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ENHANCEMENT OF HEAT TRANSFER OF WATER FOR TURBULENT FLOW THROUGH TUBE USING U-CUT TWISTED TAPE INSERTS

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Abstract: In this experiment tube side heat transfer coefficient, pressure drop, friction co-efficient and percentage of increase in those parameters for water using U-cut twisted tape inserts into the tube were measured. The test section consists of a circular tube made of copper having 26.6 mm inside diameter, 30 mm outside diameter and 900 mm in effective length. A stainless steel U-cut twisted tape insert of 5.29 twist ratio and 0.4 mm in thickness was inserted into the smooth tube. The U-cut had 8 mm depth and 8mm width. In this investigation, the test section was electrically heated. Five K-type thermocouples were used in the test section for measuring the wall temperature. The heat flux was found in the range of 18.33~28 kW/m^2 for smooth tube whereas 32.07~47.24.00 kW/m^2 for tube with insert for Reynolds number range of 10153~19217. At comparable Reynolds number, Nusselt number in the tube with U-cut twisted tape insert was enhanced by 2.76 to 3.24 times with compared to smooth tube and friction factor in the tube with U-cut twisted tape insert was also increased by 1.6 times with compared to plain tube.

Keywords: *U-shape insert, Heat transfer rate, Friction factor, Pressure drop, Heat transfer performance.*

NOMENCLATURE

A = Area
C_p = Specific heat
d = Tube diameter
d_e = U-shape depth
f = Friction factor
h = Heat transfer
k_w = Thermal conductivity
L = Tube length
L_{tap} = Twisted tape length
m = Mass flow rate
q = Heat flax
Q = Heat transfer rate
r = Radius
T_o = Hot water temperature
T_i = Cold water temperature
T_b = Bulk temperature
T_{wo} = Outer surface temperature
T_{wi} = Inner surface temperature
u_m = Mean velocity
V = Velocity
W = Twisted tape width
w = U- cut width
y = Twist ratio

Symbol:

ρ = Density
ΔP = Pressure drop
μ = Dynamic viscosity
η = Thermal enhancement factor

Number:

Nu = Nusselt number
Pr = Prandtl number
Re = Reynolds number

1. INTRODUCTION

Heat exchanger is the apparatus that facilitate heat transfer between two or more fluids, and they can be classified according to the mode of flow of fluid or their construction methods. Heat exchangers with the convective heat transfer of fluid inside the tubes are frequently used in many engineering application for example heat recovery process, air-conditioning and refrigeration, internal cooling of I. C. engine, chemical reactors

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etc. Thermal performance of heat exchangers defines the efficiency and economic competitiveness for many industrial processes. Improving the thermal performance of the heat exchangers may cause in the reduction of its size as well as initial cost. A high-performance heat exchanger of a fixed size can provide an increased heat transfer rate; it might also cause a decrease in the temperature difference between the process fluids, enabling efficient utilization of thermodynamic ability. To boost heat transfer and improve thermal performance, various heat transfer enhancement techniques are widely used from the last twentieth century. At present, the technology of the twisted-tape insert is widely used in various industries. Insertion of twisted tapes in a tube provides a simple passive technique for enhancing the convective heat transfer by introducing swirl into the bulk flow and by disrupting the boundary layer at the tube surface due to repeated changes in the surface geometry. It was explained that such tapes induce turbulence and superimposed vortex motion (swirl flow) causing a thinner boundary layer and consequently resulting in a high heat transfer coefficient and Nusselt number due to repeated changes in the twisted tape geometry.

There are a number of research works done previously in order to investigate the heat transfer characteristics using twisted tapes. Eiamsa-ard *et al.* [1] made a comparative investigation of enhanced heat transfer and pressure loss by insertion of single TT, full-length dual twisted tape (TT) and regularly-spaced dual twisted tape as swirl generators. The result shows that all dual twisted tape with free spacing yield lower heat transfer enhancement in comparison with the full-length dual twisted tape. Thianpong *et al.* [2] investigated experimentally the friction and compound heat transfer behaviours in a dimpled tube fitted with a twisted tape swirl generator, using air as working fluid. The experiments were conducted by using two dimpled tubes with different pitch ratios and three twisted tapes with three different twist ratios. The results showed that both heat transfer coefficient and friction factor (f) in the dimpled tube fitted with the twisted tape , was higher than those in the dimple tube acting alone and plain tube. Promvonge and Eiamsa-ard [3] investigated thermal characteristics in a circular tube fitted with conical-ring and a twisted tape swirl generator. Murugesan *et al.* [4] reported the heat transfer enhancement (HTE) and friction factor characteristics of tube fitted with V-cut twisted tape insert. Eiamsa-ard *et al.* [5] experimentally measured the influences on Nusselt number (Nu) and friction factor of twin counter /Co twisted tapes fitted in tube. Chang *et al.* [6] experimentally studied the spiky twisted-tape insert (swirl tube) placed in a tube. The dispersed rising air bubbles in the plain tube and the centrifugal-force induced coherent spiral stream of coalesced bubbles in the swirl-tube core considerably modify the pressure-drop and heat-transfer performances from the single-phase conditions. Heat transfer, flow friction and thermal performance factor characteristics in a tube fitted with delta-winglet twisted tape, using water as working fluid are investigated experimentally for oblique delta-winglet twisted tape (O-DWT) and straight delta-winglet twisted tape (S-DWT) arrangements over a Reynolds number range of 3000–27,000 by considering three twist ratios (y/w = 3, 4 and 5) and three depth of wing cut ratios (DR = d/w = 0.11, 0.21 and 0.32) by Wongcharee *et al.* [7]. Shabanian *et al.* [8] reported the experimental and computational fluid dynamics modeling on heat transfer, friction factor and heat transfer enhancement efficiency of an air cooled heat exchanger equipped with classic and jagged twisted tape. Saha [9] experimentally studied the heat transfer and the pressure drop characteristics of rectangular and square ducts with TT insert with oblique teeth. From experiment it is found that, the axial corrugation in combination with TT with oblique teeth performs better than those without oblique teeth. Bhuiya *et al.* [10] presented the heat transfer improvement and developed new-correlations for turbulent flow through a tube with triple helical tape inserts. Friction and heat transfer characteristics of turbulent air flowing through tubes with twisted strip swirl promoters were studied experimentally and analytically by Thorsen and Landis [11]. Data were obtained for pitch-to-diameter ratios as low as 3.15 and for Reynolds numbers up to 100,000. Both heating and cooling tests were run for tube wall to fluid bulk temperature ratios from 0.6 to 1.9 to assess compressibility and buoyancy effects.

The present investigation is aimed at studying the frictional and heat transfer characteristics in turbulent region of a circular tube using U-cut twisted tape insert of water. The experimental data are correlated and new formula of Nusselt number and friction factor are reported for this experiment. Data are compared with smooth tube heat transfer and friction values and the percentage of increase of heat transfer enhancement using U-Cut twisted tape are reported.

2. EXPERIMENTAL FACILITY

The test section is a smooth circular tube made of copper having 26.6 mm inside diameter, 30 mm outside diameter and 939.8 mm in long, of which length of 900 mm was used as the test section. A stainless steel twisted tape was made by twisting 4mm thick 20 mm width straight strip. Its length and twist ratio was respectively 940 mm and 5.29 mm. Fig. 1 displays the design of the inserts used in the current experiment.

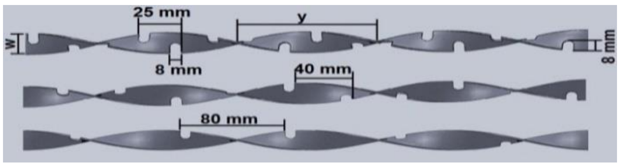
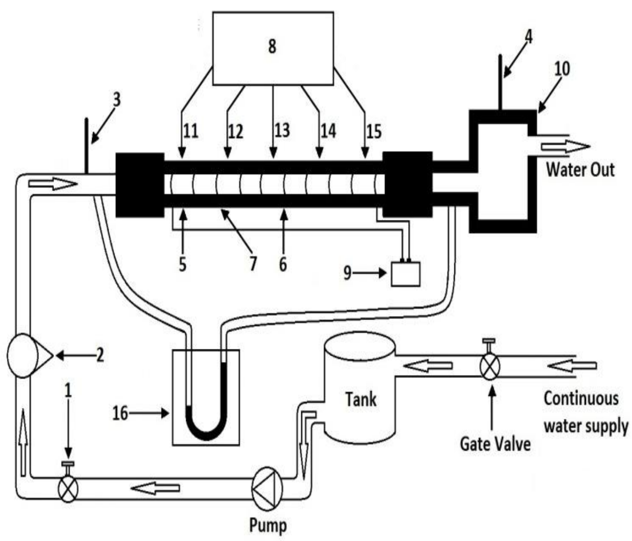


Fig. 1 U-shape twisted tape inserts.

A schematic diagram of the experimental setup is shown in Fig. 2. The nichrome wire was used as the resistance wire for the winding of the heater. A constant heat flux condition was maintained by wrapping Nichrome wire around the test section and fibre glass insulation over the wire. Power was supplied to heater form an AC source of 220 volt and it was made to 70 volt by using voltage regulator. Mica sheet was used between the tube and heating wire for electrical insulation. Outer surface temperature of the tube was measured at five points of the test section maintaining equal distance from one point to another point by five K-type thermocouples. In order to prevent leakage Teflon tape has to be used for the joining of the tube and after that M-seal was used. Then the tube was wrapped at first with mica tape before wrapping with nichrome wire spirally wounded uniformly around the tube. Two thermometers were used at the inlet and outlet section of the tube for measuring the bulk temperatures. At the outlet section the thermometer was placed in a mixing box. The rate of flow was measured with the help of Rota meter which can measure maximum of 26l t/min of water. A U-tube manometer was used to measure the pressure drop across tube. Open loop system of water supply was used. From a continuous water system, the water was forced in the reservoir and then using a pump, water was pumped into the test section have shown in Fig. 2. A gate valve was placed in between the reservoir and the test section to control the flow of water.



1. Gate valve
2. Rotameter
3. Inlet thermometer
4. Outlet thermometer
5. Insulation
6. Test section (copper tube)
7. Nichrome-wire coil
8. Thermo-electric monitor
9. Voltage regulator
10. Mixing box
11. Thermocouple 1
12. Thermocouple 2
13. Thermocouple 3
14. Thermocouple 4
15. Thermocouple 5
16. Manometer

Fig. 2 Schematic diagram of the experimental setup.

The rate of flow was measured with the support of rotameter (Metric 24G, SS float) of 26 L/min capacity in the travelling path of inlet water. Data were collected for smooth tube and using the inserts in smooth tube.

3. MATHEMATICAL MODELING

Heat transfer performance of insert in water at circular tube was calculated by using the following equations.

Outer surface area was calculated from,

$$A_0 = \pi d_0 L \tag{1}$$

Where, d_o is outer surface diameter.

Inner surface area was calculated from,

$$A_s = \pi d_i L \tag{2}$$

Where, d_i is outer surface diameter.

The experimental friction co-efficient

$$f = \frac{2\Delta P d_i}{\rho L {u_m}^2}; \tag{3}$$

Where, mean velocity,

$$u_m = \frac{m}{A_f}, \tag{4}$$

Flow area,

$$A_f = \frac{\pi}{4} d_i^2 \tag{5}$$

Pressure difference was obtained from,

$$\Delta p = \Delta h \times \rho \times g \times 13.6 \tag{6}$$

Added heat was calculated by,

$$Q = m c_p (T_o - T_i) \tag{7}$$

Velocity,

$$V = \frac{m}{A_x}; \tag{8}$$

Where, m is flow rate and

Cross sectional area, $A_x = \frac{\pi d_i^2}{4}$ (9)

Reynolds Number, $Re_D = \frac{\rho V d_i}{\mu}$ (10)

Nusselt number, $Nu_{exp} = \frac{h d_i}{k}$ (11)

Dittus Boelter equation, $Nu_D = 0.023 Re_D^{\frac{4}{5}} Pr^n$; (12)

Where n is 0.4 for heating and 0.3 for cooling.

Prandtl number, $Pr = \frac{\mu c_p}{k}$; (13)

μ and k at bulk temperature.

Convective heat transfer coefficient was calculated from,

$$h = \frac{Q}{A(T_{wi} - T_b)} \tag{14}$$

And heat flux was found from, $q = \frac{Q}{A_s}$ (15)

Theoretical friction factor was calculated from,

$$f_i = (0.79 \ln Re - 1.64)^{-2} \tag{16}$$

Bulk temperature, $T_b = \frac{T_i + T_o}{2}$ (17)

Outer surface temperature, $T_{w0} = \sum_{i=1}^{5} \frac{T_{w0,i}}{5}$ (18)

Inner surface temperature,

$$T_{wi} = T_{w0} - Q \frac{\ln(d_o - d_i)}{2\pi k_w L} \tag{19}$$

Theoretical Nusselt number was calculated from Gnielinski (1976), correlation, $Nu_{thG} = \frac{\left(\frac{f}{8}\right)(Re - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$ (20)

% of error = $\left(\frac{Nu_{exp} - Nu_D}{Nu_D}\right) 100$ (21)

All the fluid properties were evaluated at bulk temperature.

4. RESULTS AND DISCUSSION

4.1 Plain Tube Data Evaluation

Fig. 3 shows the variation of Nusselt number with the variation of Reynolds number. This figure actually demonstrates relation between experimental Nusselt number and those calculated from Gnielinski (20) and Dittus Boelter (12) correlation.

For Gnielinski (1976) correlation data fall within an error of 4.46% to 0.45% for which is acceptable range with r.m.s. value of error 3.04%. B. Salam *et al.* [11] used the same set up with rectangular cut twisted tape insert were found r.m.s. value of error 20.3%. For Dittus Boelter correlation data fall within 34% to 26.04% with r.m.s. value of error 30.2%. B. Salam *et al.* [12] found an error of 12% for the same set up where they used rectangular cut twisted tape insert for Dittus Boelter correlation.

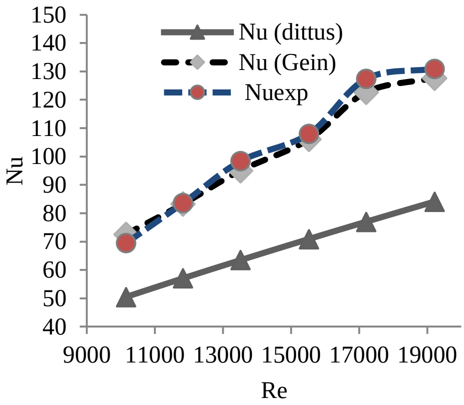


Fig. 3 Variation of Nusselt number with Reynolds number (Smooth Tube).

The variation of friction factor with Reynolds number is shown in Fig. 4. The data obtained from experiment is compared with first Petukhov (1970) equation Eq. (16). The figure shows that the friction factor determined from Petukhov equation (f_{th}), was significantly low compared with the experimental data. Data calculated from this equation showed a deviation of ± 3.7% and data fall within 29.45% to 19.46% of error.

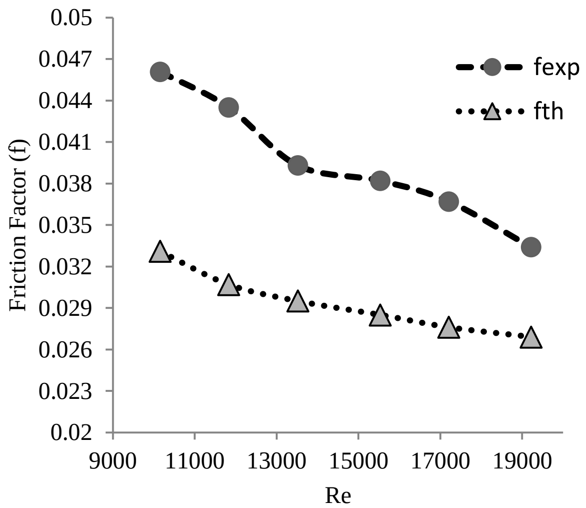
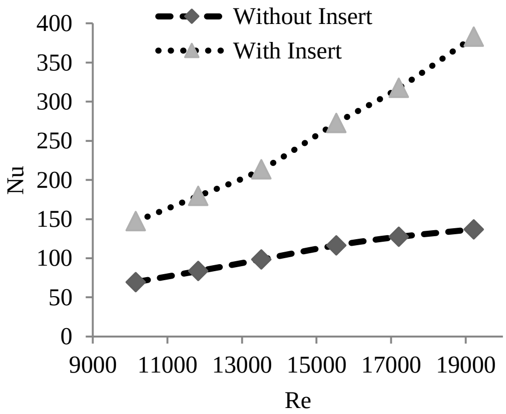


Fig. 4 Variation of friction factor with Reynolds number (Smooth Tube).

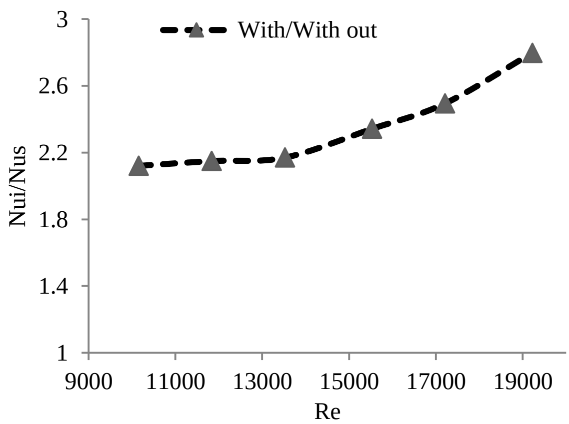
4.2 Effect of U-Cut Twisted Tape Insert

Fig. 5 (a) represents the change of Nusselt number for various Reynolds number in the tube fitted with U-cut twisted tape (UTT) and for the plain tube. From the figure, it is observed that for all of the cases, Nusselt number increases with an increase of Reynolds number. As expected U-cut twisted tape has higher heat transfer rates compared with plain tube as Eisma-ard *et al.* [13] explained that by using twisted tape, swirl flow are generated which is responsible for thinning the thermal boundary layer and increasing the mixing between the core and tube wall flows. In this experiment U-cut twisted tape was responsible for extra disturbances between the wall surface of the tube and with the secondary flow. At different Reynolds number experimental Nusselt number in the tube fitted with U-cut twisted tape were enhanced by 2.76 to 3.24 times compared to the plain tube and an average increase of 2.1 times.

The effectiveness of heat transfer enhancement of the tube equipped with the U-cut twisted tape inserts compared to that of the plain tube, in terms of Nusselt number ratio (Nu_i/Nu_s), is presented in Fig. 5(b). It is clear from Fig. 5(b) that the Nusselt number ratio of all the investigated cases was more than the unity. This implied the beneficial gain for heat transfer enhancement of using the twisted wire inserts over the plain tube. From Fig. 5(b), it could be noted that the Nussselt number ratio tended to increase with increasing Reynolds number.



(a)



(b)

Fig. 5 The variation of Nusselt Number with Reynolds number: (a) Nu and (b) Nu_i/Nu_s.

The variation of heat flux for different Reynolds number through plain tube and tube fitted with U-cut twisted tape is shown in the Fig. 6. It could be clearly stated from Fig. 6 that heat flux for the tube fitted with U-cut twisted tape was higher compared with the plain tube. Coefficient of convective heat transfer for different Reynolds number is presented in Fig. 7. As the heat flux is higher for U-cut twisted tape insert coefficient of convective heat transfer was also higher for U-cut twisted tape insert.

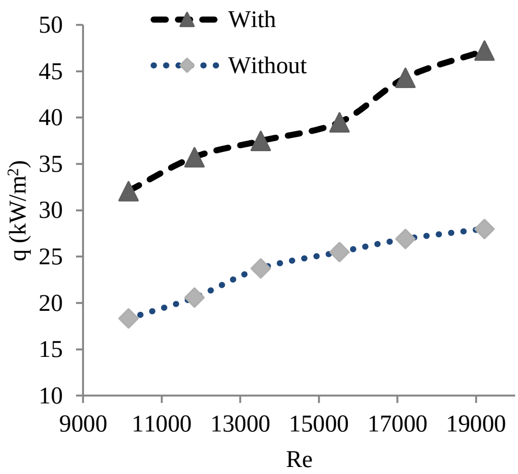


Fig. 6 Variation of heat flux with Reynolds Number.

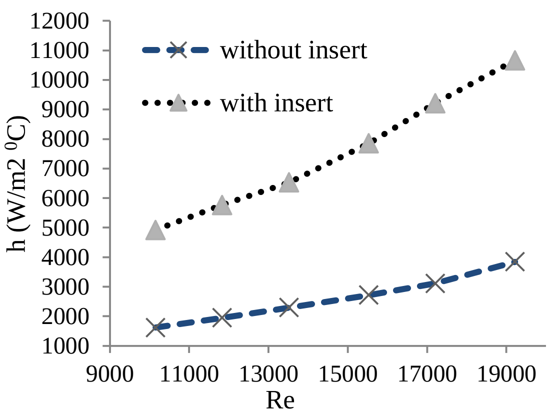
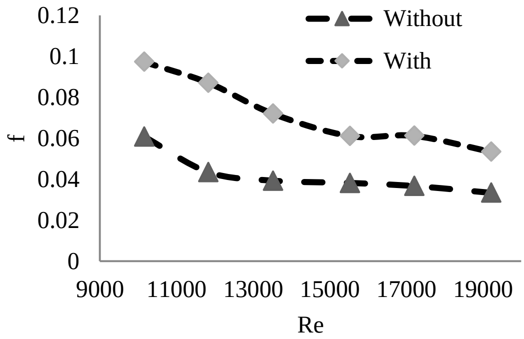


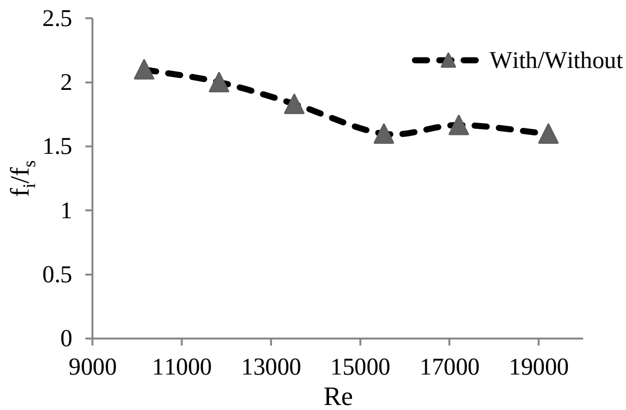
Fig. 7 The variation of convective heat transfer coefficient with Reynolds number.

Fig. 8(a) shows the variation of friction factor with Reynolds number. From the figure, it was observed that friction factor was decreased with the increase of Reynolds number for both of the cases. Friction factor with U-cut twisted tape were found to be higher than plain tube. Friction factor in the tube with U-cut twisted tape insert was increased by 1.6-2.0 times with compared to plain tube. Eisma-ard *et al.* [13] reported that swirl flow causes high viscous loss near the wall region and which is responsible for this higher value of friction factor in twisted tape compared with plain tube. Where Fig. 8(b) depicts the friction factor ration with insert to the plain tube (f_i/f_s). Friction factor ratio decreased with the increase of Reynolds number and always greater than unity. Fig. 9 represents the variation of pressure drop for different Reynolds number and it could be stated that the pressure drop was higher for the tube with U-cut twisted tape insert compared with the smooth tube.

Fig. 9 represents the variation of pressure drop for different Reynolds number and it could be stated that the pressure drop was higher for the tube with U-cut twisted tape insert compared with the smooth tube.



(a)



(b)

Fig. 8 The variation of friction factor with Reynolds number: (a) f and (b) f_i/f_s.

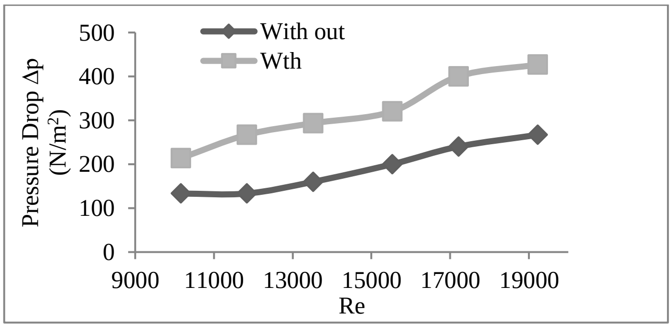


Fig. 9 Variation of Pressure drop with Reynolds Number

5. CONCLUSIONS

An experimental investigation was carried out to determine tube-side heat transfer coefficient, friction factor, heat flux of water for turbulent flow in a circular tube which was fitted with stain less steel U-cut twisted tape insert. From the experiment, it was found that, in case of plain tube,

- Nusselt number for U-cut twisted tape was increased by 2.76 to 3.24 times than that of the plain tube.
- Friction factor for U-cut twisted tape insert was increased by 1.6-2.0 times than plain tube.
- Friction factor was decreased with the increase of Reynolds number for both of the case.
- Heat flux was increased by 1.54 to 1.75 times for UTT compared to plain tube.
- Coefficient of convective heat transfer was also increased by 2.77 to 3.04 times for U-cut twisted tape insert compared to plain tube.

Further investigation can be carried out to investigate heat transfer enhancement efficiency for U-cut twisted tape insert and for different spacing of the twist and twist ratio.

6. ACKNOWLEDGEMENT

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