EXPERIMENTAL STUDIES ON EFFECT OF WIRE COILED COIL MATRIX TURBULATORS WITH AND WITHOUT BONDING ON THE WALL OF THE TEST SECTION OF CONCENTRIC TUBE HEAT EXCHANGER

by

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Abstract

This paper presents the effect of bonding and without bonding of wire coiled coil matrix turbulator on the heat transfer for a fully developed turbulent flow. Experiments are conducted by maintaining constant wall temperature. Tests are performed on 3 different wire coiled coil matrix turbulators of different pitches of 5, 10 and 15 mm without bonding of the turbulator. Three similar types of heat exchangers are fabricated and the wire coiled coil matrix turbulators with different pitches of 5, 10 and 15mm are inserted in the heat exchangers and bonding is done on the surface of the tube section. Results have indicated that the heat transfer rate enhances inversely with the pitch of the wire coiled coil matrix turbulator with bonding. With a pitch of 5 mm, the turbulators without bonding have resulted in almost 25.4% enhancement when compared with plain tube. On the other hand, for pitches of 10 mm and 15 mm the enhancement were 20.7% and 16.8%, respectively. The empirical correlations developed for turbulators with and without bonding results in ±6% deviation for Nusselt number and ±3% for friction factor. Similarly with a pitch of 5 mm, the turbulators with bonding have resulted in almost 42% enhancement. For pitches of 10mm and 15mm the enhancements were 34.7% and 25%, respectively.

Key words: Turbulent flow; Wire coiled coil matrix turbulator; Pitch; Heat Transfer; Bonding

1. Introduction

Heat transfer can be increased by active and passive techniques. In the active techniques external power is required to increase the heat transfer. For the passive technique method no external energy is required for the enhancement of heat transfer. Wire coiled coil matrix turbulator (WCCMT) falls under the category of passive techniques. To the best of the knowledge of the authors, no research works have been done based on bonding of WCCMT on the wall of the test section of the concentric tube heat exchanger. In this experimental work, turbulators are used to increase the heat transfer. Three different types of turbulators (shown in figs.1a, 1b and 1c) are used to increase the heat transfer. Due to the insertion of turbulators there is increase in pumping power due to the pressure drop. But when compared to enhancement in heat transfer the increase in pumping power is very less. This method of enhancement techniques can be used in evaporators and condensers. Due to the increased heat transfer rate the heat exchanger can be manufactured in a compact size and hence the manufacturing cost can be reduced. This paper examines the effect of bonding of the turbulator for the heat transfer rate and also the pressure drop characteristics of the concentric tube heat exchanger under constant wall temperature boundary condition. All the experimental readings are taken for the turbulent flow condition.

Passive techniques are used in various industrial applications to increase the heat transfer. Gul and Evin [1] reported heat transfer and friction characteristics in decaying turbulent swirl flow generated by a short helical tape placed at the entrance of the test section. They concluded enhancement efficiency increases with increasing momentum ratio and decreases with increasing Reynolds number. Al-Fahed, Chamra and Chakroun [2] experimentally compared pressure drop and heat transfer coefficients for a plain, microfin and twisted-tape inserts tubes under isothermal boundary conditions. From the experimental results they concluded that tight fit tape gives a better performance over the loose-fit tape. Bergles [3] presented a comprehensive survey on heat transfer enhancement by various techniques. Smithberg and Landis [4] reported friction and forced convection heat transfer characteristics in tubes fitted with twisted tape swirl generators and presented a correlation for predicting Nusselt number and friction factor. Date [5] reported the prediction of fully developed flow in a tube containing a twisted tape. Hong and Bergles [6] correlated heat transfer and pressure drop data for twisted tape inserts for uniform wall temperature conditions using water and ethylene glycol as working fluids in laminar flow. Manglik and Bergles [7,8] reported experimental data for twisted tape. Agarwal and Raja Rao [9] reported heat transfer augmentation for the flow of a viscous liquid in circular tubes using twisted tapes inserts under constant wall temperature and presented a correlation for isothermal friction factor and Nusselt number. Naphon and Sriromruln [10,11] considered effect of coil wire insert on heat transfer enhancement and pressure drop of the horizontal concentric tubes and the concentric micro-fin tubes. Eiamsa-ard andPromvonge [12,13] studied on the heat transfer characteristics in a tube fitted with helical screw-tape with/without core-rod inserts and wire coil inserts. Sivashanmugam and Suresh [14] reported heat transfer and friction factor characteristics of turbulent flow through a circular tube fitted with regularly spaced helical screw-tape inserts has been reported.

WCCMT shown in figs. 1a, 1b and 1c is a modified form of a wire mesh insert wound on a single rod to create whirling motion. To the best of our knowledge no attempt has been made on the augmentation of heat transfer studies for a tube fitted with WCCMT with varying pitch. The present paper reports the heat transfer and friction factor characteristics of turbulent flow through a circular tube fitted with WCCMT insert of various pitches with and without bonding, with water as working fluid.

Fig.1a.Wire coiled coil matrix turbulator of pitch 5mm (Tu1)

Fig.1b.Wire coiled coil matrix turbulator of pitch 10mm (Tu2)

Fig.1c.Wire coiled coil matrix turbulator of pitch 15mm (Tu3)

2. Details of WCCMT without bonding

The WCCMT insert with three different pitches is made by winding uniformly a copper wire of diameter 0.7 mm on a 5 mm copper rod. With the help of wire finning machine 15 loops per turn of the wire coil are made. For the turbulator of pitch 5 mm, 295 turns of the wire coil are made over the 5 mm copper rod of length 1500 mm. After the winding, soldering is done between the wire coil and the rod in order to avoid air gap between them. So that entrapment of water between the wire coil and copper rod is avoided. The same procedure is repeated to manufacture the turbulator of pitches 10 mm and 15 mm. For the turbulator of pitch 10 mm, 148 turns of the wire coil are made over the 5 mm copper rod of length 1500 mm. Similarly, for the turbulator of pitch 15 mm, 95 turns of the wire coil are made over the 5mm copper rod of length 1500 mm. The fabrication details of the WCCMT are clearly explained in the figs.2a-2d. The WCCMT different configuration details are given in tab.1.

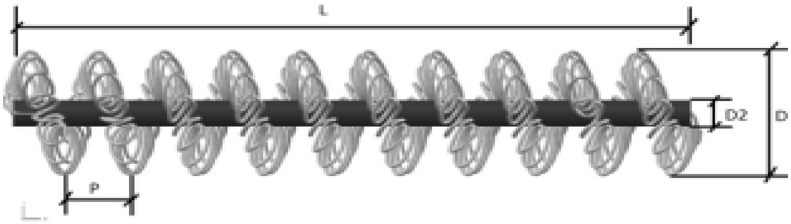


Fig.2a.Details of Wire coiled coil matrix turbulator

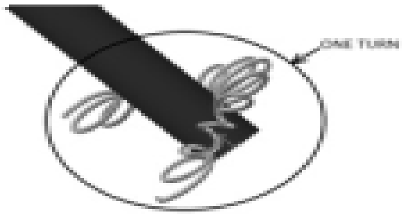


Fig.2b.Details of one turn of the Wire coiled coil matrix turbulator

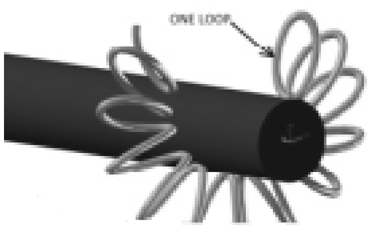


Fig.2c.Details of one loop of the Wire coiled coil matrix turbulator

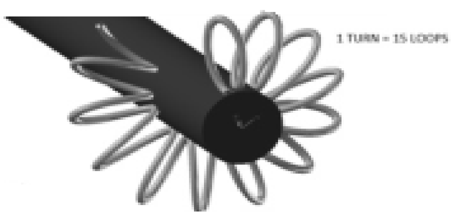


Fig.2d.Details of one turn of the Wire coiled coil matrix turbulator

Table.1 Wire coiled coil matrix turbulator configuration details.

Configuration	Length (L) (mm)	Pitch (P) (mm)	Diameter of the WCCMT (D1) (mm)	Diameter of the shaft (D2) (mm)	Number of turns
Tu1	1500	5	22	5	295
Tu2	1500	10	22	5	148
Tu3	1500	15	22	5	95

2.1 Details of WCCMT with bonding

The WCCMT of pitch 5mm is inserted into the inner tube of diameter 22.2mm (inner diameter) of the concentric tube heat exchanger. Along with the turbulator soldering wire is inserted inside the test tube. Then the test tube is immersed in the flux and heated in a furnace for about 600˚C. When the tube is maintained at this temperature the solder wire gets melted between the coil of the turbulator and the inner surface of the test tube. Then the test tube is taken from furnace and allowed for cooling, at this point the melted solder wire solidifies and bonding takes place between the coil of the turbulator and the inner surface of the test section. After this process the inner tube of the test section is inserted into the outer tube of diameter 50mm and the both ends of the heat exchanger is closed with cover plate. All the thermocouples are positioned thus the concentric tube heat exchanger is fabricated for the WCCMT pitch of 5mm. The same procedure is repeated to fabricate the concentric tube heat exchanger for turbulator of pitch 10mm and 15mm.

3. Experimental set-up

The schematic diagram of the experimental set up is shown in fig. 3a. It consists of calming section, test section, outlet mixing section, supply tank, water collection tank and rotameters. The capacity of supply tank and water collection tank are of each 1m^3. Calming section with the following dimensions is made: Length: 1650 mm; Inner diameter (ID): 22.2 mm; Outer diameter (OD): 25.4 mm. This section made of mild steel tube is used to eliminate the entrance effect and to ensure fully developed flow. The test section consists of a single tube-in-tube heat exchanger. The test section consists of a smooth copper inner tube with an OD of 25.4 mm and a wall thickness of 1.6 mm, outer tube with an OD of 50 mm and a wall thickness of 2 mm. The test section has a nominal length of 1500 mm. Saturated steam is introduced in the annulus of the test section and is used as an isothermal source. Water introduced in the test tube is used as the working fluid. All the exposed parts of the system are insulated by asbestos rope in order to avoid the heat transfer from the heat exchanger to the surroundings.

Two thermocouples are installed across each test section to measure the inlet and outlet temperatures of the water. Also, three thermocouples are inserted in the annulus of test section to monitor the steam temperature. The wall temperature of test section is monitored by 9 thermocouples installed on the outside of the tube wall (see fig. 3b). Such thermocouples are installed in three axial locations along the wall of the tube. At each location, three thermocouples are installed at 120° apart around the circumference of the tube. Each thermocouple is soldered in a groove on the outer side wall of the inner tube. In order to avoid direct exposure to the steam in the annulus after soldering, the grooves are covered with a 5 mm steel tube which brazed on the inner and outer tubes of the heat exchangers shown in fig.3b.The thermocouple wires are taken out of the annulus through this 5 mm steel tube at each of these axial locations. All the thermocouples are calibrated chromel-alumel of 0.1°C accuracy with digital indicator. One end of test section is attached with the calming section while the other end is attached with the mixing section (length 600 mm, ID 22.2 mm, OD 25.4 mm),where two baffles are provided inside the pipe at a distance of 100 mm from the flange for efficient mixing of outlet fluid. In order to avoid the heat conduction from the test section to the calming section and mixing section a 50.9 mm thick non-conducting polypropylene disc is placed in-between the connecting flanges.

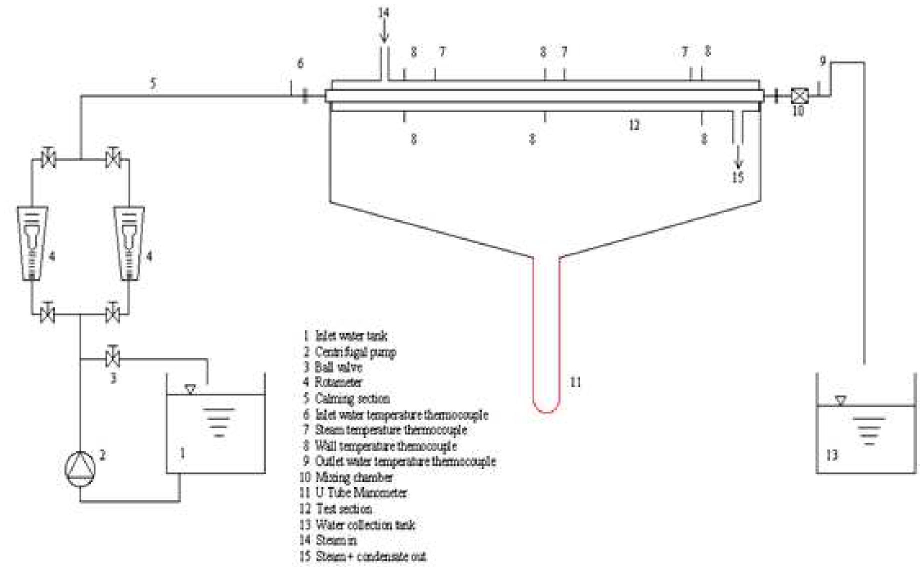
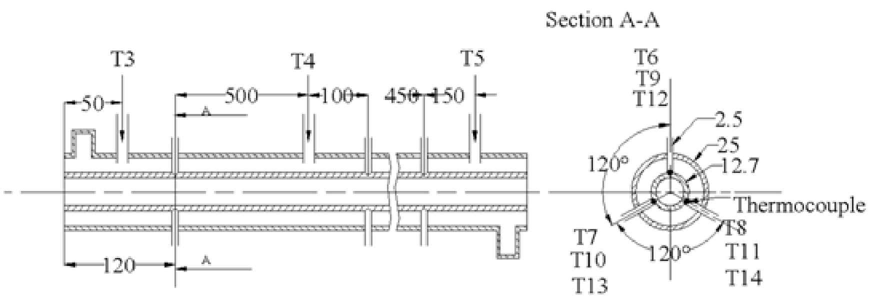


Fig.3a.

Experimental setup



T3, T4 and T5 – Thermocouples used to measure the saturation temperature of steam
T6, T7, T8, T9, T10, T11, T12, T13 and T14 – Thermocouples used to measure the wall temperature

All dimensions are in mm

Fig.3b.Thermocouples location

Two calibrated rotameters having flow ranges of 0.1 to 4×10^{-3} m^3/min and 4 to 16×10^{-3} m^3/min are attached to the calming section to measure the flow. The water at constant temperature is being taken from the inlet tank through centrifugal pump as shown in Fig. 3a. The by-pass valve attached to rotameter is used to regulate the flow rate to the test section. The two pressure taps, one at entry to the test section and the other at the exit of the test section, are provided and connected to U tube manometer for pressure drop measurement under isothermal condition.

4. Experimental procedure for WCCMT without bonding

After the leakage check of experimental setup, the water is sent into the inner tube of the tube-in-tube heat exchanger by centrifugal pump and the steam produced by a diesel fired boiler, goes into the annulus of the test section. The steam pressure is set to 0.2 MNm^{-2} to maintain constant wall temperature boundary condition and drain is opened to evacuate steam and condensate in the annulus. When the pressure drop of the manometer reaches steady state, the pressure drop reading, the temperature of water at inlet, outlet and the tube wall temperatures are recorded. Then the water flow rate is set at a different value and the test is repeated. After all the tests are completed for a given pitch of WCCMT, the WCCMT is changed and tested with other pitches.

4.1. Experimental procedure for WCCMT with bonding

Heat exchanger with bonded turbulator of pitch 5mm is positioned in the supporting stand and then the calming section and mixing section are connected with the heat exchanger. Pressure tap connection provided at the two ends of the test section. After the above mentioned procedure the experimental setup is ready to take readings. Experimental procedure is similar as explained for the WCCMT without bonding. After taking all the readings for the heat exchanger with turbulator of pitch 5mm, the heat exchanger is changed and tested with other two heat exchangers for turbulator of pitch 10 and 15mm.

5. Experimental Uncertainty

In this paper, estimation of uncertainty is done based on Moffat [15].The uncertainty for a single measurement on the experimentally calculated result, for only that one measurement can be found using eq. (1)

$$\partial R_{Xi} = \frac{\partial R}{\partial X_i} \int X_i \tag{1}$$

When more independent variables are used in the function R, the individual terms are combined by root-sum-square method (eq. (2))

$$\partial R = \left\{ \sum_{i=1}^{N} \left(\frac{\partial R}{\partial X_i} \, \partial X_i \right)^2 \right\}^{1/2} \tag{2}$$

The following equation is used to consider the relative errors in the individual factors denoted by x_n.

$$w(k, h, D, f, v, \ldots) = [(x_1)^2 + (x_{21})^2 + \cdots + (x_n)^2]^{1/2} \tag{3}$$

Reynolds Number uncertainties can be calculated by combinations of eqs.(4) and (5). Eq.(6) is the resultant equation after combining eqs. (4) and (5)

$$Re = \frac{VD}{\upsilon} \tag{4}$$

$$w_{Re} = \left[\left(\frac{\partial Re}{\partial V} w_{\upsilon}\right)^2 + \left(\frac{\partial Re}{\partial D} w_D\right)^2 + \left(\frac{\partial Re}{\partial \upsilon} w_{\upsilon}\right)^2\right]^{1/2} \tag{5}$$

$$\frac{w_{Re}}{Re} = \left[\left(\frac{w_V}{V}\right)^2 + \left(\frac{w_D}{D}\right)^2 + \left(\frac{w_{\upsilon}}{\upsilon}\right)^2\right]^{1/2} \tag{6}$$

Similarly Nusselt Number uncertainties can be calculated by combinations of eqs. (7) and (8)

$$Nu = \frac{hD}{k} \tag{7}$$

$$w_{Nu} = \left[\left(\frac{\partial Nu}{\partial h} w_h\right)^2 + \left(\frac{\partial Nu}{\partial D} w_D\right)^2 + \left(\frac{\partial Nu}{\partial k} w_k\right)^2\right]^{1/2} \tag{8}$$

$$\frac{w_{Nu}}{Nu} = \left[\left(\frac{w_h}{h}\right)^2 + \left(\frac{w_D}{D}\right)^2 + \left(\frac{w_k}{k}\right)^2\right]^{1/2} \tag{9}$$

Each of the measured physical properties consists of non dimensional parameters. The uncertainties of the non dimensional parameters for each of the measured physical properties are given in tab.2. Maximum values of uncertainty calculations for Re, Nu, and f are 7.3%, 9.8% and 13.2% respectively.

Table. 2

Uncertainties values for the relevant variable

Variable	Uncertainty (%)
Water density,ρ	1.4
Specific heat capacity of water,C_p	2.9
Diameter,D	1.2
Thermal conductivity of water,k	2.1
Dynamic viscosity of water,μ	2.6
Water flow rate,m	4.2
Pressure drop,ΔP	4.4

5.1. Friction factor calculation

The pressure drop is determined from the differences in the level of manometer fluid. The fully developed friction factor is calculated from the following equation.

$$\mathbf{f} = \frac{2\Delta P D_i^{\ 2}}{\rho u_m} L \tag{10}$$

Where ΔP is the pressure drop over length L.

6. Data reduction equations

The average inside heat transfer coefficient and the mean Nusselt number for the plain and the WCCMT cases are evaluated as follows:

$$\mathbf{Q} = \dot{m}C_p(T_o - T_i) = \boldsymbol{h}_i A_i (\Delta \mathbf{T_i})_\mathbf{m} \tag{11}$$

Where

$$\mathrm{Ai} = \pi D_i L \tag{12}$$

$$(\Delta \mathbf{T_i})_\mathbf{m} = \frac{(\overline{\mathbf{T}}_\mathbf{w} - \mathbf{T_i}) - (\overline{\mathbf{T}}_\mathbf{w} - \mathbf{T_o})}{\ln\left[\dfrac{\overline{\mathbf{T}}_\mathbf{w} - \mathbf{T_i}}{\overline{\mathbf{T}}_\mathbf{w} - \mathbf{T_o}}\right]} \tag{13}$$

and

$$\overline{T}_w = T_w/9 \tag{14}$$

T_w is the local wall temperature of the tube.

It must be noted that the local wall temperature T_w is measured within the tube wall (at a position of 0.5 mm from the inner wall). Analysis has demonstrated that such a measurement technique for the wall temperature resulted with a maximum error of 2% in the Nusselt number because of negligible wall resistance.The average inside heat transfer coefficient and the mean Nusselt number are determined by:

$$\boldsymbol{h}_i = \frac{\mathbf{Q}}{\mathbf{Ai}(\Delta \mathbf{T_i})_\mathbf{m}} \tag{15}$$

$$Nu = \boldsymbol{h}_i D_i / k \tag{16}$$

All properties are evaluated at the bulk mean temperature.

7. Results and discussion

7.1. Plain tube data

The present experimental results on heat transfer and friction characteristics in a plain tube are first validated in terms of Nusselt number and friction factor. It is important to compare the experimental results obtained for the fully developed turbulent flow with the correlations from the literature. The Nusselt number and friction factor obtained from experiment on the plain tube are compared with the correlations of Sieder and Tate [16], Petukhov [17] and Blassius correlations respectively found in the open literature for turbulent flow in circular tubes.

Sieder and Tate correlation,

$$Nu = 0.027 Pr^{0.33} Re^{0.8} \left(\mu_b/\mu_w\right)^{0.14} \tag{17}$$

Petukhov correlation,

$$NU = \frac{\left(f/_{(8)} Re^{Pr}\right)}{1.07 + 12.7\sqrt{f/8\left(Pr^{2/3}\right) - 1}} \left(\frac{\mu_b}{\mu_w}\right)^{0.11} \tag{18}$$

where f is the friction factor and for plain tube it is given as [17]

$$f = (1.82log_{10}(Re) - 1.64)^{-2} \quad (19)$$

Blassius correlation

$$f = 0.3164(Re)^{-0.25} \quad (20)$$

Fig. 4 shows variation of Nusselt number obtained from experiment and Nusselt number estimated using Sieder and Tate with Reynolds number for the case of plain tube.

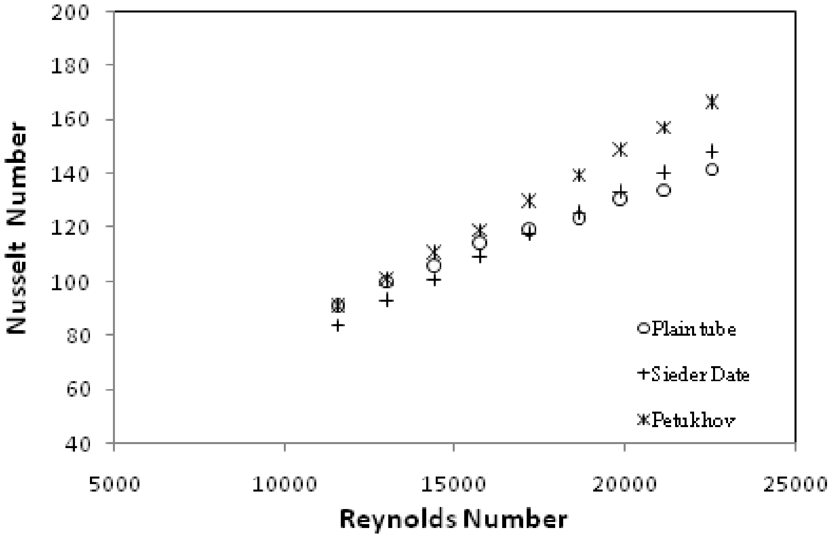


Fig.4. Data verification of Nusselt number for plain tube.

It is observed from fig.4 that the Nusselt number estimated from experimental data lies within ±15% that of theoretical values calculated using Sieder and Tate and Petukhov correlation.

Plain tube experimental correlation for Nusselt number and friction factor are obtained as

$$Nu = 0.228\ Re^{0.641} \quad (21)$$

$$f = 2.248\ Re^{-0.45} \quad (22)$$

Fig.5 shows variation of friction factor with Reynolds number. The experimental data are matching with the Blassius and Petukhov correlation for plain tube with a discrepancy of less than ±7%.

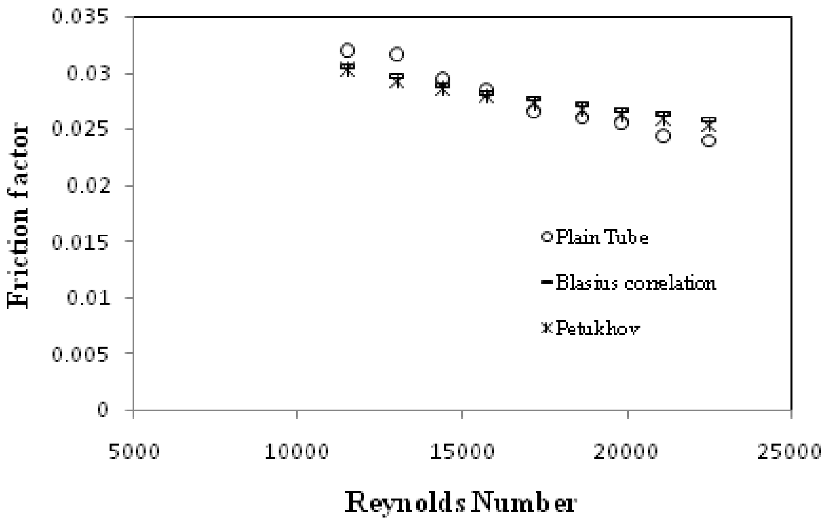


Fig.5. Data verification of Friction factor for plain tube.

7.2. Effect of pitch on heat transfer enhancement without bonding

Fig.6 shows the variation of Nusselt number with Reynolds number for the tube fitted with WCCMT inserts of three different pitches (5,10,15 mm). From fig.6 it can be concluded that the Nusselt number for the tube fitted with WCCMT inserts are higher than that of plain tube for a given Reynolds number. This is because the WCCMT interrupts the development of the boundary layer of the fluid flow near the wall of the test section. Hence it increases the average temperature of the fluid in the radial direction. Due to the larger contact surface area the heat transfer rate increases. Also it creates the turbulence and whirling motion to the water which is flowing inside the test section. The whirling makes the flow to be highly turbulent, which leads to improved convection heat transfer.

As the Reynolds number increases for a given pitch, the Nusselt number also increases, indicating enhanced heat transfer coefficient. It is also observed from fig.6 that Nusselt number for a given Reynolds number increases with decreasing pitch of the coil. As the pitch of the coil decreases, the intensity of swirl flow increases leading to higher heat transfer rate and the maximum being for the WCCMT of pitch 5 mm. Throughout the experimental results, it is seen that the smaller pitch (5 mm) yields the higher values of heat transfer of about 25.4% than plain tube. Similarly for pitches of 10 mm and 15 mm the enhancement are 20.7% and 16.8% respectively.

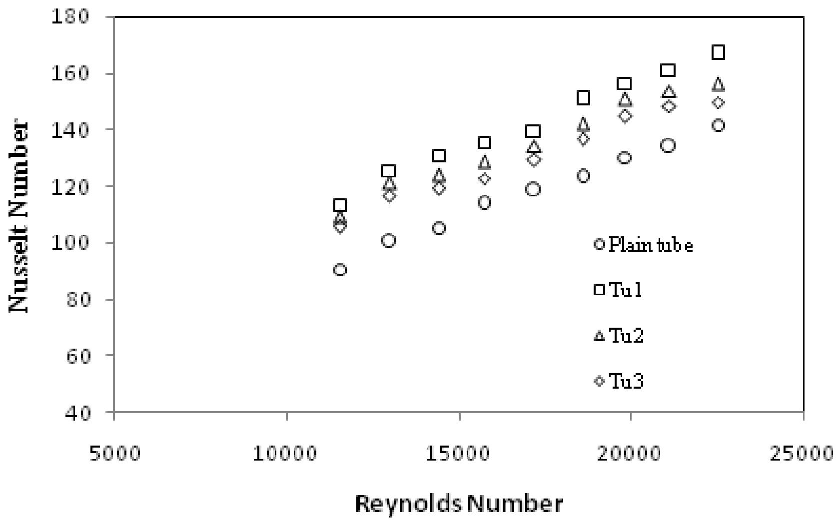


Fig. 6. Nusselt number vs. Reynolds number for wire coiled coil matrix turbulator of different pitches without bonding.

7.3. Effect of pitch on friction factor without bonding

Generally, the friction factor decreases conventionally with the increasing Reynolds number for different pitches. From fig. 7 it can be seen that friction factor for the tube fitted with WCCMT inserts is higher for a given Reynolds number. It indicates that friction factor for a given Reynolds number increases with the decreasing pitch due to swirl flow generated by WCCMT and reaches the maximum for the pitch of 5 mm. From fig.7 it can be seen that the friction factor is less when compared with turbulator of pitch 5mm.This is due to the less contact surface area of the turbulator, because the number of turns of the coil is only 148, so more area is available for the water to flow in the test section.

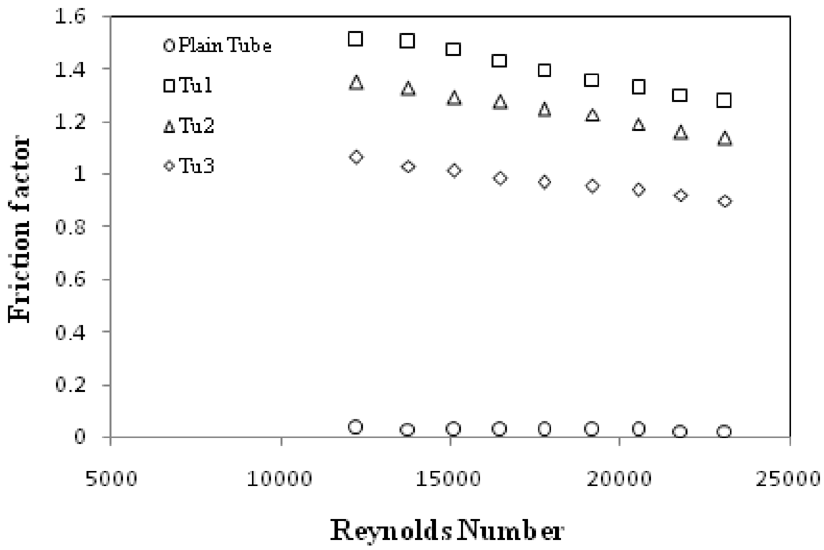


Fig.7. Friction factor vs. Reynolds number for wire coiled coil matrix turbulator of different pitches without bonding.

Hence the pressure drop for the turbulator of pitch 10mm is less when compared to turbulator of pitch 5mm. Similarly there is more area for the water to flow in the test section when the turbulator of pitch 15mm is used in the heat exchanger. This is because only 95 turns are provided for this turbulator. Hence the friction factor for the turbulator of pitch 15mm is less when compared with pitch 5 and 10mm.

7.4. Effect of pitch on heat transfer enhancement with bonding

From fig.8, it can be seen that the Nusselt number increases with increase in Reynolds number. From fig. 9 it can be seen that friction factor for the tube fitted with WCCMT inserts with bonding is higher for a given Reynolds number when compared with plain tube.

The increase in heat transfer is because of two reasons, first due to the turbulence and swirl motion created by the turbulator to the water which is flowing inside the tube. Second is due to the bonding of the turbulator with the inner surface of the tube. When the steam is allowed in the annulus region of the concentric tube heat exchanger, the outer surface of the inner tube of the heat exchanger picks the heat and the heat is conducted through the wall of the inner tube. The turbulator coil which is bonded with the inner surface picks heat and the heat will conduct through the coil of the turbulator to the centre rod of the turbulator. Hence the water core which is flowing in the center receives heat due to the bonding effect.

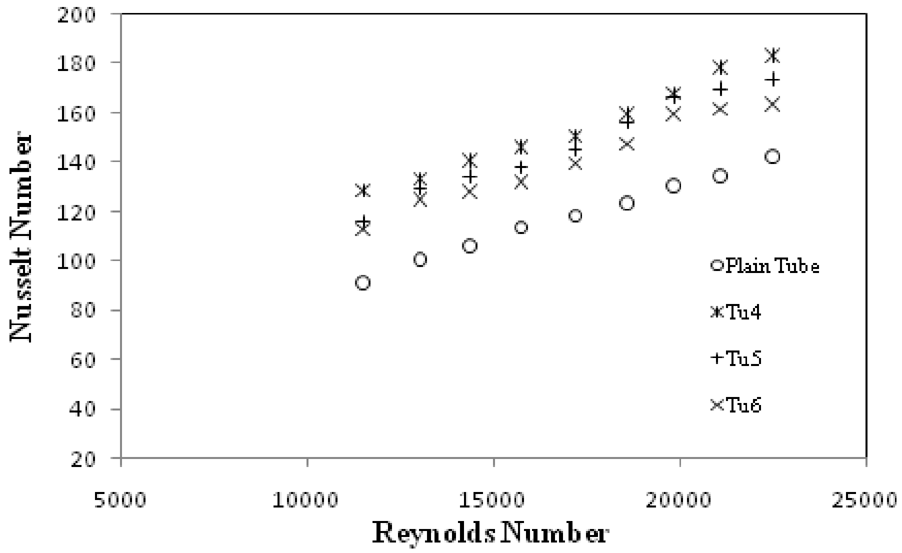


Fig. 8. Nusselt number vs. Reynolds number for wire coiled coil matrix turbulator of different pitches with bonding.

Due to the above mentioned reasons the heat transfer rate is high when compared with the turbulator without bonding which can be seen from the fig.10. This is due to the air gap between the turbulator and the inner surface of the tube in the case of turbulator without bonding. But in the case of turbulator with bonding the air gap is filled with soldering, so there is metal to metal contact, hence the heat is transferred both by conduction and convection. From fig.11 it can be seen that there is only marginal increase in friction factor for turbulator with bonding when compared with turbulator without bonding.

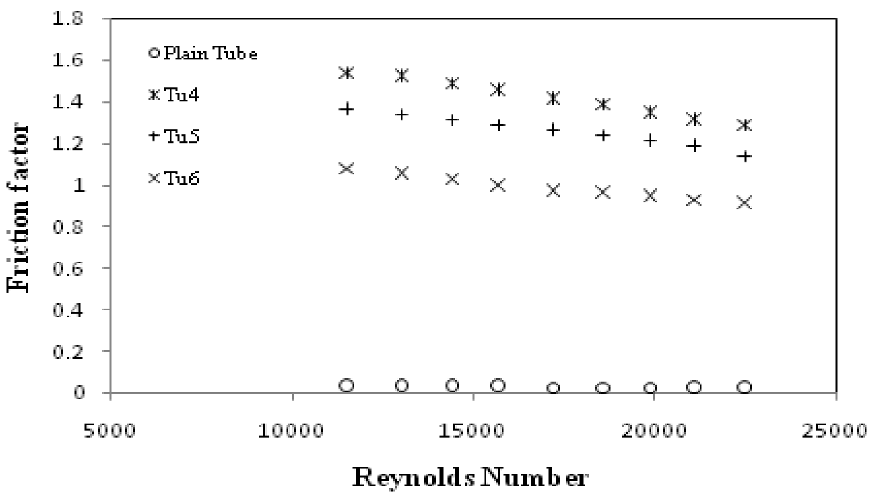


Fig.9. Friction factor vs. Reynolds number for wire coiled coil matrix turbulator of different pitches with bonding.

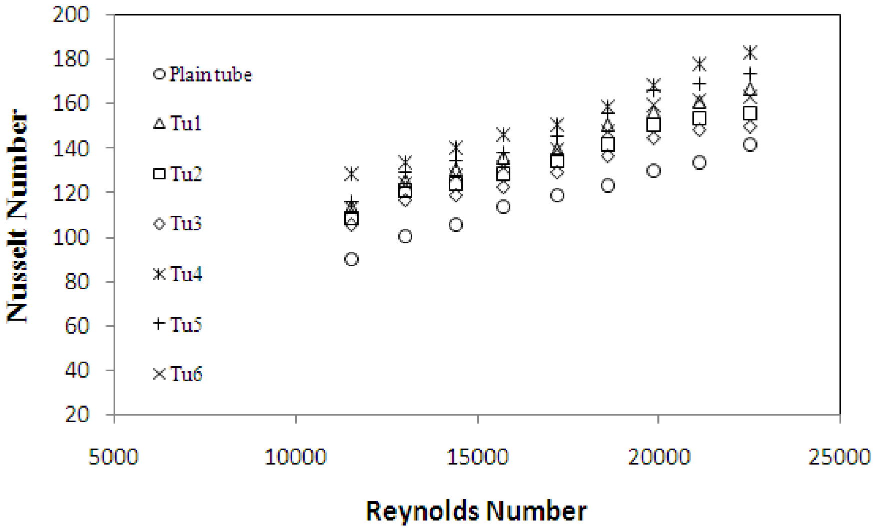


Fig.10. Comparison of experimental Nusselt number for the entire turbulator configuration.

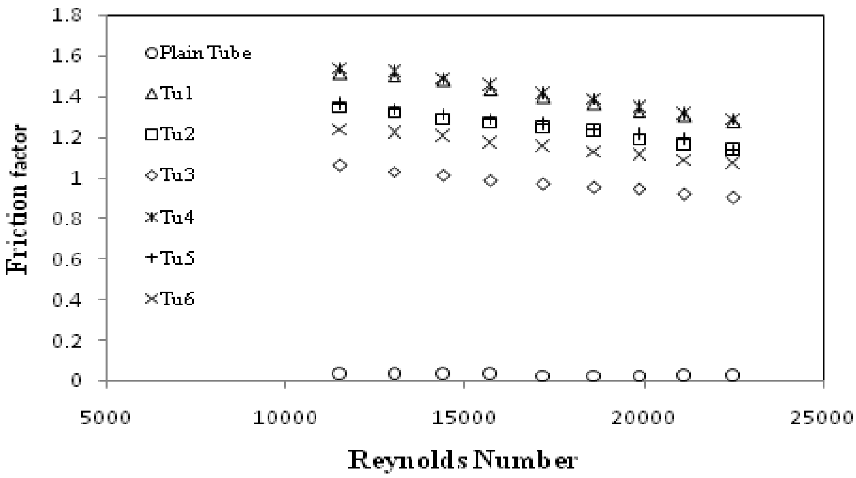


Fig.11. Comparison of experimental friction factor for the entire turbulator configuration.

7.5. Empirical correlation for WCCMT with different pitches without bonding

The data were fitted by the following empirical correlations

$$Nu = 0.613\, Re^{0.57}\, p^{-0.071} \tag{23}$$

$$f = 19.46\, Re^{-0.244}\, p^{-0.071} \tag{24}$$

The fitted values of Nusselt number by eq. (23) and friction factor by eq. (24) are compared with the experimental values and are shown in figs.12 and 13 respectively. The fitted values coincide with experimental data within ±3, and ±4%, respectively, for Nusselt number and friction factor.

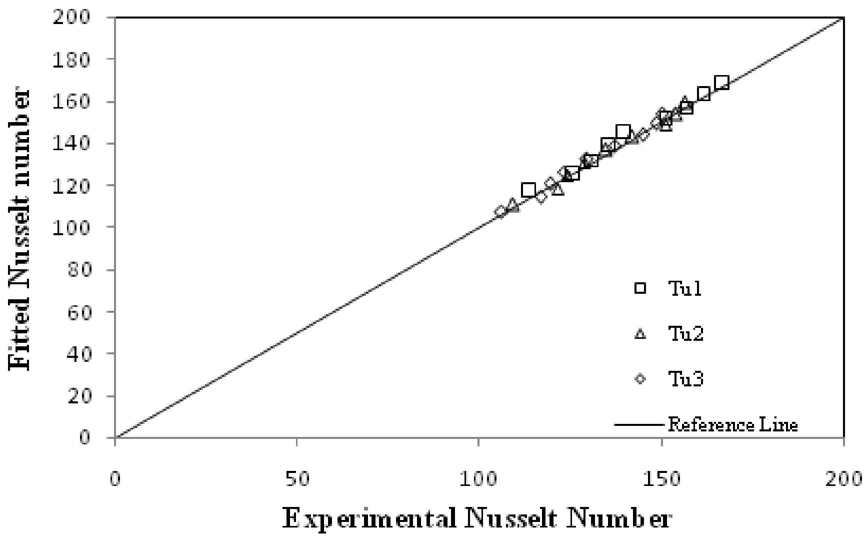


Fig.12. Comparison of Experimental Nusselt number with the fitted value without bonding.

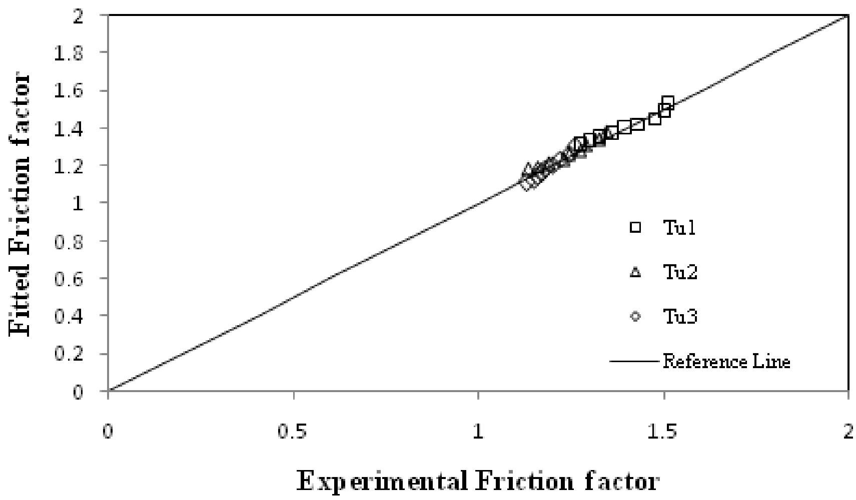


Fig.13. Comparison of Experimental friction factor with the fitted value without bonding

7.5. Empirical correlation for WCCMT with different pitches with bonding

The data were fitted by the following empirical correlations

$$Nu = 0.459\, Re^{0.606}\ p^{-0.073} \tag{25}$$

$$f = 26.43\, Re^{-0.268}\ p^{-0.189} \tag{26}$$

The fitted values of Nusselt number by Eq. (25) and friction factor by Eq. (26) are compared with the experimental values and are shown in Figs.14 and 15 respectively. The fitted values coincide with experimental data within ±6, and ±3%, respectively, for Nusselt number and friction factor.

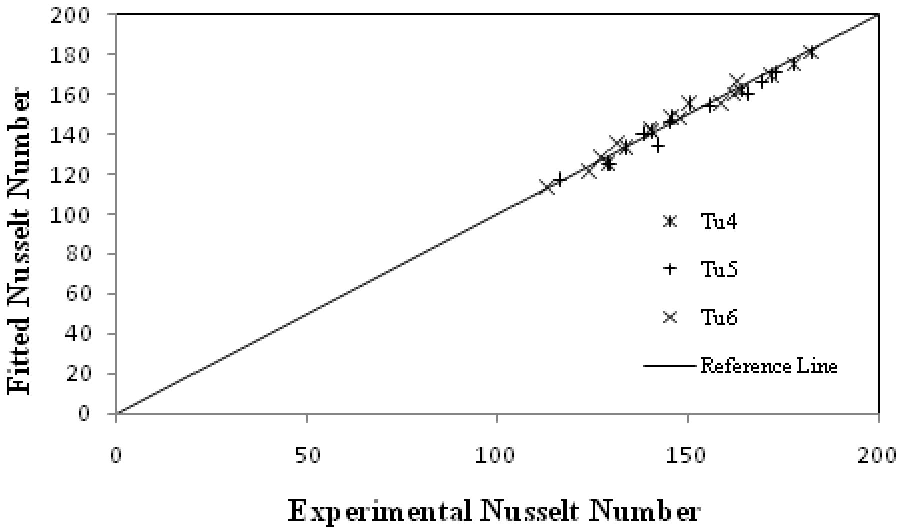


Fig.14. Comparison of Experimental Nusselt number with the fitted value with bonding

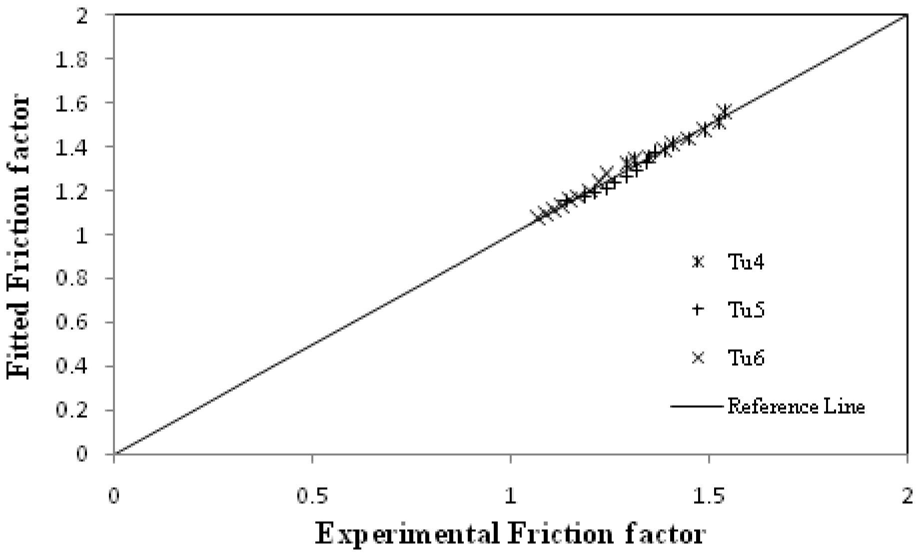


Fig.15. Comparison of Experimental friction factor with the fitted value with bonding

8. Conclusion

For practical application of thermal systems operating under turbulent flow conditions WCCMT are desirable in order to increase the heat transfer rate. Experimental investigations of heat transfer and friction factor characteristics of a circular tube fitted with full-length WCCMT of different pitches have been studied for the turbulent regime, Re=10000 -23 000 and the following conclusions were drawn.

(1) Experimental data obtained were compared with those obtained from the theoretical data of plain tube.

(2) The maximum Nusselt number for pitch 5 mm was obtained which indicates that heat transfer coefficient increases with the decreasing pitch with bonding of the turbulator.

(3) The friction factor also increases with the decreasing pitch.

(4) The empirical correlations developed relating pitch and Reynolds number were matching with the experimental data within ±6, and ±13% for Nusselt number and friction factor respectively.

(6) The above findings indicate that the use of wire coiled coil matrix turbulator in the tube-in-tube heat exchanger enhances the heat transfer with considerable pressure drop.

Appendix A. Nomenclature

A_i	inside surface area of test section area [m^2]
C_p	specific heat at constant pressure [KJ kg^{-1} $^{\circ}C^{-1}$]
D_i	inside diameter of test section [mm]
f	friction factor
h_i	average convective heat transfer coefficient [Wm^{-2} $^{\circ}C^{-1}$]
k	thermal conductivity of fluid [Wm^{-1} $^{\circ}C^{-1}$]
L	length of the test section [m]
Nu	Nusselt number $Nu = hiDi/k$
Re	Reynolds number based on internal diameter of the tube
ΔP	pressure drop of fluid (Nm^{-2}]
Q	heat transfer rate [KW]
$\overline{T}_w$	average wall surface temperature in the test section [$^{\circ}C$]
T_w	local wall temperature [$^{\circ}C$]
T_i	inlet temperature of fluid [$^{\circ}C$]
T_o	outlet temperature of fluid [$^{\circ}C$]
T_b	bulk mean temperature of water [$^{\circ}C$]
U_m	bulk average fluid velocity [ms^{-1}]
P	Pitch of the wire coil matrix turbulator [mm]
m	mass flow rate of water [kgs^{-1}]

Greek letters

ρ	density of fluid (kg/m^3)
μ_b	Viscosity at bulk mean temperature of fluid [Nsm^{-2}]
μ_w	Viscosity at average wall surface temperature [Nsm^{-2}]

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