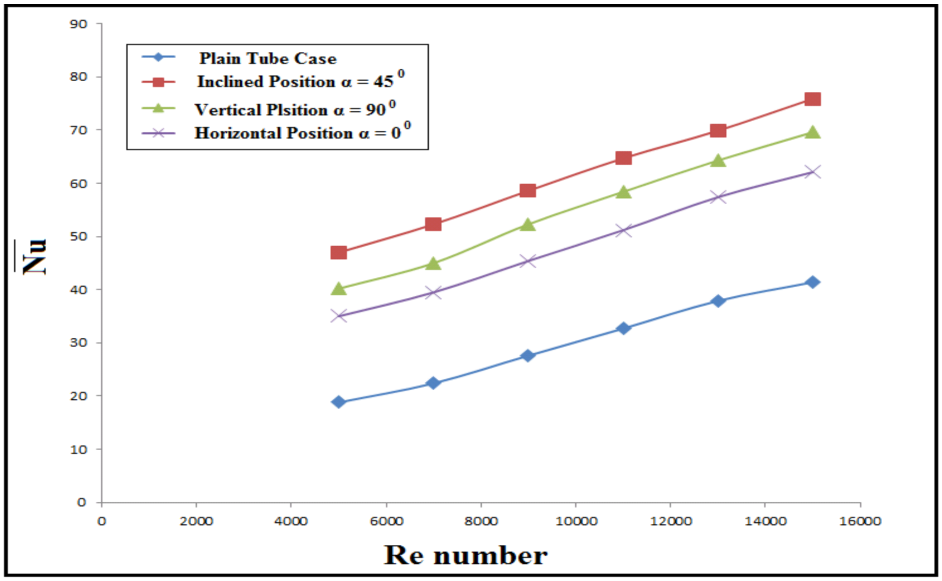

Figure 7. The relationship between Nu and Re for different positions.

Fig.8 shows the variation of friction factor with Reynolds number values. It can be observed that the friction factor has an inverse relationship with Re values due to the inverse relationship between the square velocity flow and friction factor as in Eq. (1) **Hussein, 2017**. Also, it can be seen that inserting turbulators in vertical, inclined or horizontal position raises the friction factor value by 76.8%, 70.5% and 60.1% with respect to the plain tube case. The reason behind this rising is increasing surface area as well as turbulence intensity due to inserting turbulators **Tamna, et al., 2016**.

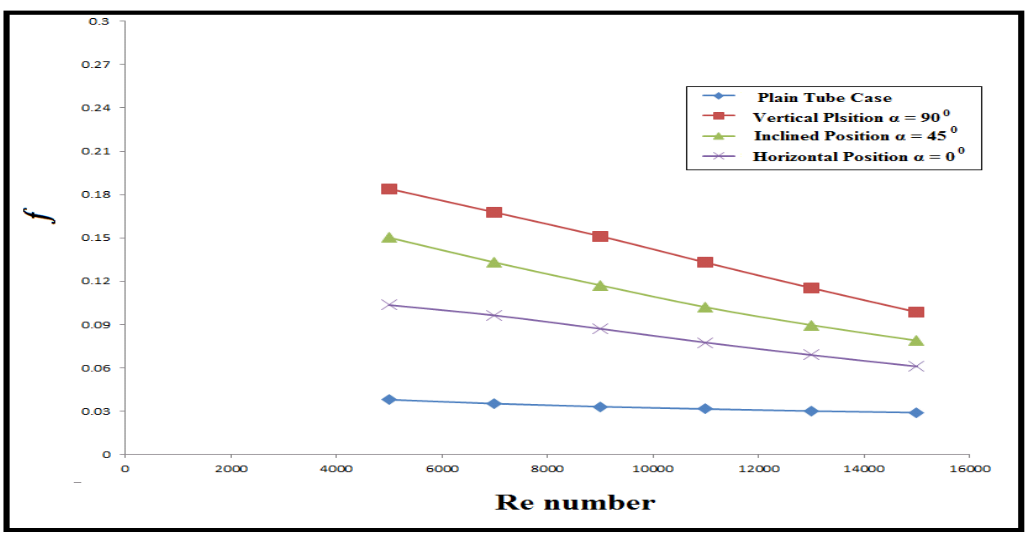

Figure 8. The relationship between *f* & Re for different positions.

The theoretical correlations that connect both Nusselt number and friction factor with Reynolds number (5000- 15000) for the plain case, vertical position, inclined position, and horizontal position were developed and listed in **Table 3**.

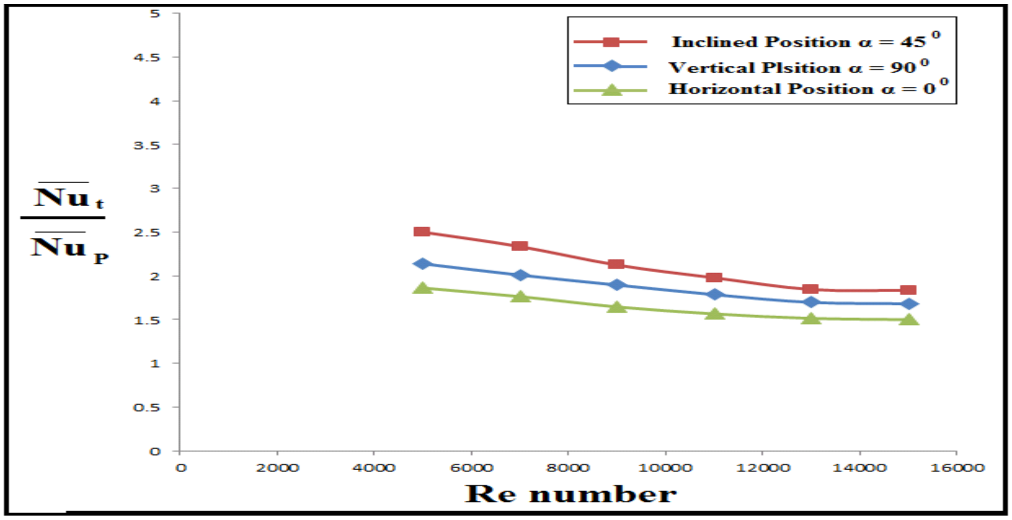

Figure 9. $\overline{Nu}_t / \overline{Nu}_p$ against Re of the three positions.

Fig.9 shows a comparison between the three cases ($\alpha=90^0$, $\alpha=45^0$ and $\alpha=0^0$) in term $\overline{Nu}_t / \overline{Nu}_p$ against Re. It can be seen from this figure that the inclined position offers maximum enhancement than the other cases.

Table 3. Theoretical correlations for present work.

No.	Case	Correlations of (Nu)	Correlations of (*f*)
1	Plain tube	$\overline{Nu} = 0.036.Re^{0.746}.Pr^{0.4}$	$f = 0.311.Re^{-0.246}$
2	Vertical position $\alpha=90^0$	$\overline{Nu} = 0.565.Re^{0.513}.\ Pr^{0.4}$	$f = 22.304.Re^{-0.556}$
3	Inclined position $\alpha=45^0$	$\overline{Nu} = 0.582.Re^{0.521}.\ Pr^{0.4}$	$f = 23.09.Re^{-0.586}$
4	Horizontal position $\alpha=0^0$	$\overline{Nu} = 0.406.Re^{0.537}.\ Pr^{0.4}$	$f = 6.512.Re^{-0.48}$

Fig.10 shows a comparison of thermal performance between the three cases of positions. It is worth mentioning that thermal performance is a coefficient which evaluates if the turbulators are realistically applicable or not. It can be observed that all position cases have thermal performance more than unity. Thus it can be used these turbulators as an improver for heat transfer rate. Moreover, the inclined position shows the maximum thermal performance by (1.4).

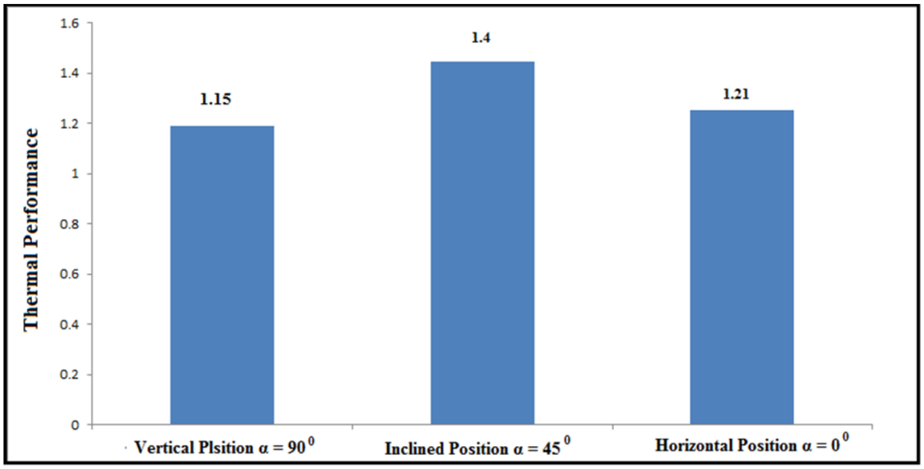

Figure 10. Thermal performance of the three positions.

7. CONCLUSIONS

Depending on the outcomes of the present work, the following points can be concluded:

1- The heat transfer coefficient defined by ($\overline{Nu}$) increases considerably by inserting turbulators by about 60.7 -103.7 % comparing with the plain tube case.
2- Inserting discrete twisted tapes by the inclined position ($\alpha = 45^0$) offers maximum enhancing in heat transfer rate by about 103.7% with respect to the plain case.
3- Thermal performance of all positions is more than unity. Thus, inserting discrete twisted tapes in these positions can be demonstrated as a useful way to enhance the heat transfer rate.
4- Maximum thermal performance (1.4) can be achieved by inserting turbulators in an inclined position ($\alpha=45^0$).

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9. NOMENCLATURE

A_s = inner Surface Area of tube Test Section, m^2.
C_p = specific Heat of working fluid, J/kg.K.
D = inner Diameter of Tube, m.
f = friction Factor, dimensionless.
f_p = friction Factor of Plain tube Case, dimensionless.
f_t = friction Factor of using Turbulators Case, dimensionless.
h = coefficient of Heat Transfer, W/m^2.K.
I = input Current, A.
K_a= thermal Conductivity, W/m.K.
L = length of Test Section, m.
$\dot{m}$ = mass Flow Rate, kg/s.
$\overline{Nu}$ = average Nusselt Number, dimensionless.
$\overline{Nu}_p$ = nusselt Number of Plain Tube Case, dimensionless.
$\overline{Nu}_t$ = nusselt Number of using turbulators case, dimensionless.
Pr = prandtl Number, dimensionless.
PR = pitch Ratio, dimensionless.
Q = actual Heat Input to the Test Section, W.
Q_{net} = heat Supplied, W.
$Q_{conv.}$= convection Heat Transfer from the Test tube Section, W.
Re =reynolds Number, dimensionless.
R_{th}= thermal resistance of insulation.
$T_{b,in}$ = temperature of working fluid at the Entrance of Test Section, ^{0}K.
$T_{b,out}$ = temperature of working fluid at the Exit of Test Section, ^{0}K.
T_{bulk} = bulk Temperature, ^{0}K.
$\bar{T}_{wall}$ = average Surface Temperature of tube test Section, ^{0}K.
TR = twisted Ratio, dimensionless.
U = velocity of air flow, m/s.
V = input Voltage, Volt.
ΔP = pressure Drop, Pa.
ΔT = difference between the outer and inner lagging surface temperatures.
ρ = density, kg/m^3.
α = angle of inclination, Degree.
ν_a= kinematic Viscosity.