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Experimental Study on Enhancement of Laminar Convective Heat Transfer Using twisted tape inserts

Engineering

KEYWORDS : Laminar flow , Twisted tape , Heat Transfer Enhancement , Passive techniques , Inserts .

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ABSTRACT

The enhancement of laminar convective heat transfer inside pipes by means of twisted tape of many shapes as an augmented device is experimentally studied and its performance is compared with smooth pipes under constant heat flux conditions .

For this purpose , an aluminum pipe (0.048) m internal diameter, (0.054) m external diameter (3) mm thickness and one meter long is used with metal twisted tapes (0.7) mm thickness manufactured from aluminum material , the length of twisted tape is equal to the length of the test section .

The range of Reynolds number used is (690 – 2195) and the range of Prandtle number used is (2.9 – 3.55) besides using four levels of heating , where the range of average heat flow rate provided on the tube surface is (150 – 650) W . Five levels of discharge has been used for every level of heating used , where the range of discharge is (1.538×10-5 - 4×10-5) m3/sec and the water has been used as a medium fluid for heat transfer .

Metal twisted tapes that used in experiments have width changing with change configuration of twisted tape at every time . The twisted tapes that are used have (1.5 , 3 , 4.5) Cm width , respectively at constant twist ratio which is equal to (2.77) . Every twisted tape from the used tapes is used as twisted tape time and twisted tape with triangular notches distributed as uniform shapes (aligned shapes) on the tape sides another time and twisted tape with triangular notches and holes spread out in the middle for the tape by constant distances between each hole and the hole that it follows third time , where the test is to be in operation for each tape from these tapes for all heating and discharge levels that are mentioned above .

The performance of pipe with twisted tapes should have assessment by depending on the calculation of the pumping power and overall enhancement ratio . The experimental results show that Nusselt number is directly proportional with the pumping power and overall enhancement ratio , where as the twisted tape has (4.5) Cm width with triangular notches and holes which explain the higher pumping power and the overall enhancement ratio for all levels of heating . The overall enhancement ratio was (575%) for the heat flow rate (150)W , (830%) for the heat flow rate (260) W, (217%) for the heat flow rate (400) W , (400%) for the heat flow rate (650)W , respectively compared with a plain tube without any tape at Reynolds number which is equal to (2195) .

1. Introduction :

In recent years, energy and material saving consideration have increased the efforts which aim at producing more efficient heat exchanger equipment through the augmentation of heat transfer. The potentials of heat transfer in engineering applications are high due to a large number of industrial applications for heat and mass transfer phenomena as in chemical, petrochemical, biomedical, food processing, heating and cooling in evaporators, thermal power plant, air – conditioning equipment, refrigerators, radiators in space vehicle, automobiles, etc. These activities involve multi – million dollar investment annually for both operation and capital cost . In early days only plain tubes were used in the shell and tube heat exchanger or fire tube boilers. The increase of heat exchanger efficiency by augmentation or enhanced heat transfer may result in considerable saving in the material required. The use of enhanced surfaces allows the designer to increase the heat duty for a given exchanger, usually with pressure drop penalty, or to reduce the size of heat exchanger for a given heat duty. Variety of different techniques employed for heat transfer process are generally referred to "Enhancement" [1] .

2. Enhancement techniques :

Heat transfer enhancement techniques can be classified into two broad categories: "passive" and "active" techniques. In active techniques some external power needed for the enhancement process. Furthermore, external power is not easy to provide in several applications. Some examples of active techniques are heated surface rotation, surface vibration, fluid vibration, electrostatic fields and suction at the heated surface. This technique involves the need of external power in addition to the complexity in design and difficulty in manufacturing .

Passive techniques (as it used in this paper) do not need any external power input and the additional power needed to enhance heat transfer is taken from the available power in the system, which ultimately leads to a fluid pressure drop. The heat exchanger design has been striving for improving thermal contact (enhance heat transfer coefficient) and reducing the pumping power in order to improve the thermo hydraulic efficiency of the heat exchanger. A good heat exchanger design should have an efficient thermo hydraulic performance. It is impossible to stop energy loss completely but it can be minimized through an efficient design. This technique involves a problem of manufacturing cost as in fin tubes or ribbed tubes, or cost of adding material as in twisted tape or helical wire. Among those augmentation techniques, researchers show in the present time more interest in large-scale surface roughness in the form of ribs or enhancement devices (twisted tape and helical wire) . These orientations are called turbulence promoters. These promoters at the wall surface help to destroy a very small zone at the boundaries where the flow is essentially viscous and the influence of turbulent flow does not reach at all that so called the viscous sub layer.

Consequently passive techniques are often preferred and they have seen their wider application because it's relatively low in cost, easy to insert and easy to take off for cleaning operation.

For **laminar flow,** the dominant thermal resistance is limited to a thicker region compared with a turbulent flow. Thus a wire coil insert is not effective in a laminar flow because it cannot mix the bulk flow well and reverse is true. Hence a twisted tap is generally preferred in laminar flow. Performance and cost are the two major factors that play an important role in the selection of any passive technique for heat transfer enhancement. Generally, helical wire and twisted tape are more widely applied and have been preferred in the recent past to other methods such as the extended surfaces which suffer from a relatively high cost.

3. Twisted tape in circular tube :

Many studies were conducted previously to analyze heat transfer and pressure drop with twisted tape as swirl generators in a circular tubes .

Saha et al. [2,3,4] reported experimental data on a twisted tape generated laminar swirl flow friction factor and Nusselt number for a large prandtle number (205 < Pr < 518) and observed that , on the basis of a constant pumping power , short-length twisted tape is a good choice because in this case swirl generated by the twisted tape decays slowly downstream which increases the heat transfer coefficient with minimum pressure drop, as compared with a full-length twisted tape . Regularly spaced twisted tape decreases the friction factor and reduces the heat transfer coefficient because the spacing of the twisted tape disturbs the swirl flow .Hong and Bergles [5] reported heat transfer enhancement in laminar , viscous liquid flows in a tube with uniform heat flux boundary conditions , but their correlation has limited applicability as it is valid for a high Prandtl number (approximately 730) . The circumferential temperature profile for swirl flow is related to tape orientation. Tariq et al . [6] found that in a laminar flow the introduction of turbulent promoters , such as an internally threaded tube , is not efficient compared with a twisted tape insert on the basis of the overall efficiency. Manglik and Bergles [7] depended on the main parameters such as the flow rate and tape geometry , the enhancement in heat transfer is due to the tube partitioning and flow blockage , the large flow path and secondary fluid circulation . They are developed laminar correlations for friction factor and Nusselt number , including swirl parameter, which defines the interaction between viscous , convective inertia and centrifugal forces . These correlations pertain to the constant wall temperature case for fully developed flow , based on both pervious data and their own experimental data . The heat transfer correlation as proposed by them is depend upon prandtle number , tube diameter and thickness of twisted insert .

A similar correlation for friction factor was developed by Manglik and Beregles [9]

Lokanath and Misal [8] studied the performance of a plate heat exchanger and augmented shell and tube heat exchanger for different fluids. They found that twisted tapes of tighter twists are expected to give higher overall heat transfer coefficients in the augmented shell and tube heat exchanger. Saha et al [9] found that pinching (placing of a twisted tape exactly at the center of the tube) of twisted tape in a tube performs better than a twisted tape inserted by a loose fit . They further showed that a non-zero phase angle in-between the segmented twisted tape gives poor results because the swirl will break easily in-between the two segmented twisted tapes. Furthermore , reduction in the width of the twisted tape is not effective , compared with a twisted tape of width equal to the inside diameter of the tube. AI-Fahed et al.[10] observed that , for a low twist ratio (Y=5.4) and high pressure drop , a loose fit is recommended for design of the heat exchanger , since it is easier to install and to remove for cleaning purposes . Other than this twist ratio , a tight-fit twisted tape provides better performance that a loose-fit twisted tape. Liao and Xin [11] reported experimental data on the compound heat transfer enhancement technique and concluded that the enhancement of heat transfer in a tube with three-dimensional internal extended surfaces by replacing continuous twisted tape with almost segmented twisted tape inserts results in a decrease in the friction factor but with a comparatively small decrease in the Stanton number (St) .

The Stanton number is defined as the ratio of heat transfer rate to the enthalpy difference and is a measure of the heat transfer coefficient . Suresh Kumar et al .[12,13] investigated the thermo hydraulic performance of twisted tape inserts in a large hydraulic diameter annulus .

The thermo hydraulic performance in laminar flow with a twisted tape is better than the wire coil for the same helix angle and thickness ratio . This is probably due to the fact that , in laminar flow , the dominant thermal resistance is not limited in a thin wall region but extends over the entire cross-section. Thus, a twisted tape insert mixes the bulk flow and is probably effective . Considering the overall enhancement ratio , twisted tape is effective for small Prandtl number fluids and wire coil is effective for high Prandtl number fluids . Saha and chakraborty [14] observed laminar flow heat transfer and pressure characteristics in a circular tube fitted with regularly spaced twisted tape and concluded that there is a drastic reduction in the pressure drop , more than the corresponding reduction in heat transfer. Thus, it appears that , on the basis of a constant pumping power , a large number of turns may yield improved thermo hydraulic performance compared with a single turn on twisted tape.

Lokanath [15] represented experimental data on laminar flow of water through a horizontal tube under uniform heat flux condition and fitted with half-length twisted tape. He found that , on the basis of unit pressure drop and unit pumping power , half-length tapes are more effective than full-length tapes. Marner and Bergles [16] were the first investigators to recognize the importance of uniform wall temperature (UWT) boundary condition to a major group of heat exchanger used chemical industry . They studied UWT heating and cooling of ethylene glycol (Pr = 24-85 , Re= 380-3470) using single twisted tape insert of Y = 5.4 in a tube and internally finned tubes , and observed that both heat transfer and friction factor increased substantially beyond particular Reynolds numbers , at which secondary swirl flow and turbulence were induced in the flowing fluid. Agarwal and Raja Rao [17] reported experimental investigations of isothermal and non isothermal friction factor and mean Nusselt number for uniform wall temperature (UWT) heating and cooling of servo herm oil (Pr = 195-375) in a circular tube (Re= 70 - 4000) with twisted tape inserts (Y=2.41-4.84) . Isothermal friction factor wasfound to be 3.13-9.71 times the plain tube values.

The Nusselt number was found to be (2.28-5.35) and (1.21-3.7) times the plain tube forced convection values based on constant flow rate and constant pumping power respectively . They proposed correlation representing effect of heat transfer on friction factor for practical application. Yadav [18] studied influences of the half length twisted tape insertion on heat transfer and pressure drop characteristics in a U-bend double pipe heat exchanger using oil as working fluid . The experimental results revealed that the increase in heat transfer rate of the twisted tape inserts was found to be strongly influenced by tape – induced swirl .

The heat transfer coefficient is found to increase by (40%) with half –length twisted tape inserts when compared with plain heat exchanger. On the basis of equal mass flow rate the heat transfer performance of half length twisted tape is better than the plain heat exchanger , and on the basis of unit pressure drop the heat transfer performance of smooth tube is better than half-length twisted tape. It was also observed that the thermal performance of plain heat exchanger was found better than half length twisted tape by 1.3-1.5 times.

4 . Experimental Work :

- **General Description of the Rig :**

A schematic diagram of the experimental equipment used in this work is shown diagrammatically in Fig. (1) .

The test section consists of aluminum tube , the aluminum is used for the insert assembly . Its dimensions are (48 mm) inside diameter , (54 mm) outer diameter and a length of (1150 mm) packed by twisted tapes with (1.5 , 3 , 4.5 Cm) widths respectively .

The aluminum tube , is considered as a test section heated by using an electrical heater wound uniformly along it to obtain the constant heat flux condition .

To measure the pressure drop up the twisted tape inserts , two pressure taps are placed , the first tap is placed on distance (7.5 Cm) from the entrance of the test section and the second tap is placed on distance (100 Cm) from the first tap.

The tube surface temperature is measured by (20) copper constantan Type (T) thermocouples , fixed longitudinally and circumferentially . The inlet bulk water temperature is measured by one thermocouple placed in the entrance section ; also one thermocouple measured the outlet bulk water temperature located at the exit section .

To reduce heat losses from the test section to the surrounding , Teflon connection pieces present the test section entrance and exit to reduce the end losses , also the outside of test section is thermally insulated. The test section is standing on the wood base at the Teflon region .

7. Twisted Tapes Inserts :
A schematic of twisted tape insert is shown in **Fig.(2)** .The inserts are metal twisted tapes manufactured from aluminum material (1 mm) thickness , the twisted tapes have the same twist ratio which equal (2.77).

The tapes used in the experimental works have different widths , where the tapes widths are (1.5, 3 , 4.5 Cm) respectively . Eeach tape has one twist length equal (15 Cm) . Each tape is used as a normal twisted tape , twisted tape with notches another and as a twisted tape with notches and holes another time .

The notches were on the isosceles triangular form , where the distance between two notches are (7.5 Cm) and the distance between two holes is (3.75 Cm), (5 Cm) hole diameter is used .

5. The Measurements Procedure :
During each test run, the following readings are recorded :

1. The local wall temperature is read from the outputs of the 20 thermocouples on the test tube .
2. The inlet and outlet water bulk temperature is read one thermocouple on the entrance and the exit section respectively .
3. The air surrounding temperature by one thermocouple T type is connected in selector switch .
4. The new height of the mercury of two taps .
5. Water volumetric flow rate through the insert, which measured by using a scaled cylinder and stopwatch .
6. The heater current in ampere .
7. The heater voltage in volts .

The channel is emptied after each experiment by closing the globe valve at the entrance section and opening the gate valve in the draining connection . The consideration is given to observe the presence of any bubbles and water vapor in the test section during the experimental runs .

6. Heat Transfer Analysis :
A simplified step is used to analyze the heat transfer process for water flow in inserted column . The outer surface of the aluminum tube was subjected to a uniform heat flux while the outer surface of the column was subjected to an ambient temperature .

The average heat flux to the inserted tube is obtained by the measurement of the average increase in bulk temperature of the water across the test section . The average increase in bulk temperature may be found by measuring the bulk temperature at the entrance and exit of the inserted tube by the thermocouples that are found in entrance and exit of the test section .

$$Q=m^{\cdot}(Cp_oT_o-Cp_iT_i) \quad \textbf{(1)}$$

Where the (T_i) and (T_o) are the water bulk temperature at the entrance and exit respectively .

The total input power supplied to cylinder can be calculated :

$$Q=\mathbf{V.I.COS}(\beta) \quad \textbf{(2)}$$

The heat balance between the heat flux supplied and heat flux calculated from Eq. (1) illustrated that the difference between them did not exceed (5%). This was due to the good insulation and the high thermal conductivity of aluminum .

The wall heat flux can be calculated by :

$$q_w=\frac{Q}{A_s} \quad \textbf{(3)}$$

Where :

$$A_s=2\pi r_i L \quad \textbf{(4)}$$

The local heat transfer coefficient can be obtained by :

$$h_x=\frac{q_w}{(\Delta T)_x} \quad \textbf{(5)}$$

Where $(\Delta T)_x$ is the temperature difference between local surface temperature $(T_s)_x$ and local bulk water temperature $(T_b)_x$ at length (x) from the test tube entrance .

The local bulk water temperature $(T_b)_x$ at each measuring level was calculated by assuming a linear water temperature variation a long the flow channel [19] :

$$(T_b)_x=T_i+\frac{x}{l}(T_o-T_i) \quad \textbf{(7)}$$

Then the local heat transfer coefficient can be obtained by :

$$h_x=\frac{q_w}{(T_s)_x-(T_b)_x} \quad \textbf{(8)}$$

The local Nusselt number Nu_x can be determined by :

$$Nu_x=\frac{h_x \cdot D_i}{k_f} \quad \textbf{(9)}$$

All the properties are evaluated at the mean film water temperature **[20]** :

$$(T_m)_x=\frac{(T_s)_x+(T_b)_x}{2} \quad \textbf{(10)}$$

The average heat transfer coefficient (h) is found by integration as follows :

$$h=\frac{1}{L}\int_{x=0}^{x=L} h_x.dx \quad \textbf{(11)}$$

The average values of Nusselt number (Nu) can be calculated as :

$$Nu=\frac{1}{L}\int_{x=0}^{x=L} Nu_x.dx \quad \textbf{(12)}$$

To calculate the integration above trapezoidal is used.

The water velocity at inlet of the test tube (V_i) can be calculated by [21] :

$$V_i = \frac{4m^{\cdot}}{\pi Di^2 \rho} \qquad (13)$$

Reynolds number might be calculated according to water velocity at inlet of the test tube and inner diameter of test tube .

$$R_e = \frac{V_i \ D_i}{v} \qquad (14)$$

The friction factor (F) is the parameter in the pressure drop calculation where it is calculated from Darcy equation [22] :

$$F = \frac{\Delta P}{\frac{L}{Di} \ \frac{\rho V_i^2}{2}} \qquad (15)$$

Not that the friction factor (F) is related to the pressure drop in the fluid.

The required pumping power (P.P) to overcome a specified pressure drop ΔP is determined from **[22]** :

$$P.P = V^{\cdot} \Delta P \qquad (16)$$

Where $V^{\cdot}$ is the volume flow rate of the water through the test section .

7. Results and Discussions :

1. Local Nusselt number (Nux):

The variation of the local Nusselt number (Nux) with the tube length is plotted in Figs.(3)

As shown in these figures , the local Nusselt number decreases with axial position along the flow direction and the decrease in the Reynolds number , also that the local Nusselt number increases with the increase of the heat flux .

The local Nusselt number begins with high value at the inlet entry length region due to low difference between the water and tube wall temperature and thin thermal boundary layer is formed , then progressively the difference between water and tube wall temperature increases and the thickness of the thermal boundary layer increase until reaching the radius of tube at the end of entry length region .

The figures mentioned above in this article revealed that at any level of heating , the local Nusselt number lower for smooth pipe because it affects the thermal boundary layer that starts by growing from the entrance edges by direction center of the pipe .

The local Nusselt number started by raising when the twisted tapes are inserted , where the maximum value of local Nusselt number occurrs at the twisted tapes that have widths (y=4.5) Cm with triangular notches and holes at all level of heating .

2. Average Nusselt number (Nu) :

Nusselt number is the measure of the convective heat transfer . There for, it is important to consider this parameter to evaluate the rate of heat transfer , Figs.(4) show the variation of heat transfer in term average Nusselt number with Reynolds number for the tube fitted with different configurations of twisted tapes for various widths of it at constant twist ratio for four level of heating and five level of discharge .

These figures also show that the heat transfer for the twisted insert is highly influences on the Reynolds number than that of the smooth pipe. These figures also show the effect of width and configurations of twisted tapes on the heat transfer rate , where it is noticed in these figures that the increasing in the Reynolds number it is complying increasing in the Nusselt number , also we noticed that the Nusselt number increase by increasing width of the twisted tapes and changing figures of these twisted tapes .

The larger increasing in Nusselt number was at the twisted tapes that have width (y=4.5) Cm with triangular notches and holes at a given Reynolds number at all levels of heating . This is because the twisted tape interrupts the development of the boundary layer of the fluid flow near the wall of the tube . Hence it increases the average temperature of the fluid in the radial direction due to the larger contact surface area the heat transfer rate increase . Also these inserts create the turbulence and whirling motion to the water which is flows inside the test section . The whirling makes the flow be highly turbulent which leads to improved convection heat transfer .

3. Friction Factor (F) :

Friction factor is a measure of the pressure losses in a system to the kinetic energy of the fluid [24] . In the present work, the pressure losses include losses due to friction and due to drag force exerted by obstacles. Fig. (5) shows the variation friction factor with Reynolds number for tube fitted with different twisted tapes . It is noticed that the increase in Reynolds number leads to decrease in the friction factor , because the friction factor is proportional with pressure drop and inversely proportional to the square root of flow speed . These figures also indicate that the larger width with triangular notches and holes for twisted tape causes higher – pressure drop because each increase in the width and changing the configuration of twisted tape means increase in the size of obstacles and hence the pressure drop also increases friction factor . Also , this increasing is attributed to the increase in surface area , increase in the disturbance in the main core flow and increase the disturbance in the laminar sub layer of the boundary layer of the flow . These figures have showed that the higher width of twisted tape has the higher – pressure drop. The higher width and different shapes of twisted tape provide a larger number of obstacles , which ultimately lead to higher – pressure drop , which causes a higher friction factor and vice versa .

4. Overall Enhancement Ratio (O.E.R) :

Overall enhancement ratio is defined as the ratio of heat transfer enhancement ratio to the friction factor ratio . This parameter is used to differentiate passive technique and a comparison of different configurations for the technique itself .

Figure (6) show the overall enhancement ratio against Reynolds number for all level of heating . Generally, the increase in Reynolds number leads to increase overall enhancement ratio (O.E.R) because increasing in Nusselt number and minimizing in the friction factor accompany with this increase in Reynolds number . It is observed that the increase in the width of twisted tape and changing the configurations of twisted tape for all these figures lead to increase in (O.E.R) for the pervious reasons related with the width and shape of insert . Also these figures indicate that the highest performance occrued at (y) = 4.5 Cm with triangular notches and holes . The increase of width leads to increase in Nusselt number and friction factor .

11.Conclusions: The main conclusions that are drawn from this experimental study can be outlined as follows :

1. For all experiments , the local Nusselt number increases as the heat flux increases .
2. The average Nusselt number increases as the Reynolds number increases for all inserts used in experiments .

3. The friction factor decreases with increase of Reynolds number
4. The fluid and heat transfer parameter have the same behavior at the experimental analysis for all inserts used and for all level of heating .
5. In general, the twisted tape insert enhances the heat transfer coefficient inside the tube for laminar flow in the tested range of Reynolds number .
6. The larger width (y=4.5 Cm) of twisted tape with triangular notches and holes has the highest Nusselt number as well as the highest friction factor .
7. The overall enhancement ratio show that the twisted tape width (y = 4.5 Cm) with triangular notches and holes the best performs as compared with (1.5 , 3) Cm twisted tape width with other configurations .

Symbols

Di	Inner diameter of the test section	m
F	Friction factor	-
h	Average heat transfer coefficient	$W/m^2.k$
h_x	Local heat transfer coefficient	$W/m^2.k$
I	The current	Amper
K_f	Thermal conductivity of the fluid	W/m.k
L	Length of the test section	m
Nu	Average Nusselt number	-
Nu_x	Local Nusselt number	-
P.P	Pumping Power	W
Pr	Prandtle number	-
q_w	The heat flux	W/m^2
Re	Reynolds number	-
r_i	Internal radius of the test section	m
T_b	The bulk temperature	°C
Ti	Inlet Temperature	°C
To	Outlet Temperature	°C
T_s	The surface Temperature	°C
t	thickness	mm
V	Electric voltage	Volt
V·	Volumetric flow rate	m^3/sec
Vi	Inlet velocity	m/sec
W	The mass flow rate of water	Kg/sec
x	The distance between two points on the surface test section	m
β	Difference phase angle between voltage and current	degree
ρ	Density of water	kg/m^3
μ	Dynamic viscosity	$N.s/m^2$
$μ_s$	Dynamic viscosity at the surface Temperature	$N.s/m^2$
ν	Kinematic viscosity	m^2/s

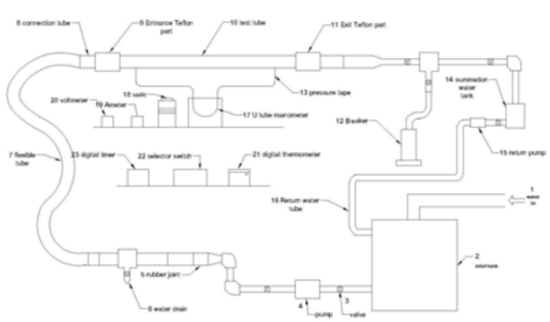


Figure (1) The rig that used in experiments.

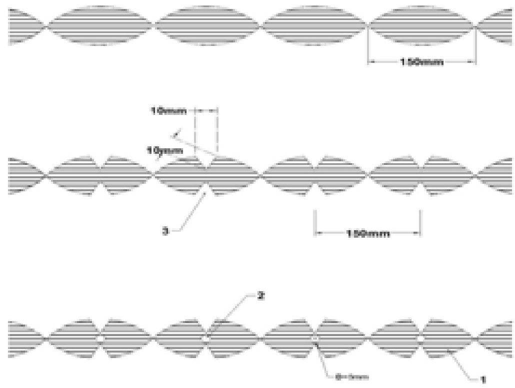


Figure (2) Configurations Twisted Tapes that used in experiments .

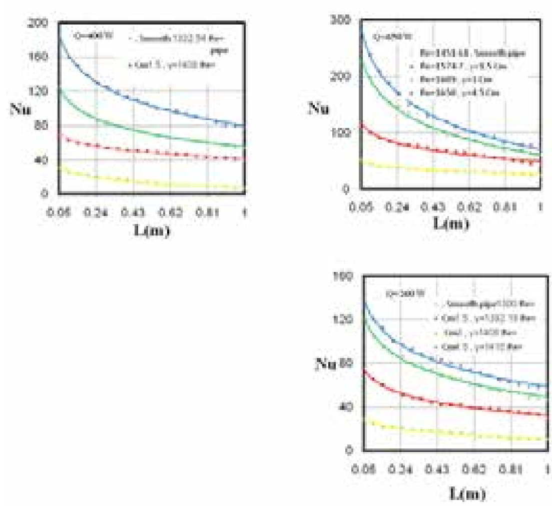
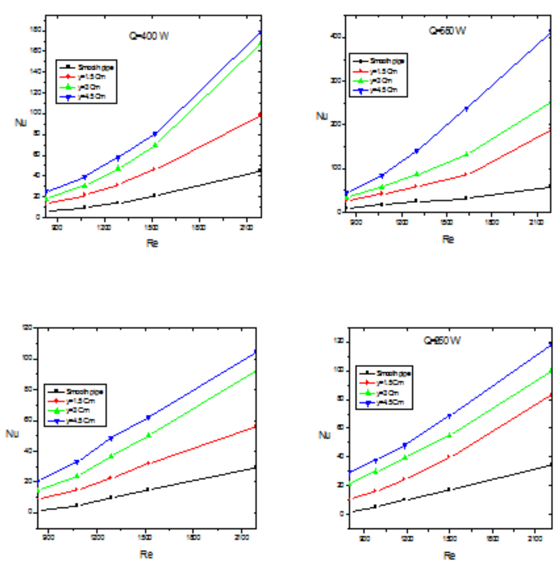


Figure (3) Variation Local Nusselt number with axial position for the test section inserted by twisted tape with triangular notches and holes .



Figure(4) Variation Reynolds number with Nusselt number for twisted tape with triangular notches and holes for all levels of heating.

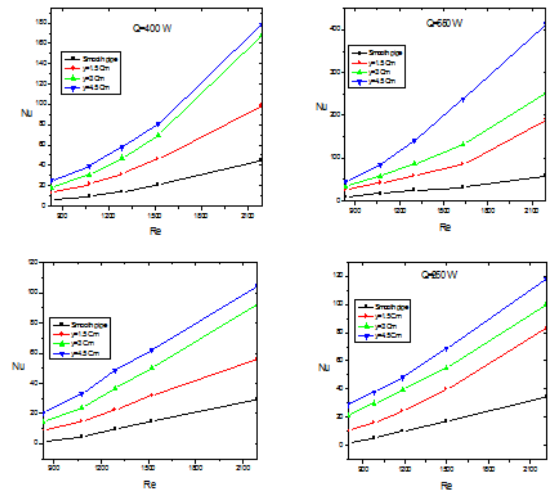


Figure (5) Variation friction factor with Reynolds number for twisted tape has triangular notches and holes .

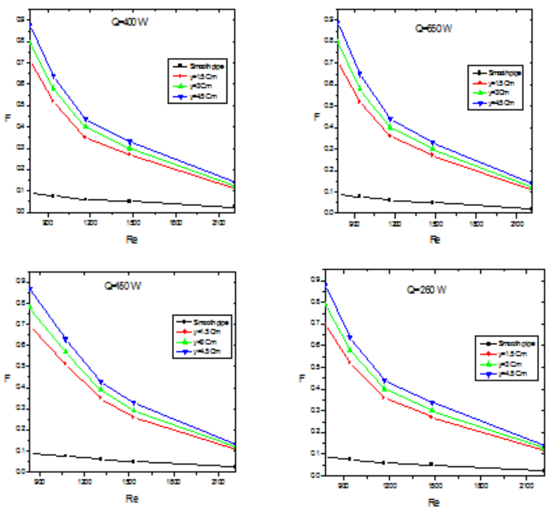
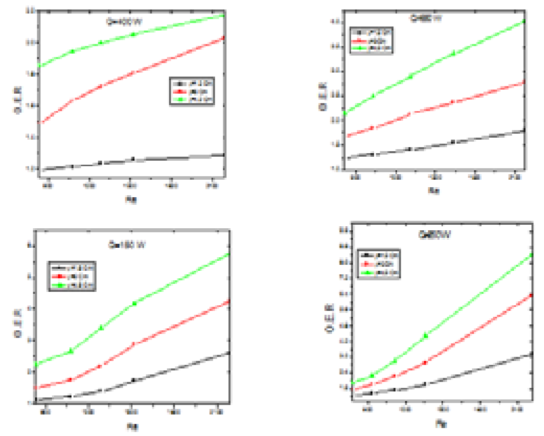


Figure (6) Variation O.E.R . with Reynolds number for twisted tape with triangular notches and holes .



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