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EXPERIMENTAL ANALYSIS FOR THE ENHANCEMENT OF HEAT TRANSFER IN A TUBE USING DOUBLE COUNTER TWISTED TAPE

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Abstract- *In this analysis heat transfer enhancement of water using double counter twisted tape has been carried out. Heat transfer coefficient, Nusselt Number, frictional coefficient have been calculated for both plain tube with and without inserts. The experimental set up consist of a long copper tube of 26.6 mm internal diameter and 30 mm outer diameter and effective length is 900 mm. The insert is 850 mm in length, 8.5 mm in width, 1.5 mm in thickness having twisted ratio 6.25. Bulk temperature and outer surface temperature are measured using thermometers at the inlet and outlet section of the circular tube and five K type thermocouples respectively. Definite amount of mass flow rate is measured by rotameter for calculation of water flow velocity. From manometer reading, pressure head is measured for determining pressure difference. For Reynolds Number 2960-5382 Nusselt Number, friction factor and heat transfer coefficient have increased upto 63%, 1.4-1.6 times and 58.57%-65% respectively comparing with plain tube using double counter insert.*

Keywords: *Heat transfer coefficient, friction factor, heat flux, Double counter twisted tape insert, Reynolds number, Nusselt number*

1. INTRODUCTION

Heat transfer enhancement is an important issue to increase the efficiency of modern thermal systems. From the passive way of heat transfer enhancement, using twisted tapes have been extensively reported due to compact structure and high thermal performance. Twisted tapes are commonly installed in a tube heat exchanger to promote the fluid mixing between central region and nearly the wall region. Shabanian et al. [1] showed that maximum Nu_{rs} had been gained with butterfly insert having an angle of 90^0 performing an empirical and numerical analysis of air cooled heat exchanger with inserts. There are three types of inserts were used in the analysis which were butterfly, classic and jagged twisted tapes. Wongcharee and Eiamsa [2] performed an experiment to show the character of heat transfer as well as pressure drop of turbulent flow using different types of inserts with a tube of heat exchanger. The inserts were mainly trapezoidal, rectangular and triangular.

Thianpong et al. [3] narrated that thermal performance of perforated twisted tapes with parallel wings, which are used in a tube, enhanced by 208% in comparison with plain tube. Eiamsa-ard et al. [4] investigated the performance of Nusselt number, friction factor of heat exchanger tube with different types of helical twisted tapes. The inserts were single, double and triple helical twisted tapes and showed that usage of double along with triple twisted tape insert enhanced Nu_{rs} by 15.6 to 23.4% as well as f_{rs} by 83 to 206%. Gunes et al. [5] did an experiment on thermal characteristics in a tube with inserts at different twist ratios. They used loose-fit twisted tape and performed the experiment with different hole diameter ratios. They showed that utmost Nu_{rs} was gained for having hole diameter ratio of 0.0714 and twist ratio of 2.0. Wen et al. [6] summarized and examined on single phase heat transfer enhancement techniques of liquid in pipes having corrugations, internal integral-fins, dimpled and twisted tape inserts tube showing that the heat transfer enhancement of experimental Nusselt number, in turbulent flow, over Dittus-Boelter equation for internal intrgral finned tube was in the range of 2-4; corrugated tube 1.5-4 twisted tape insert 1.5-6 along with dimpled tube 1.5-4. Eiamsa-ard et al. [7] analyzed the characteristics of heat transfer as well as pressure drop for a heat exchanger along with inserts. The inserts were opposite-parallel wing twisted tape inserts. Mwesigye et al. [8] used wall-detached twisted tapes to increase heat transfer rate and showed that heat transfer increases up to 169% due to the application of this types of arrangement. Vashistha et al. [9] found that counter twisted tape shows more Nu_{rs} and f_{rs}, which increase for decreasing the twist ratio, in the comparison with co-swirl. The experiment revealed that Nu_{rs} enhanced due to the reducing penetrability except for 1.3%. Skullong et al. [10]

showed that Nu_{rs} and f_{rs} for winglet perforated tapes enhanced due to the enhancement in blockage ratio as well as reduce in three winglet pitch ratio performing the study on heat transfer and turbulent flow friction in a tube having staggererd-winglet perforated tapes. Bhuiya et al. [11] observed the performance of heat transfer rate as well as pressure drop in turbulent flow which was flown through a circular tube with inserts. The inserts were perforated double counter twisted tape.

Zheng et al. [12] performed a numerical investigation on nano-fluid flow behavior through circular tubes. The tubes fitted with dimpled twisted tapes. They found that the turbulent kinetic energy level increases in the core flow region. Maximum enhancement of friction factor was 5.05% in this experiment. Saysroy and Eiamsa-ard [13] showed that the execution of heat transfer system withers due to the increasing the number of multi-channel twisted tapes in the range of 2 to 8 performing an experiment on heat transfer enrichment inside circular tube using multi-channel twisted tape inserts. Wen et al. [14] summarized and examined the heat transfer enhancement techniques of gas through internally finned, corrugated and dimpled, of total 436 pipes using twisted tape inserts. They found that the pressure-drop for gases, involving air, N_2, exhaust gases and helium, were higher than liquids. On the other hands the ratios of heat transfer enhancement were very similar to liquid. He et al. [15] performed an experiment on the thermal performance of water fluid through a circular tube which is framed with cross hollow twisted tapes. They revealed that due to the presence of cross hollow twisted tapes.
Nusselt number enhanced up to 120%. Heat transfer enhancement of water using double counter twisted tape insert is performed in this experiment. The double counter twisted tape inserts form swirl flow which increases turbulence thus leads to increase in Nusselt number rate.

2. EXPERIMENTAL SETUP

A long copper tube of 26.6 mm internal diameter and 30 mm outer diameter of which partial length will be used as the test section. A constant heat flux will be maintained by wrapping nichrome-wire around the test section and fiber glass insulation over the wire. Outer surface temperature will be measured by five k-thermocouple.

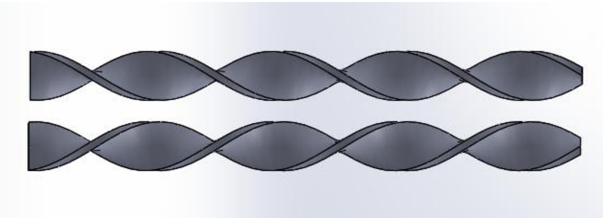

Fig.2.1. Double counter twisted tape

Pressure drop will be measured by manometer. Flow rate will be measured by rotameter. Inlet and outlet temperature of the fluids will be measured by thermometer. Adding all the thermocouples reading divided by five to calculate the outside surface temperature. Outside surface temperature deducted by wall temperature difference will call calculate inside surface temperature. Precaution should be taken to keep the double counter insert an accurate place. Precaution should be taken that not take touch on the nichrome-wire test section when it will be heated.

At first, water was stored in a tank and then supplied to the copper tube through Rota meter by a pump providing a gate valve, to control the flow rate. The flow rate of water was varied by the gate valve for different data and kept constant during the experiment. The hot water coming from the test section was flown to the drain through the mixing box. After switching on the heating power the sufficient time was given to attain the steady state condition. In each run, data were taken for water flow rate, water inlet, outlet, tube outer surface temperatures and pressure drop readings.

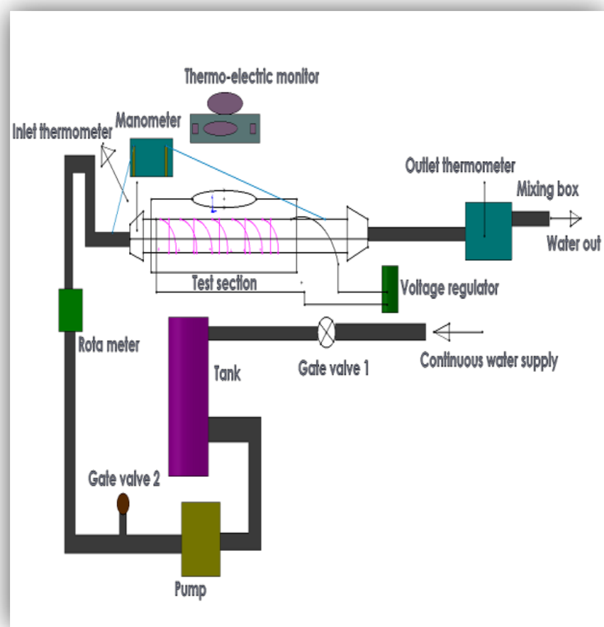

Fig.2.2. Experimental setup

3. MATHEMATICAL MODELING

Heat transfer rate by the heater to water was calculated by measuring heat added to the water. Heat added to water was calculated by

$$Q = mc_p(T_o - T_{in}) \quad (1)$$

where c_p is the specific heat of water, m is the mass flow rate of water, T_{in} and T_o are inlet and outlet temperature of water respectively.
Convective heat transfer co-efficient was calculated from

$$h = \frac{Q}{A(T_{wi} - T_b)} \quad (2)$$

where A is the surface area of the tube, T_{wi} and T_b are the tube inner surface temperature and fluid bulk temperature respectively.
Heat flux was found from

$$q = Q/A \quad (3)$$

Surface Area

$$A = \pi d_i L \quad (4)$$

where d_i and L are the tube inner diameter and length respectively.

The bulk temperature was obtained from the average of water inlet and outlet temperatures

$$T_b = (T_{in} + T_o)/2 \quad (5)$$

From the heat transfer equation of tube, tube inner surface temperature was calculated from

$$T_{wi} = T_{wo} - Q.\frac{\ln\left({}^{d_o}/_{d_i}\right)}{2\pi k_w L} \quad (6)$$

where T_{wo} is tube outer surface temperature, d_o is tube outside diameter and k_w is the thermal conductivity of the tube wall.

Tube outer surface temperature was calculated from the average of five local tube outer surface temperatures

$$T_{wo} = \sum_{i=1}^{5} \frac{T_T}{5} \quad (7)$$

where T_T is the thermocouple temperature.

Mass flow rate was calculated by

$$m = \rho A_x V \quad (8)$$

where A_x is the cross-section area of the test section, ρ is the density of water and V is the mean velocity of water.

Cross section area of test section determined from

$$A_x = \frac{\pi}{4} d_i^2 \quad (9)$$

The mean inlet velocity was measure by

$$V = \sqrt{2gh_r} \quad (10)$$

where h_r is rotameter reading in cm.

Theoretical Nusselt number was calculated from Gnielinski (1976), correlation,

$$Nu_{thG} = \frac{\left(\frac{f}{8}\right)(Re-1000)Pr}{1+12.7(f/8)^{1/2}(Pr^{2/3}-1)} \quad (11)$$

where f is friction factor, Re is Reynolds number and Pr is the prandlt number measured at bulk temperature of the water.

Where friction factor was calculated from first Petukhov (1970) equation,

$$f_{th} = (0.790 \ln Re - 1.64)^{-2} \quad (12)$$

Reynolds number measured from

$$Re = \rho V d_i/\mu \quad (13)$$

where ρ and μ are the density and dynamic viscosity of water determined in the fluid bulk temperature

Prandlt number calculated from

$$Pr = \mu C_p/k \quad (14)$$

where k is the thermal conductivity of water at bulk temperature.

Experimental Nusselt number determined from

$$Nu = hd_i/k \quad (15)$$

Nusselt number using Dittus-Boelter correlation

$$Nu_D = .023\, Re^{0.8} Pr^n \quad (16)$$

Where n= .4 for heating and .3 for cooling

The experimental friction factor calculated from,

$$f_{exp} = \frac{2\Delta P d_i}{\rho L V^2} \quad (17)$$

where ΔP ,the pressure is drop across the tapings and was calculated from

$$\Delta p = \Delta h \times \rho \times g \times 13.6 \quad (18)$$

where Δh is the U-tube manometer reading.

Percentage of error was calculated by using

$$\%\, of\ error = \{(Nu_{exp} - Nu_{th})/Nu_{th}\} \times 100 \quad (19)$$

4.RESULT AND DISCUSSION

4.1 Plain Tube Data Evaluation

In figure 4.1 and 4.2 variation of friction factor and Nusselt number with Reynolds number are shown. The theoretical data have been calculated from Gnielinski (11) and Petukhob (12) correlations for evaluating Nusselt number and friction factor respectively.

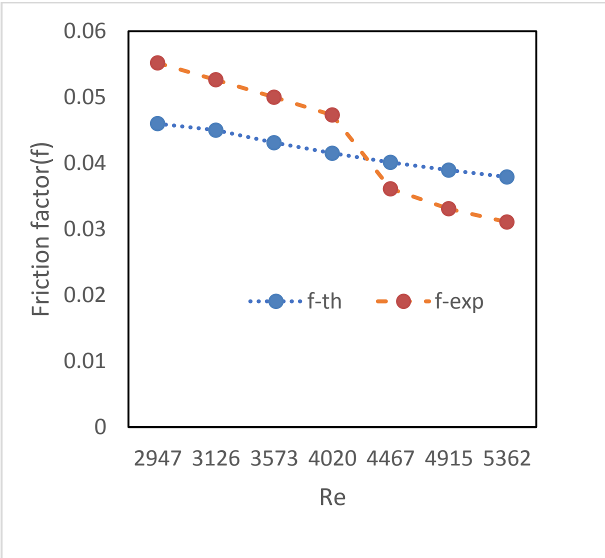

Fig.4.1. Variation of Friction factor with Reynolds number

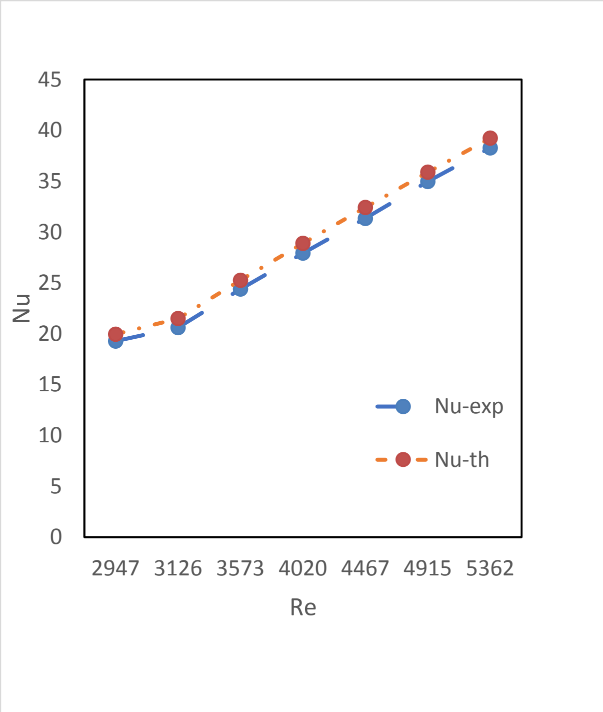

Fig.4.2. Variation of Nusselt number with Reynolds number

From figure 4.1 and 4.2 it is clear that friction factor decreases and Nusselt number increases with the increase of Reynolds number. The experimental values vary within 20% comparing to the theoretical values from Gnielinski and Petukhob correlations.

3.3 EFFECT OF DOUBLE COUNTER TWISTED TAPE INSERT

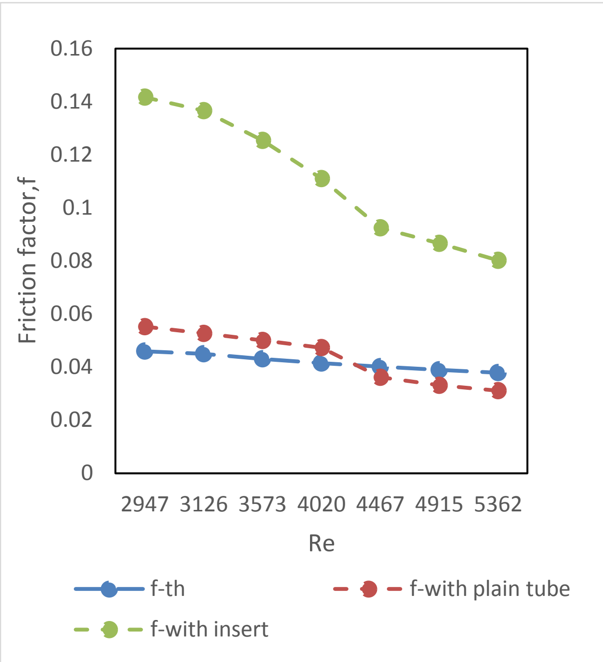

Fig.4.3. Variation of Friction factor with Reynolds number

Figure 4.3 shows that experimental friction factor decreases with increasing Reynolds number. It was found that where disturbance in water flow is greater, breaking down of the water film increase the pressure drop thus decreasing friction factor.

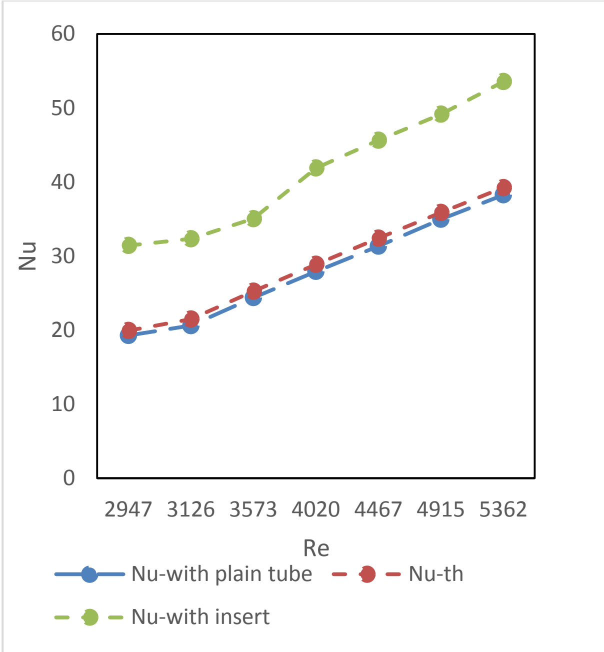

Fig.4.4. Variation of Nusselt number with Reynolds number

Figure 4.4 shows that Nu increasing with Reynolds number because more Reynolds number indicates more mixing which increases heat transfer coefficient. Thus greater value of h results in greater value of Nu.

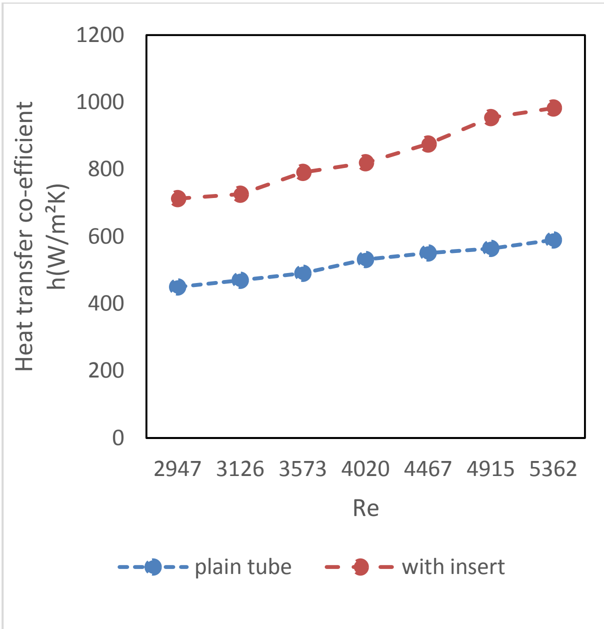

Fig.4.5. Variation of Heat transfer coefficient with Reynolds number

Figure 4.5 shows the average heat transfer coefficient increases with increasing Reynolds number for plain tube as well as the insert. At high Reynolds number mixing of fluid occurs and more heat is taken away from tube. So, temperature difference decreases but the heat transfer coefficient increases.

5. CONCLUSION

The measurement of tube side heat transfer coefficient is an important part of heat transfer related subjects. Some of the more successful enhancement techniques currently used for heat transfer augmentation have been reviewed here. Several active techniques have been identified as possibilities for heat transfer enhancement. These techniques do require external power. But there is a power cost that needs to be considered. There are also passive techniques that have been identified as possibilities for tube side heat transfer enhancement. Insertion of twisted tape into a tube provides a simple passive technique for enhancing the convective heat transfer by introducing swirl into the bulk flow. It may be guessed that the swirl flow helped in decreasing the boundary layer thickness.

- Using insert Nusselt number increased upto 63% respectively compared to the plain tube.
- Using insert friction factor increased upto 160% respectively compared to the plain tube.
- Using insert heat transfer coefficient increases by 58.57%-65% respectively comparing with plain tube.

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