Dept. of Mech. Eng.
CUET

Published Online April 2017 (http://www.cuet.ac.bd/merj/index.html)

Mechanical Engineering Research Journal

Vol. 10, pp. 07-13, 2016

ISSN: 1990-5491

ENHANCEMENT OF HEAT TRANSFER USING U-SHAPED TWISTED TAPE INSERTS IN DIFFERENT SPACING FOR TURBULENT FLOW

M. A. Razzaq, M. A. M. Hossain, J. U. Ahamed* and H. Barman

Department of Mechanical Engineering, Chittagong University of Engineering and Technology, Chittagong-4349, Bangladesh

Abstract: In this experimental study, turbulent flow heat transfer in a tube having different spacings of a U shaped twisted tape inserts with water as working fluid was evaluated. The test section consisted of a circular copper tube of 26.6 mm inner diameter, 900 mm length with five K-type thermocouples. Bulk temperature and pressure drops were measured. The aluminum inserts were 800 mm in length, 25 mm width, 1.5 mm thickness and twisted ratio of 5 was used. Heat flux, friction factor and Nusselt number were calculated to analysis heat transfer performance of circular tube fitted with and without inserts in turbulent regimes (4000<Re<20000). Heat transfer rate for inserts 25 mm, 40 mm and 80 mm in spacings in U-shape was increased by 4, 3, 2 and Nusselt number was risen by 1.2, 1.18, 1.06 times respectively compared to smooth tube. Heat transfer performance for inserts were found 1.57~1.67, 1.32~1.40, 1.17~1.27 times better than smooth 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 transfer augmentation technique is improving continuously. The study to increase heat transfer has been started at the beginning of twentieth century. Many investigations are already successfully completed and being used in many sectors. According to the modern studies, it is observed that the performance dealt with ‘U-shape twisted tape inserts ‘is the most economical heat transfer enhancement tool. It is important to verify the implementation of heat exchanger tool to know the tube side as well as shell side heat transfer. Heat exchanger is the tool which provides heat transfer between two or more fluids and develops the high performance thermal system. It improves

* **Corresponding author:** Email: jamal293@yahoo.com; Tel: 880-1720641636

the heat exchanger effectiveness for internal and external flows. Naturally, the mixing of the fluids are increased by increasing flow vorticity, unsteadiness, turbulence or by limiting the growth of fluid boundary layers close to the heat transfer surfaces.

Heat transfer augmentation techniques are widely used in areas specified as temperature effort transform, air conditioning and preservation systems and chemical reactors. Basically the experiments are studied, using different shapes of inserts, different types of materials, in varieties experimental works for dissimilar environments. Due to rapid growth of industrialization the research on this topic has been enlarged significantly. Nowadays it is one of the most important topics to enhance heat transfer with minimizing the material cost. Many researches have been carried out to increase convective heat transfer for various flow conditions, reducing friction factor and use of power.

Bergles *et al.* [1] declared the obtainable literature on convection temperature soul. According to the literature, the augmentation techniques were classified into two categories (Passive and active).Passive augmentation techniques, which required no external power, and active techniques, which do required external power. The fourteen techniques were grouped in terms of their application to the various modes of heat transfer. Mass transfer was included for completeness. The nonviolent methods were supported on the identical principle. Use of this technique causes the whirl in the volume of the fluids and disturbs the literal line layer so as to growth strong articulator mass devices were generally victimized as supine alter transpose enhancement techniques:

- Inserts
- Extended aboveground
- Shallow modifications
- Use of additives.

U-shape twisted tap insert provides an additional disturbance to the fluid in the section of the tube wall and vorticity behind the cuts and thus leads to a higher energy soul improvement in comparison with smooth tube. Salam et *al.* [2] made an investigation to enhance tube side heat transfer with rectangular shaped twisted tape insert. It was observed that, the Reynolds numbers were varied from 10000 to 19000 ranges with heat flux variation rangingfrom14 to 22 $kW/m^2$for smooth tube and 23 to 40 kW/m^2 for tube with insert. Nusselt numbers for smooth tube were compared with Gnielinski correlation and errors were found from -6% to -25% ranges with R.M.S. value of 20%.Nusselt numbers in tube with rectangular-cut twisted tape insert were enhanced by 2.3 to 2.9 times with the increase of friction factors by 1.4 to 1.8 times compared to the smooth tube. Heat transfer enhancement efficiencies were found in the range of 1.9 to 2.3 and increased with the increase of Reynolds number. Mesh or spiral brush inserts were used by Megerlin *et al.* [3] to enhance heat transfer in short channels subjected to high heat flux. For turbulent flows the heat transfer coefficient can be developed as much as 8.5 times that in a smooth tube, but pressure drop was very high. Sarada *et al.* [4] made an investigation with varying width twisted tape inserts ranging from 10 mm to 26 mm. The Reynolds number varied from 6000 to 13500. It was found that the increment of heat transfer with twisted tape inserts as compared to plain tube varied from 36~48% for width 26 mm and 33~39% for width 22 mm inserts. A new idea was postulated by Hsieh and Huang [5] to foretell heat transfer and pressure drop of laminar flow in horizontal tubes with/without longitudinal inserts. They said that enhancement of heat transfer as compared to a conventional bare tube at the same Reynolds number to be a factor of 16 at Re ≤ 4000, while a friction factor rise of 4.5.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 [6]. 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. Eiamsa-ard *et al.* [7] conducted an experimental study for a round tube with short-length twisted tape insert on the mean Nusselt numbers, friction factor and enhancement efficiency characteristics under uniform wall heat flux boundary conditions. For swirling flow, the short-length tape was established for generating a strong swirl flow at the tube entry when the full-length twisted tape was inserted into the tube at a single twist ratio of y/w = 4.0. The enhancement efficiency with the short-length insert was found to be lower than that with the full-length insert. Eiamsa-ard *et al.* [8] were continued the investigations and used DI-coil in common with the insert at lower Reynolds number and found the highest thermal performance factor of around 1.25. Again, Eiamsa-ard *et al.* [9] made an experiment using single twisted tapes and full-length dual twisted tapes with three different twist ratios (y/w = 3.0, 4.0 and 5.0) and also regularly-spaced dual twisted tapes with three different space ratios (s/D = 0.75, 1.5 and 2.25). The result observed that the heat transfer of the tube with dual twisted tapes was higher than that of the plain tube with/without single twisted tape insert. Altaie *et al.* [10] studied with ribs assembly of 5 x 5 mm cross section and fitted in the tube and separated by 8cm pitch. Results of temperature and velocity distribution along the tube center line for the case of tube with internal ribs were compared with that of plain tube, those results showed that the use of internal ribs enhance the heat transfer rate and found to possess the highest performance factors for turbulent flow. Heat transfer performance for turbulent flow in a circular tube with a porous/perforated twisted tape insert was experimentally investigated by Wazed *et al.* [11]. In the analysis a wide range of Reynolds numbers (1.3×10^4 to 5.2×10^4) were observed. The heat transfer coefficient was enhanced (up to 5.5 times) in the cost of increasing pumping power (1.8 times) in turbulent flow through a tube with the inserts. The heat transfer effectiveness in a tube with a perforated twisted tape insert was found to increase up to 4.0 times compared with the value for the plain tube. An investigational research of heat transfer performance of porous twisted tape insert in a circular tube was carried out by Ahamed *et al.* [12]. In arrange of Reynolds number 1.3 x 10^4 to 5.2 x 10^4 the results found for tube with porous twisted tape insert, the average heat transfer coefficient was 2.60 times higher, the heat flux was 1.55 times higher, the friction factor was 2.25 times higher and the pumping power was 2.0 times higher than plain tube values. U-shape twisted tape is very economic and also reduce the pressure drop as well as uses of material. The experiment focuses on convection heat transfer enhancement inside the tube by using U-shaped twisted tape inserts. The main aim of the current research was to find the heat flux and pressure drop using U-shaped twisted tape inserts of different U-shaped spacing's to choose the best one. Another important finding was to determine the efficiency of three different inserts and choose the best among them comparing with smooth tube. There were some limitations according to the study. It was not easy to use twisted inserts in turbulent flow region as the value of pressure drop was high. So, high cost required to do this experiment. Lower twist ratio offered higher heat transfer rate than the higher twist ratio because intensity of turbulence and flow length obtained from lower twist ratio was high. It was hard to handle this process without heat loss. The heater could not wrap properly, therefore equal distance was not found in every curl. The bulk temperatures were measured by average of input and output temperatures.

2. MATERIALS AND METHODS

In this experimental work, three U- cut twisted tape inserts of aluminum strips were used. The inserts length were 800 mm, width were 25 mm and thickness were 1.5 mm. Twisted ratio of 5 and the width and depth of 8 mm of the U-shaped was used. The inserts varied in the U-shaped spacings and the spacing's were 25 mm, 40 mm and 80 mm respectively. Fig. 1 displays the design of the inserts used in the current experiment.

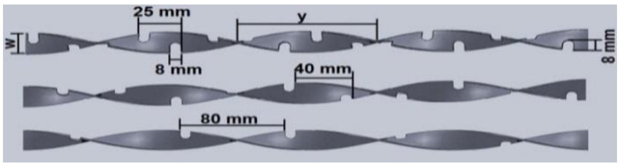
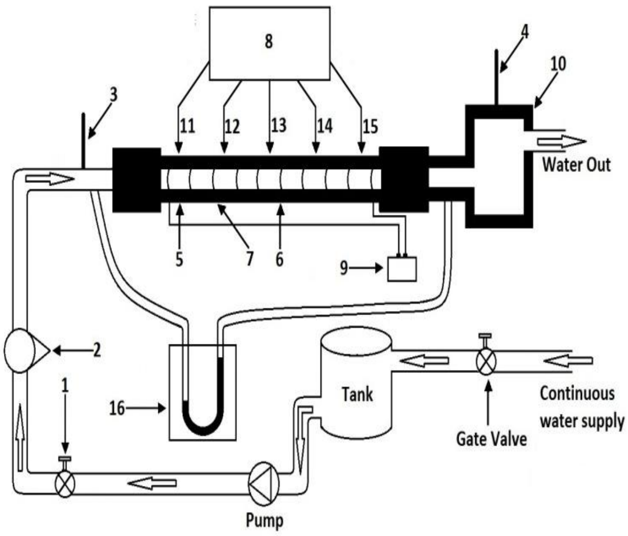


Fig. 1 U-shape twisted tape inserts.

A schematic diagram of the experimental setup is shown in Fig. 2. A long copper tube of 26.6 mm internal diameter and 30 mm outer diameter and length of 900 mm was used as the test section. A nichrome wire was wrapped around the test section in a periodic distance to heat the whole section equally. Therefore, constant heat flux condition was maintained. To avoid heat loss, sufficient quantity of fiber glass insulation was used with mica sheet wrapped over the wire. 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 K-type thermocouples. Two thermometers were used at the inlet and outlet section of the tube for evaluating the bulk temperature. Pressure drop was measured at inlet and outlet of the test section. A manometer was used to determine the pressure drop at inlet and outlet of the test section. 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	9. Voltage regulator
2. Rotameter	10. Mixing box
3. Inlet thermometer	11. Thermocouple 1
4. Outlet thermometer	12. Thermocouple 2
5. Insulation	13. Thermocouple 3
6. Test section (copper tube)	14. Thermocouple 4
7. Nichrome-wire coil	15. Thermocouple 5
8. Thermo-electric monitor	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.

In the beginning, the tank was filled by water from a continuous source and then pumped to the test section through rotameter. The flow was controlled by a gate valve to make it steady. The flow rate of water was varied from a range of minimum 5.4 L/min and maximum 20.7 L/min measured by the rotameter. After switching on the heating unit, enough time was given to achieve the steady state condition. Then using thermometers, the inlet and outlet temperature of water were taken. After that, data were collected from the K-type thermocouples. A menometric pressure drop was counted for a single flow rate by a manometer. In each run of water into the tube, data were taken for water flow rate, inlet temperature, outlet temperature, tube outer surface temperatures and pressure drop readings in the prospective way.

3. MATHEMATICAL FORMULATIONS

Heat transfer performance of insert in water at circular tube was calculated by using the following equations.
Outer surface area was calculated from,

$A_0 = \pi d_0 L$ (1)
Where, d_o is outer surface diameter.

Inner surface area was calculated from,
$A_s = \pi d_i L$ (2)
Where, d_i is outer surface diameter.
The experimental friction co-efficient
$f = \frac{2\Delta P d_i}{\rho L {u_m}^2}$; (3)

Where, mean velocity,
$u_m = \frac{m}{A_f}$, (4)

Flow area,
$A_f = \frac{\pi}{4} d_i^2$ (5)

Pressure difference was obtained from,
$\Delta p = \Delta h \times \rho \times g \times 13.6$ (6)

Added heat was calculated by,
$Q = m c_p (T_o - T_i)$ (7)

Velocity,
$V = \frac{m}{A_x}$; (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,
$h = \frac{Q}{A(T_{wi} - T_b)}$ (14)

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

Theoretical friction factor was calculated from,
$f_i = (0.79 \ln Re - 1.64)^{-2}$ (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,
$T_{wi} = T_{w0} - Q \frac{\ln(d_o - d_i)}{2\pi k_w L}$ (19)

% of error = $\left(\frac{Nu_{exp} - Nu_D}{Nu_D}\right) 100$ (20)

Efficiency, $\eta = (\frac{Nu_{exp}}{Nu_D}) / (\left(\frac{f}{f_i}\right)^{\frac{1}{3}})$ (21)
All the fluid properties were evaluated at bulk temperature.

4. RESULTS AND DISCUSSION

Nusselt numbers were calculated to compare the data with Dittus and Boelter [13] values found from Eq. (12) and errors were found within -3.82% and +6.5% ranges with R.M.S. error value of 4.75%. For smooth tube, using Eq. (10) the results were found that the Reynolds number values varied in 4949.14 to 18971.72ranges, which were remain unchanged after using inserts.

Heat transfer rate was found from the Eq. (7) in the range of 3.76 to 14.42 kW for smooth tube. Convective heat transfer coefficient was found in a ranges of 790.16 to 3057.83 W/m^2.^{o}c, Nusselt number was calculated from Eq. (11) and got the result in between 33.96 to 131.62 ranges, and friction factor raised in the range of 0.022 to 0.057 and heat transfer enhancement efficiency varied in the range of 0.94 to 1.26 for plain tube.

Afterward, using the insert of 25 mm spacing between U-shapes the results were obtained that heat transfer rate was found in the range of 1.544 to 5.77 W, convective heat transfer coefficient in 956.50 to 3709.03 W/m^2.^{o}c ranges, Nusselt number in between 41.10 to 159.64 ranges, friction factor was found from 0.18 to 0.391 ranges, where pressure drop was measured by applying Eq. (6), and heat transfer enhancement efficiency was improved from 1.57 to 1.98 ranges.

By using the insert of 40 mm spacing between U-shapes the results were obtained that heat transfer rate found between 11.28 to 43.25 kW ranges, convective heat transfer coefficient varied from 931.37 to 3610.45 W/m^2.^{o}c ranges, Nusselt number was found in the range of 40.02 to 155.40, friction factor values were obtained from 0.098 to 0.213 ranges, and heat transfer enhancement efficiency increased from 1.32 to 1.66 ranges.

For, the insert with 80 mm spacing, the results found for heat transfer rate in 0.722 to 2.88 kW ranges, convective heat transfer coefficient varied from 834.83 to 3232.45 W/m^2.^{o}c ranges, Nusselt number in 35.87 to 139.13 ranges, friction factor values found in between 0.052 to 0.107 ranges and heat transfer enhancement efficiency raised in 1.19 to 1.47 ranges.

The variation of Nusselt number with Reynolds number for smooth tube and tube with U-shape twisted tape inserts are shown in Fig. 3. It was found that, Nusselt number increased with the increment of Reynolds number. The maximum value was found for inserts of 25 mm spacing's in U-shape. Nusselt number increased with three inserts of 25 mm, 40 mm and 80 mm in spacing between U-shapes by 1.20, 1.18, 1.06 times respectively compared with the smooth tube.

The rate of heat transfer is reported in Fig .4. It was found that for the increased Reynolds number, heat transfer rate increased for three inserts of 25mm, 40 mm and 80 mm in spacing between U-shapes by 4.00, 3.00, 2.00 times respectively compared with the smooth tube. As insert was used, the heat transfer rate was increased and more disturbances in flow causes more heat transfer.

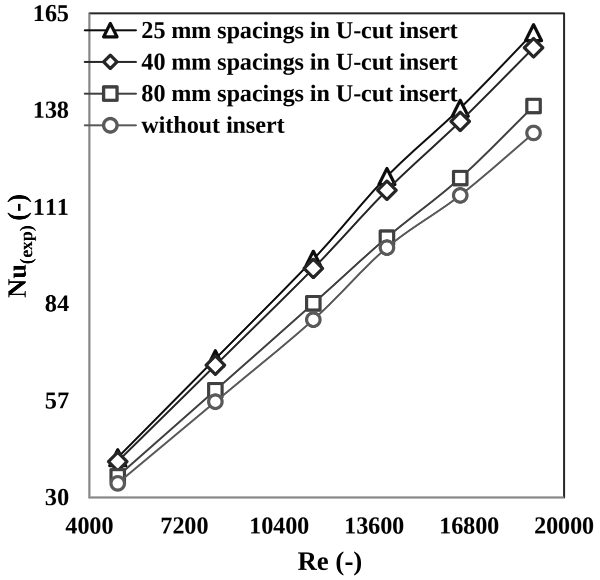


Fig. 3 Variation of Nusselt number with Reynolds number.

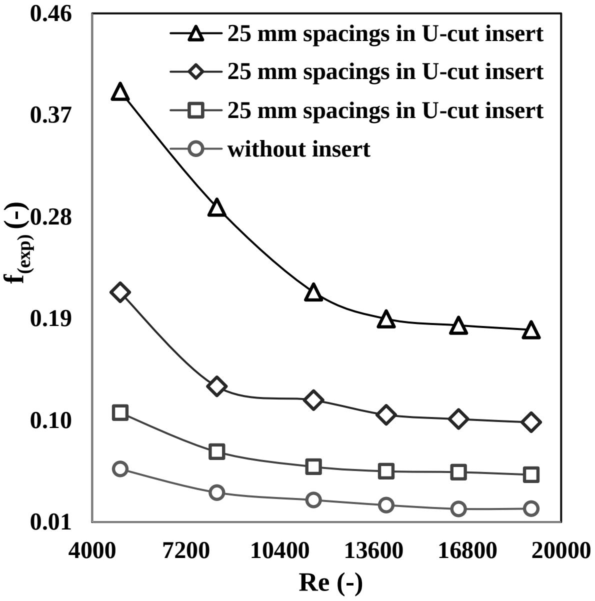


Fig. 5 Variation of friction factor with Reynolds number.

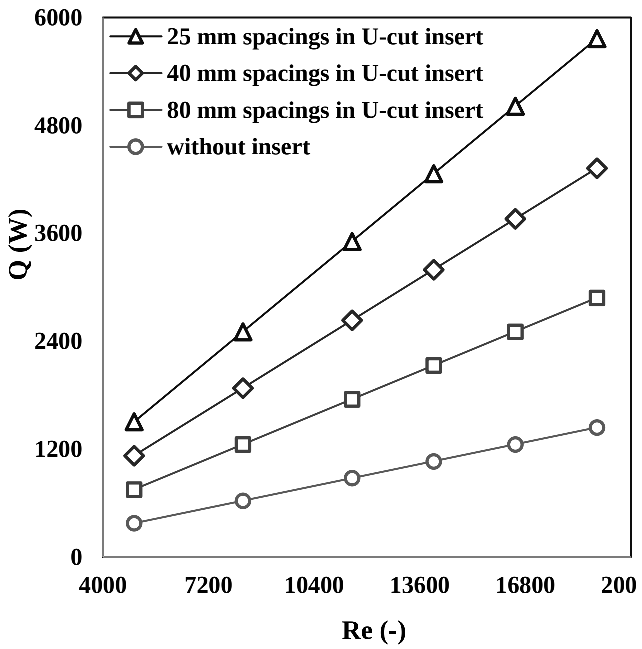


Fig. 4 Variation of heat transfer rate with Reynolds number.

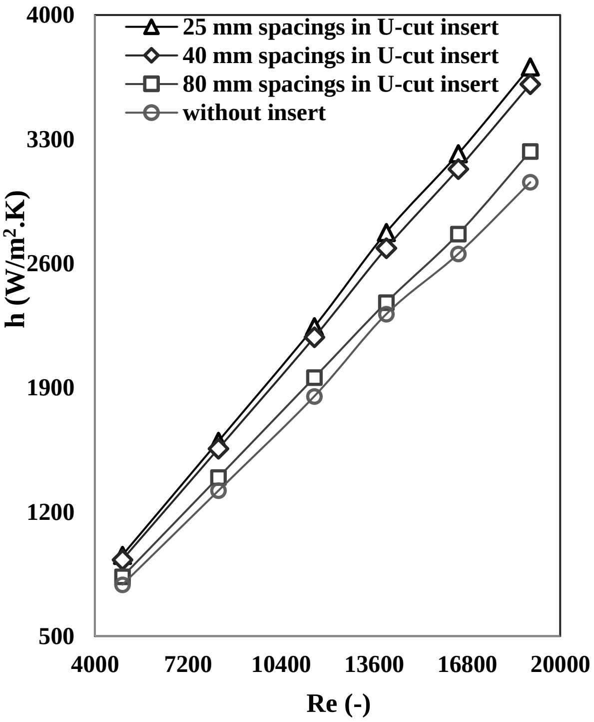


Fig. 6 Variation of convective heat transfer coefficient with Reynolds number.

From Fig. 5 it was found that friction factor decreased with an increase of Reynolds number for the three inserts than that of the smooth tube.

Fig. 6 shows the increment of convective heat transfer coefficient with the increase of Reynolds number. Heat transfer coefficient was detected 1.20, 1.18 and 1.06 times better than the smooth tube for the three different inserts. The resulting value of insert with 25 mm spacing showed maximum heat transfer coefficient.

Fig. 7, represents the variation of heat transfer enhancement efficiency for varying Reynolds number. It was found that for the increasing Reynolds number, heat transfer enhancement efficiency was increased.

Comparing with the smooth tube the increment found in the range of 1.57~1.67, 1.32~1.40, 1.17~1.27 times for three inserts of 25 mm, 40 mm and 80 mm in spacing between U-shapes respectively. But the efficiency with insert of 25 mm spacing was higher than the others as it provided more eddy during the flow of fluid.

So that, when inserts were used efficiency was increased and among the used inserts, 25 mm spacings in U-shapes twisted tape insert was better comparing with others.

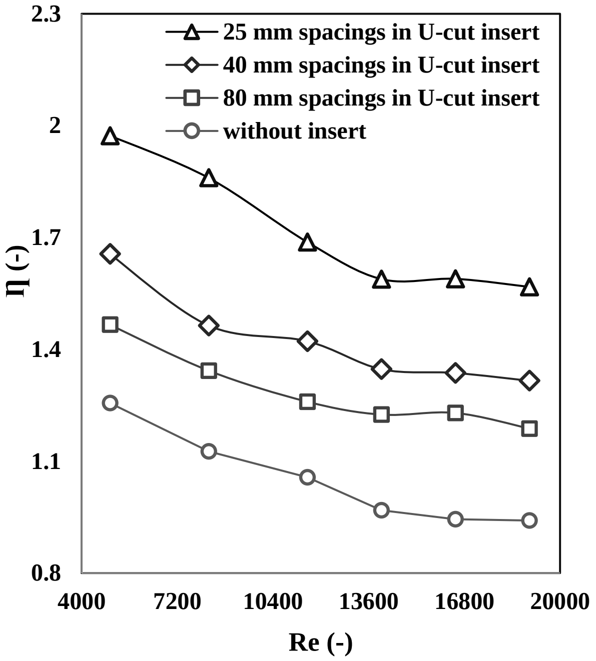


Fig. 7 Variation of heat transfer enhancement efficiency with Reynolds number.

5. CONCLUSIONS

Experimental exploration of heat transfer, friction factor and heat transfer enhancement efficiency of a circular tube with U-shaped twisted tape inserts with different spacings of 25 mm, 40 mm and 80 mm are hit off in the current narration. The results can be drawn as follows:

I. The Nusselt number and friction factor values for the tube with U-shape twisted tape inserts were noticeably higher than that of smooth tube and highest value found for 25 mm spacings in U-shapes insert. The values found 1.2 times better in Nusselt Number and 6.85 to 8.18 times improved in friction factor with respect to the smooth tube.

II. Over the range of Reynolds number, the heat transfer rate was also increased markedly for insert having 25 mm spacing in U-shapes. It increases 4 times better compared with the smooth tube.

III. Heat transfer coefficient was increased by 1.2, 1.18, 1.06 times improved for three different spacings in U-shape inserts of 25 mm, 40 mm, 80 mm than the smooth tube. And the former insert was obviously better than the others.

IV. Overall heat transfer enhancement efficiency was effectively increased for the three inserts likened with the smooth tube. Efficiency was increased 1.57 to 1.67, 1.32 to 1.40, 1.17 to 1.27 times respectively for 25 mm, 40 mm, 80 mm spacings in U-shape inserts. It is undoubtedly mark that, U-shape twisted insert of 25 mm spacings in U-shapes was the best among the others in the current experiment.

6. ACKNOWLEDGEMENT

Authors would like to thank Department of Mechanical Engineering of Chittagong University of Engineering and Technology for their continuous help and support for this research work. Authors also like to thank Heat Transfer lab for the help provided.

REFERENCES

[1] A. E. Bergles, W. M. Rohsenow, J. P. Hartnett and E. Ganie, “Techniques to Augment Heat Transfer”, Handbook of Heat Transfer Application, McGraw Hill, New York, NY, USA, 1985.

[2] B. Salam, S. Biswas, S. Saha and M. M. K. Bhuiya, “Heat transfer enhancement in a tube using rectangular-cut twisted tape insert.” Procedia Engineering, vol. 56, pp. 96–103, 2013.

[3] F. E. Megerlin, R. W. Murphy and A. E. Bergles, “Augmentation of heat transfer in tubes by use of mesh and brush inserts”, Journal of Heat Transfer, vol. 96, no. 2, pp. 145–151, 1974.

[4] S. N. Sarada, A. V. S. R. Raju, K. K. Radha and L. S. Sunder, “Enhancement of heat transfer using varying width twisted tape inserts”, International Journal of Engineering, Science and Technology, Vol. 2, No. 6, pp. 107-118, 2010.

[5] S. S. Hsieh and I. W. Huang, “Heat transfer and pressure drop of laminar flow in horizontal tubes with/without longitudinal inserts”, Journal of Heat Transfer, Vol. 122, No. 3, pp. 465-475, 2000.

[6] R. Thorsen and F. Landis, “Friction and heat transfer characteristics in turbulent swirl flow subjected to large transverse temperature gradients.” Journal of Heat Transfer, Vol. 90, No. 1, pp. 87-97, 1968.

[7] S. Eiamsa-ard, P. Nivesrangsan, S. Chokphoemphun and P. Promvonge, “Influence of combined non-uniform wire coil and twisted tape inserts on thermal performance characteristics”, International Communications in Heat and Mass Transfer, Vol. 37, No. 7, pp. 850–856, 2010.

[8] S. Eiamsa-ard, C. Thianpong, P. Eiamsa-ard and P. Promvonge, “Thermal characteristics in a heat exchanger tube fitted with dual twisted tape elements in tandem”, International Communications in Heat and Mass Transfer, Vol. 37, No.1, pp. 39-46, 2010.

[9] S. Eiamsa-ard, C. Thianpong, P. Eiamsa-ard. and P. Promvonge, “Convective heat transfer in a circular tube with short-length twisted tape insert”, International Communications in Heat and Mass Transfer, Vol. 36, No. 4, pp. 365-371, 2009.

[10] A. Altaie, M. R. Hasan and F. L. Rashid, “Numerical heat transfer and turbulent flow in a circular tube fitted with opened rings having square cross section”, Journal of Basic and Applied Scientific Research, Vol. 4, No. 11, pp. 28-36, 2014.

[11] M. A. Wazed, J. U. Ahamed, S. Ahmed and M. A. R. Sarkar, “Enhancement of heat transfer in turbulent flow

through a tube with a perforated twisted tape insert", Journal of Enhanced Heat Transfer, Vol. 18, No. 1, pp. 1–13, 2011.

[12] J. U. Ahamed, M. M. K. Bhuiya, R. Saidur, H. H. Masjuki, M. A. R. Sarkar, A. S. M. Sayem, and M. Islam, "Forced convection heat transfer performance of porous twisted tape insert", Engineering e-Transaction, Vol. 5, No. 2, pp. 67-79, 2010.

[13] F. W. Dittus and L. M. K. Boelter, Univ. Calif. (Berkely), Pub. Eng., Vol. 2, pp. 443, 1930.