Heat Transfer Enhancement in Circular Tube using Twisted Tape Inserts of Different Width Ratio under Constant Wall Heat flux Condition

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***Abstract*- The present study reports an experimental analysis on heat transfer, friction factor and thermal performance characteristics of turbulent flow (7500 ≤ Re ≥ 13,000) in heat exchanger tubes with twisted tape as swirl generators under constant wall heat flux condition. The experiments were carried out for twisted tape inserts at five different width ratios (W/D) of 0.35, 0.44, 0.53, 0.62 and 0.71 at constant twist ratio (H/D) of 2.5. The experimental results show that Nusselt number increases with increasing Reynolds number and increasing width ratio (W/D) of the swirl generators. Also the friction factor decreases with increasing Reynolds number and decreasing width ratio. Nusselt number increased in ranges of 6.04% to 17.26% compared to the results of the tube without twisted tape (the plain tube), depending on the operating conditions. At the same pumping power, the use of twisted tape inserts results in thermal performance factor up to 1.17 times of those of the plain tube. In addition, the correlations that developed from the present experimental data for Nusselt number.**

Keywords—Heat transfer coefficient, Twisted tape inserts, width ratio, Pressure drop and Thermal enhancement efficiency.

I. INTRODUCTION

Heat exchanger as equipment to facilitate the convective heat transfer of fluid inside tubes is frequently utilized in many industrial applications, such as chemical engineering process, heat recovery, air conditioning and refrigeration systems, power plant and radiators for automobiles. In general, heat transfer enhancement in heat exchangers can be divided into two methods. One is the active method requiring extra external power sources such as fluid vibration, injection and suction of the fluid, jet impingement and electrostatic fields. The other is the passive method that requires no other power source. The devices in this category are surface coating, rough surfaces, turbulent/swirl flow devices, extended surfaces etc.

Twisted tapes, as one of the passive heat transfer enhancement technology, have been extensively studied due to their advantages of steady performance, simple configuration and ease of installation. By generating swirls which enhance the fluid mixing of the near-wall and central regions, the heat transfer in tubes with twisted tapes could be enhanced [1-2]. Moreover, the twisted tape can partition and block the flow, reduce the hydraulic diameter, elongate the twisted flow path and generate a fin effect [1, 2]. All these lead to additional heat transfer improvements. However, the thermal improvements are accompanied by increased pressure drop. Optimization in the thermohydraulic performance of tubes fitted with twisted tapes has gained increasing attention .For instance, Saha et al. [3] experimentally studied the heat transfer and pressure drop characteristics of laminar flow in a circular tube fitted with regularly spaced twisted tape elements connected with rod. The results showed that the pressure drop of the tube fitted with the segmented twisted tape elements is 40% smaller than that of the tube fitted with a continuous twisted tape, and the former has a better thermohydraulic performance. Eiamsa-ard et al. [4] experimentally investigated the convective heat transfer behaviors in a circular tube fitted with regularly spaced twisted tape elements in laminar and turbulent flows, and they found that the heat transfer coefficient and friction factor were both significantly reduced as compared with those of the tube fitted with a continuous twisted tape. Later, Saha et al. [5] further investigated the effects of the width of tape elements and the diameter of connecting rod on heat transfer and pressure drop characteristics. This work indicated that a narrower width of tape elements led to a worse thermohydraulic performance, while a thinner connecting rod resulted in a better one. Therefore, he proposed to abolish the connecting rod and 'pinch' the tube to fix the segmented twisted tape elements. S. Eiamsa-ard, K. Wongcharee [6] studied the 3-D Numerical simulation of swirling flow and convective heat transfer in a circular tube induced by means of loose-fit twisted tapes. The effects of the clearance ratio defined as ratio of clearance between the edge of tape and tube wall to tube diameter (CR=c/D=0.0 (tight-fit), 0.1, 0.2 and 0.3) on heat transfer enhancement (Nu), friction factor (f) and thermal performance factor (η) are numerically investigated for twisted tapes at two different twist ratios (y/w=2.5 and 5.0).The mean flow patterns in a tube with loose-fit twisted tapes in terms of contour plots of velocity, pathline, pressure, temperature and turbulent kinetics energy (TKE) are presented and compared with those in a tube fitted with tight-fit twisted tapes. It is visible that the twisted tape inserts for y/w=2.5 with CR=0.0 (tight-fit), 0.1, 0.2 and 0.3 can enhance heat transfer rates up to 73.6%, 46.6%, 17.5% and 20%, respectively and increase friction factors up to 330%, 262%, 189%, and 160%, respectively, in comparison with those of the plain tube. The tube with loose-fit twisted tape inserts with CR=0.1, 0.2 and 0.3 provide heat transfer enhancement around 15.6%, 33.3%

and 31.6% lower than those with CR=0.0 (the tight-fit twisted tape).Later, Liu et al. [7] conducted a numerical investigation on heat transfer behaviors of laminar and turbulent flows in a circular tube fitted with short-width twisted tapes. Their results of laminar flow also showed a worse overall performance of this method.

Recently, some new types of twisted tapes were developed by some investigators [8-10]. Eiamsa-ard and Promvonge [8] developed a twisted tape with serrated edges. Their experiment demonstrated that the heat transfer rate and thermal performance factor in the tube with this type of twisted tape insert were about 1.04-1.27 and 1.02-1.12 times those in the tube with smooth twisted tape insert, respectively. Later, Rahimia et al. [9] studied the heat transfer and friction factor characteristics of the tube fitted with perforated, notched and jagged twisted tapes. The results revealed that only the jagged insert was better than the conventional twisted tape in the heat transfer coefficient and thermal performance factor. More recently, Chang et al. [10] invented a broken twisted tape insert which can induce a better fluid mixing. They reported that, as compared with those of the tube fitted with smooth twisted tape, the heat transfer coefficient, mean Fanning friction factor and thermal performance factor of the tube fitted with broken twisted tape were augmented up to 1.28-2.4, 2.0¬-4.7 and 0.99-1.8 times, respectively, in a Re range of 1000-40000.

Regarding reduced width twisted tape inserts, Ayub and Al-Fahed [11] reported an experimental investigation on fluid friction twisted tape inserts placed separately from the wall. They showed the gap between the tube and the tape to be responsible for the enormous pressure drop .Al-Fahed and Chakroun [12] later investigated the use of wall separated twisted tape inserts on heat transfer for fully developed turbulent flow. They reported a decrease in heat transfer enhancement with increasing tube-tape clearance.

Jian Guo, Aiwu Fan [13] has done a numerical study on heat transfer and friction factor characteristics of laminar flow in a circular tube fitted with center-cleared twisted tape. The computation results demonstrated that the flow resistance can be reduced by both methods; however, the thermal behaviors are very different from each other. For tubes with short width twisted tapes, the heat transfer and thermohydraulic performance are weakened by cutting off the tape edge. Contrarily, for tubes with center-cleared twisted tapes, the heat transfer can be even enhanced in the cases with a suitable central clearance ratio. The thermal performance factor of the tube with center-cleared twisted tape can be enhanced by 7e20% as compared with the tube with conventional twisted tape.

Recently, Bas and Ozceyhan [14] presented an experimental investigation on the use of twisted tape inserts placed separately from the wall. They showed the twist ratio to have a major effect on heat transfer enhancement compared to the clearance ratio. Enhancement efficiencies based on constant pumping power between 1.2 and 1.79 were obtained. The pumping power was shown to reduce as the clearance ratio increased. Patil et al. [15] studied the frictional and heat transfer characteristics of laminar swirl flow of pseudo plastic type power law fluid in a circular tube using varying width twisted tapes under a uniform wall temperature condition. Reduced width twisted tape inserts gave 18%–56% lower isothermal friction factors than the full width tapes. Nusselt numbers decreased only slightly by 5% and 25%, for tape widths of 19.7 and 11.0 mm, respectively inside a 25mm diameter tube. Also, the reduced width tapes offered 20% 50% savings in the tape material as compared to the full width tapes, which is more economical.

S.N.SARDA et al. [16] studied the effect of varying width twisted tape inserts with air as the working fluid in horizontal tube. It is observed that the reduction in tape width causes reduction in Nusselt numbers as well as reduction in pressure drop, the percentage increase in Nusselt numbers for reduced width tapes compared to plain tube are about 11–22%, 16–31%, 24–34% and 39–44% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =3. For full width tapes, the percentage increase is observed to be 58 to 70% compared to plain tube, the percentage increase in Nusselt numbers for reduced width tapes compared to plain tube are about 5–12%, 9–22%, 13–30% and 23–36% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =4. For full width tapes, the percentage increase is observed to be 36 to 42% compared to plain tube.

From the above literature review, the use twisted tape inserts has been shown to be an effective heat transfer enhancement technique. Most studies report high heat transfer enhancement and fluid friction for low twist ratios for both modified and typical twisted tape inserts.

The present investigation is aimed at studying the frictional and heat transfer characteristics in turbulent region using reduced width twisted tape inserts under constant wall heat flux. The objective of using reduced (varying) width twisted tapes is to reduce the pressure drops associated with full width twisted tapes without seriously impairing the heat transfer augmentation rates and to achieve material savings.

II. NOMENCLATURE

Symbol	Meaning	Unit
A	Surface area of tube	m^2
A_c	Cross sectional area of tube	m^2
D	Inside diameter of tube	mm
H	Length between 180˚ Twist	mm
Y	Twist ratio (H/D_i)	-
D_h	Hydraulic diameter	mm
f	Friction factor	-
Nu	Nusselt number	-
Re	Reynolds number	-
$\dot{m}$	Mass flow rate	Kg/s
L	Length of Tube	mm
W	Width of Twisted tape	mm
t	Thickness of Twisted tape	mm
w	Width ratio (W/D_i)	-
ΔP	Pressure Drop	N/m^2
v	Fluid Kinematic viscosity	m^2/s
$C_{p, air}$	Specific heat of air	kJ/kg °C
K	Thermal conductivity	W/m K
ρ	Density	Kg/m^3

T	Temperature	°C
Q	Heat transfer	W
q	Heat flux	W/m^2
h	Average convection heat transfer coefficient	W/m^2K
I	Current	Ampere
V	Voltage	Volts
Pr	Prandtl number	
U	Mean Velocity	m/s
η	Thermal enhancement factor	
$\dot{V}$	Volume flow rate	m^3/s
Subscripts		
b	Bulk	
p	Plain tube	
conv	Convection	
i	Inlet	
o	Outlet	
pp	Pumping power	
s	Surface	
t	Twisted tape	

III. EXPERIMENTAL DETAILS

3.1. *TEST SECTION AND APPARATUS*

The experiments were carried out in an open-loop experimental facility as shown in Fig. 4. The loop consisted of 0.5 hp high pressure blower, orifice meter to measure the flow rate, and the heat transfer test section. The aluminium test tube has a length of L = 500 mm, with 56 mm inner diameter (ID), 60 mm outer diameter(OD), and 2 mm thickness (t). The tube was heated by continually winding flexible electrical wire to provide a constant heat flux boundary condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 3 amps. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings, and necessary precautions were taken to prevent leakages from the system. The inner and outer temperatures of the bulk air were measured with a multi-channel temperature measurement unit in conjunction with the K-type thermocouples. Five K-type thermocouples were tapped on the local wall of the tube. The mean local wall temperature was determined by means of calculations based on the reading of K-type thermocouples.

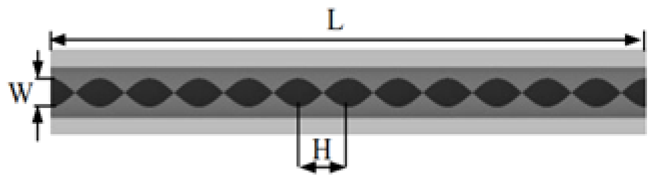

Fig.1. Physical model of a circular tube with reduced width twisted tape inserts

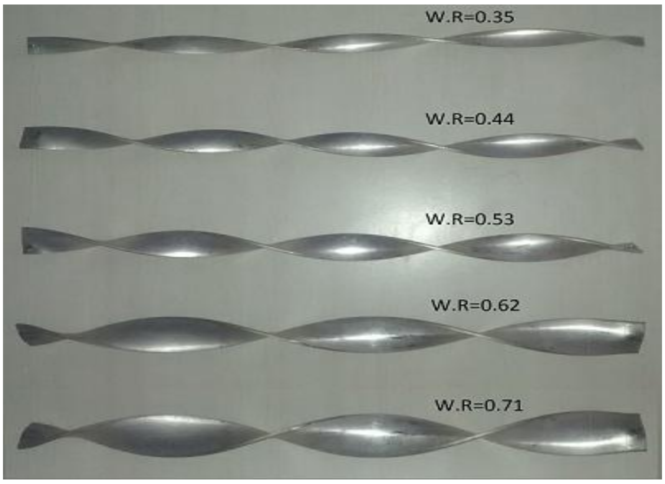

Fig. 2.Photograph of twisted tape inserts with various width ratio

3.2. *EXPERIMENTAL PROCEDURE*

During experiments, inlet bulk air, was drawn through the clam section to achieve a fully developed flow prior to being heated by an adjustable electrical heater wrapping along the test section, Air flow rate was varied corresponding to Reynolds number (Re) between 7500 and 13000. The electrical output power was controlled via variac transformer by keeping the current below 3 amps to achieve a constant heat flux condition along the entire test section. Pressure drop across the test section was measured using an inclined U-tube manometer filled by the manometric fluid having low specific gravity (SG.) of 0.816 to ensure reasonably accurate measurement of the low pressure drop encountered at low Reynolds numbers. In the present test, all data were taken at steady state.

Fig.3.Photograph of experimental setup

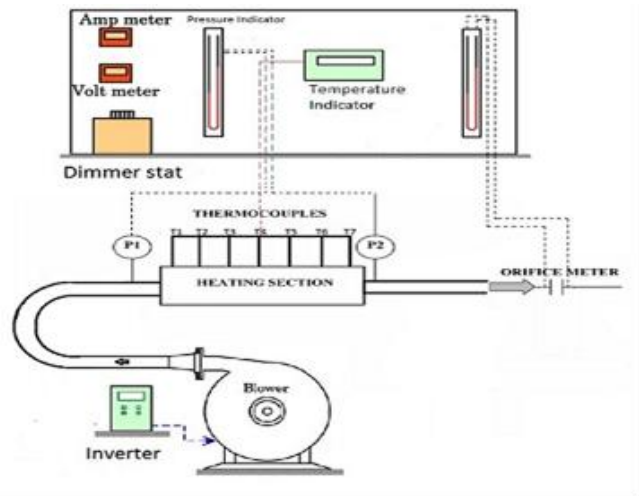

Fig.4. Schematic diagram of experimental heat transfer set-up

IV. DATA REDUCTION

4.1. *HEAT TRANSFER EVALUATION*

In the present work, the air used as the test fluid is flowed through a uniform heat-flux and insulated tube. The steady state of the heat transfer rate is assumed to be equal to the heat loss in the test section which can be expressed as:

$$Q_{air} = Q_{conv} \quad (1)$$

in which

$$Q_{air} = \dot{m}\, C_{p,air} (T_o - T_i) = V\, I \quad (2)$$

The heat supplied by electrical winding in the test tube is found to be 3 to 8% higher than the heat absorbed by the fluid for thermal equilibrium test. Thus, only the heat transfer absorbed by the fluid is taken for internal convective heat transfer coefficient calculation. The convection heat transfer from the test section can be written by

$$Q_{conv} = h\, A\, (T_s - T_b) \quad (3)$$

Where

$$T_b = (T_o + T_i) / 2 \quad (4)$$

And

$$T_s = (T_1+T_2+T_3+T_4+T_5) / 5 \quad (5)$$

where for a constant heat flux, the average surface temperature T_s can be calculated from 5 points of the local surface temperatures, lined equally apart between the inlet and the exit of the test tube. The average heat transfer coefficient, h is estimated as follows

$$h = \dot{m}\, C_{p,air} (T_o - T_i) / A (T_s - T_b) \quad (6)$$

On the other hand, the calculation of a local heat transfer coefficient is based on a specific local wall temperature.
Nusselt number can be calculated using the following equation;

$$Nu = h\, D / k \quad (7)$$

where D is an inner diameter of the test tube and k is a thermal conductivity of the fluid (air).

4.2. *Friction Factor Evaluation*

The pressure drop (ΔP) across the test section length (L) was subjected to the calculation of friction factor via the following equation

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right).\left(\rho.\frac{U^2}{2}\right)} \tag{8}$$

where U is an average velocity calculated by dividing the measured volumetric air flow rate by the inlet cross-section area (A).

The Reynolds number based on an inner diameter of the test tube diameter can be expressed as

$$Re = U\,D / \nu \tag{9}$$

All of the thermo-physical properties (k, ρ, μ, C_p) used for the calculations of the dimensionless numbers (Nu, Pr) are all evaluated at the bulk fluid temperature (T_b) from Eq.(4).

4.3. *Thermal Performance Factor Evaluation*

In the present study, the concept of equal pumping power is applied for evaluating thermal performance. The constant pumping power criteria can be expressed as

For a constant pumping power,

$$(\tilde{V}\Delta P)_p = (\tilde{V}\Delta P)_t \tag{10}$$

Eq. (10) leads to the relationship between friction and Reynolds number under constant pumping power criteria as

$$(f\,Re^3)_p = (f\,Re^3)_t \tag{11}$$

A thermal performance factor (η) of enhancement is defined as the ratio of heat transfer coefficient of the system with enhancement device (h_t) to that of the system without a device (h_p), at the same pumping power (pp) as shown below.

$$\eta = \left(\frac{h_t}{h_p}\right)_{pp} \tag{12}$$

From Equations (11) and (12), a thermal performance factor can be expressed in another form as

$$\eta = (Nu_t / Nu_p) / (f_t / f_p)^{(1/3)} \tag{13}$$

A thermal performance factor with a value above unity indicates that the use of the enhancement device results in overall energy saving as compared to the operation without the device.

V. Results and Discussion

5.1. *Verification of Plain Tube*

The present experimental results on heat transfer and friction characteristics in a smooth wall tube are first validated in terms of Nusselt number and friction factor. The Nusselt number and friction factor obtained from the present smooth tube are, respectively, compared with the correlations of Dittus-Boelter and Blasius found in the open literature for turbulent flow in circular tubes.

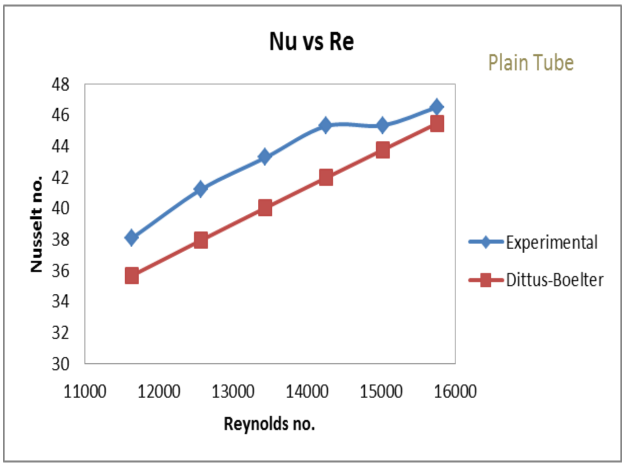

Fig.5. Comparison of the experimental results and the theoretical data of the Nusselt number (Nu)

5.2. *Effects of the Tape Width Ratio*

The experimental data of the Nusselt number and friction factor and their variation with a Reynolds number of twisted tape inserts with width ratios (w = 0.35, 0.44, 0.53, 0.62 and 0.71) are shown in Figures 6 and 7. Figure 6 indicates that the Nusselt number increases with Reynolds number increasing and the heat transfer rate is higher for the twist tape set than for the plain tube because of strong swirl flow in the presence of the twist tape. It is found that the heat transfer rate with the width ratio (w = 0.71) is higher than those with other ratios (w = 0.35, 0.44, 0.53 and 0.62); this means that the turbulent intensity obtained from the higher width ratio is higher than those from lower width ratios (w). Moreover, it is also noted that Nu decreases more rapidly when w is relatively large and then the decrease of Nu slows down with further reduction of w.

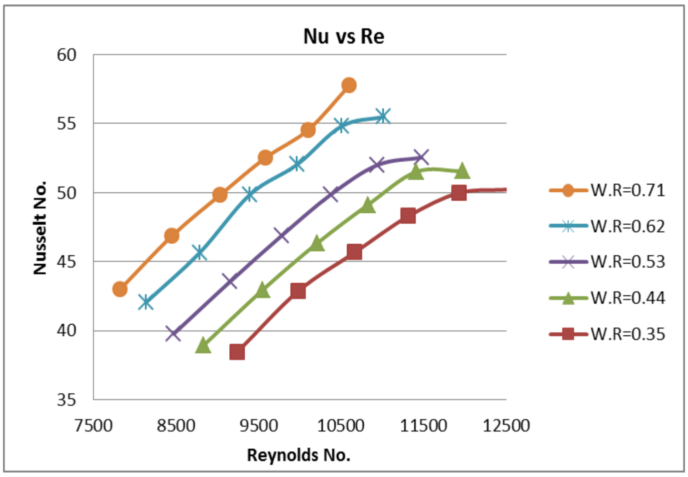

Fig.6. Variation of the Nusselt number (Nu) versus the Reynolds number for width ratio (w) of twisted tape.

Figure 7 shows the variation of friction factor with a Reynolds number for different width ratios (w = 0.35, 0.44, 0.53, 0.62 and 0.71). The friction factor obtained from the tube with twisted tape insert is significantly higher than the plain tube. Moreover, the use of higher width ratio leads to higher tangential contact between the swirling flow and the tube surface. Therefore, the twisted tape with width ratio (w = 0.71) has a maximum friction factor.

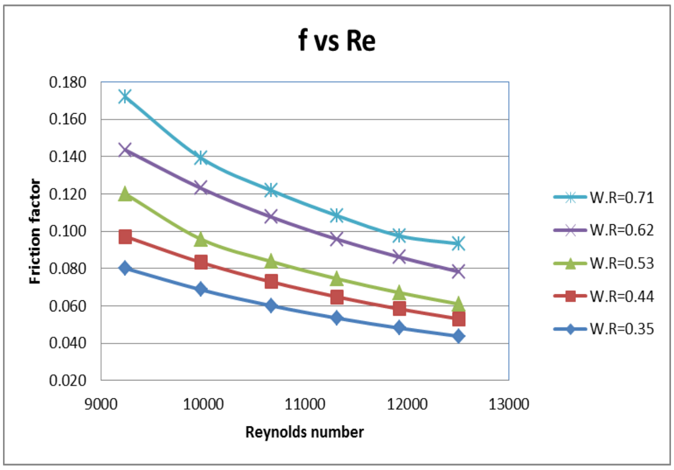

Fig.7. Variation of the friction factor (*f*) versus the Reynolds number for width ratio (w) of twisted tape

It is also noted that the decrement of friction factor is not so rapid as compared with that of Nu when w is relatively large. For instance, when w is reduced from 0.71 to 0.62, the friction factor only decreases by 14-19%, while the decrement of Nu is up to 13-26%. Furthermore, the friction factor drop when w is reduced from 0.71 to 0.62 does not possess that large a proportion as compared with Nu in the total decrement when w is reduced from 0.71to 0.35.

5.3.CORRELATIONS FOR PREDICTION OF HEAT TRANSFER

The correlations were developed for the turbulent flow region in the range of Reynolds number 7500 to 13000. The correlation developed for Nusselt number obtained from the present experimental results of the tube fitted with the reduced width twisted inserts could be written in terms of width ratio (w =W/D), Reynolds number (Re), and Prandtl number (Pr) in Eqs. (14), respectively.

$$\mathbf{Nu = 0.01193\ Re^{0.9404}\ w^{0.3687}\ Pr^{0.33}} \qquad (14)$$

The Nusselt number values predicted from the above correlation Eq. (14). were compared with the experimental values respectively. It could be noted that the Nusselt number values obtained from the predicted correlation Eq. (14). agreed well with the experimental values for all the investigated cases of the proposed correlations.

VI. CONCLUSION

In this present study, the heat transfer enhancement in circular tube fitted with the twisted tapes of having width ratio (W/D) =0.35, 0.44, 0.53, 0.62 & 0.71 at constant twist ratio of 2.5 is experimentally studied in turbulent flow region, Reynolds number = 7500 to 13000.

The key findings based on experimental results of present study are summarized as follows:

i. The heat transfer enhancement offered by twisted tapes of different width ratio is significant compared to plain tube.

ii. The Nusselt number increases with increasing Reynolds no. and width ratio of twisted tape.

iii. The friction factor increases with decreasing Reynolds no. and increasing width ratio of twisted tape.

iv. The enhancement in heat transfer is higher for twisted tape having width ratio=0.71 and lower for width ratio=0.35.

v. The increment in Nusselt no. is 6.04%, 7.95%, 9.56%, 15.46% & 17.26% for width ratio of twisted tape =0.35, 0.44, 0.53, 0.62 & 0.71 respectively.

vi. The increment in thermal enhancement efficiency is higher for higher width ratio and it decreases with decreasing width ratio of twisted tape at constant pumping power.

vii. In general observation, it is found that the heat transfer, friction factor and thermal efficiency increased with increasing width ratio.

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