Evaluation Of Thermo Hydraulic Performance Of Passive And Compound Inserts

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In the present investigation, the heat transfer and pressure drop performance of a tube in tube heat exchanger fitted with Bakelite helical screw tape and aluminum hemispherical cup shape with threaded core rod insert is experimentally reported for turbulent flow in individual and compound insertion cases. Performance is evaluated using water in the Reynolds number range of 8000 to 32000. For helical screw tape, the average Nusselt number ratios for augmented tube case to plain tube case (Nu_a/Nu_p) were found to be ranging between 2.51 – 2.7 along with average friction factor ratios (f_a/f_p) ranging between 4.12 – 4.13. For cup shape inserts, the average Nusselt number ratios for augmented tube case to plain tube case (Nu_a/Nu_p) were reported between 1.38 – 1.62 along with friction factor ratios (f_a/f_p) ranging between 4.93 – 5.44 . The average Nusselt number ratios (Nu_a/Nu_p) and friction factor ratios (f_a/f_p) for two compound insertion cases are also reported by varying the cross-section of inserts for 1/3rd length of the heat exchanger and observed in the range of 2.26 – 2.65, and 4.06 – 4.8 respectively. For equal pumping power criteria, the average performance ratios (Nu_a/Nu_c) are reported in the range of 1.02 -1.11 for helical screw tape, 0.6 – 0.67 for cup shape insert and 0.99 – 1.22 for two compound insertion cases.

Keywords: compound inserts, twisted rod insert, turbulent flow, Average performance ratio

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1. Introduction

Many engineering applications typically use heat exchangers. Numerous engineering methods have been developed during the past few decades to improve heat transfer in heat exchangers. Using turbulator elements to improve fluid turbulence and, consequently, the heat transfer coefficient from the flow surface is one of these strategies. This method has been widely used in various heat exchanger applications, including refrigeration, automotive, process industries, solar water heaters, etc. since it has been shown to lower heat exchanger sizes while saving energy. Heat transfer enhancement techniques involving a use of turbulator elements can be broadly classified into active, passive, and compound type. Active heat transfer techniques involve the use of external power, whilst passive techniques focus on creating a swirl in the fluid to disrupt the boundary layer and increase heat transfer rates. Compound technique approaches combined active and/or passive strategies to improve heat transfer.

In the present work, the performance of two separate passive inserts in terms of pressure drop and heat transfer has been reported using passive and compound approaches along with a brief review of work done by different researchers on existing individual and compound inserts.

Kumar et al. [1] reviewed the work related to heat transfer enhancement using inserts and elaborated mechanisms of fluid flow sustained by different types of inserts like twisted tapes, coiled wires etc. Traditional twisted tapes are widely used for heat transfer enhancement as they promote heat transfer with less fluid pressure drop. The heat transfer enhancement with twisted tapes significantly depends on a pitch and twist ratio.

Eiamsa-Ard et al. [2] experimentally compared the thermo hydraulic performance of a single and dual twisted tape with Reynolds number ranging from 4000 to 19000 and recorded enhanced heat transferred for dual twisted tapes over single twisted tapes along with Nusselt number enhancement up to 146% and friction factor rise up to 256% over a plain tube. Mehdi Bahiraei et al. [3] performed an experimental study to find the geometry effects of twisted conical strip inserts on a Nusselt number and friction coefficient. The study noted that the Nusselt number and friction coefficient were increased for all inserted cases and decreased with twist angle and pitch ratio. Also, a greater Nusselt number increment was observed at lower Reynolds numbers. Abolarin et al. [4] performed an experimental investigation to study the effect of an angle of connection for clockwise as well as counter clockwise twisted tapes on a thermo hydraulic performance in a laminar, transition and turbulent flow regimes. It is reported that the increment in connection angle improved the heat transfer performance in transitional flow regime and the increment in heat flux improved the heat transfer performance in laminar flow regime. Sarviya and Fuskele [5] performed experimental work to propose a new turbulator named continuously cut edges type twisted tape insert. The experimental data of Nusselt number and friction factor is collected for a new turbulator using two different twist ratios of 3 and 5. Based on the results, the authors recommended the use of a proposed insert noting higher heat transfer rates with a reasonable increase of pressure drop. Suri et al. [6] examined the heat transfer and friction of a tube inserted with multiple square perforated twisted tape-types inserts with a perforation width ratio range of 0.083-0.333 and a twist ratio range of 2 – 3.5. The Authors reported considerable heat transfer enhancement for the arrangement. The maximum augmentation is seen at a perforation width ratio of 0.250 and a twist ratio of 2.5.

Many researchers have tested coiled wires as another low-cost passive insert for heat transfer enhancement. Wires can be manufactured with simple processes and are easy to install and remove from the tubes. Dang and Wang [7] studied the heat transfer rate in a tube fitted with twined coil insert experimentally and numerically. It is seen that the insert geometry increases secondary flow intensity resulting in enhanced heat transfer performance compared to a plain tube under similar operating conditions. Promvonge [8] tested coiled wires with a square cross sections between Reynolds numbers 5000 to 25000 using air and recorded a rapid decrement in heat transfer augmentation at increased Reynolds numbers. Swirl generators are passive inserts that impart a spiral motion to the flow also called tangential or azimuthal velocity component.

Zhang et al. [9] tested parallel type and V-shaped type rectangular winglet vortex generators and reported increments in heat transfer and flow friction for parallel type in the range of 54–118% and 152–568% over a plain tube, and the values obtained for V- type are 60–118% and 141–644%, respectively. Yang et al. [10] observed monotonical increments in Nusselt number with Reynolds number and a number of swirls in their investigation of convergent pipe fitted with a pre-swirl device at entry with air as a working medium between Reynolds numbers 7970 to 47820. Eiamsa-Ard and Promvonge [11] studied the effect of helical screw tape with and without a core rod on heat transfer and friction factor in double pipe heat exchanger for a loose fit arrangement. Authors reported average Nusselt number enhancement of 230% and 340% over a plain tube respectively for screw tape with and without core rod. Depaiwa et al. [12] tested rectangular winglet vortex generators in the Reynolds number range of 5000 to 23000 with air as a working medium and observed that solar air heater channel with rectangular winglet vortex generators delivers a heat transfer rate and friction loss that are much higher than those of a channel with a smooth wall. Hasan et al. [13] have experimentally studied the effect of a novel vortex generator insert having ten crescent holes at an equal circumferential angles on heat transfer in the Reynolds number range of 6000 to 13500 with air as a working medium and reported a maximum heat transfer enhancement of 4.3 times over the plain tube along with a friction factor rise of 1.28 times over plain tube.

Different inserts like conical nozzle, converging-diverging conical rings, conical-nozzle turbulators, V-nozzle turbulators are categorized into conical rings group of inserts. Chingtuaythong et al. [14] presented the effect of V-shaped rings on thermo hydraulic performance of a circular tube heated with constant heat flux using air as a working medium. The reported results reveal that the selected insert increases the heat transfer and flow friction up to 5.8 and 82 times respectively for the Reynolds number between 5000 to 25000. Muthusamy et al. [15] performed experiments with divergent and convergent conical inserts for airflow with Reynolds numbers ranging between 6800 to 9700. They obtained a maximum heat transfer rate, thermal performance factor, and friction factor of 315%, 2.4 and

3.2 times respectively for diverging arrangement and 225%, 1.9 and 2.8 times respectively for converging arrangement than that of a plain tube. Keklikcioglu and Ozceyhan [16] reported the influence of convergent, convergent-divergent, and divergent conical wire coils on heat transfer using ethylene glycol and water of three different mixture volumes as a working fluid between the Reynolds number range of 4627 to 25,099. It is seen that both heat transfer and frictional resistance have been increased for all wire coils. The addition of ethylene glycol in water reduced the heat transfer rate and increased the frictional resistance slightly. Promvonge and Eiamsa-Ard [17] experimentally investigated the heat transfer for a circular tube inserted with snail entry type V nozzle turbulators between the Reynolds number 8000 to 18000 and observed that V nozzle turbulators perform better alone than in combination.

Ribs are another heat transfer enhancement tools that changes flow the effective cross-sectional area of flow through different shapes and treated surfaces. Nanan et al. [18] reported the effect of various baffle tabulators arrangement on a heat transfer performance of a tube between the Reynolds numbers 6000 to 20000. Among selected types, the heat transfer increment along with the thermal performance factor were found to be maximum for twisted cross baffles arrangement while minimum values were reported for straight cross baffles arrangement. The highest thermal performance factor of 1.7 was reported for twisted cross baffles arrangement. Tandiroglu [19] experimentally investigated the heat transfer and flow friction characteristics of turbulent flow with baffle plate inserts in the Reynolds number range of 3000 to 20000 and proposed empirical correlations for Nusselt number as well as for friction factor in terms of Reynolds number and parameters of flow geometry.

Some researchers have performed numerical studies and suggested suitable methods for an analysis of the heat exchangers using inserts. Gong et al. [20] proposed a numerical method to evaluate the overall heat transfer performance of a parabolic trough receiver tube fitted with pin fin arrays using a finite volume method coupled to the Monte Carlo ray tracing method. The numerical results indicated the Nusselt number and heat transfer performance factor enhancement of 9% and 12% respectively for the proposed insertion. Göksu et al. [21] performed a numerical investigation of double and triple wire coil inserts by modeling inserted tube arrangement in Ansys Workbench and solving it in the Ansys Fluent. Results revealed that the proposed tool enhances the heat transfer and performance evaluation criteria up to 5 times and 103% respectively over a plain tube. It is reported that the performance evaluation criteria up to 1.1 and 1.14 could be obtained for double wire coil and triple wire coil respectively.

Recently few researchers have investigated the suitability of nano synthesized phase change materials for heat transfer augmentation. Ho et al. [22] have experimentally investigated the convective heat transfer performance of a Nano-Encapsulated Phase Change Material (NEPCM) water suspension in a divergent eight mini channel heat sink having a divergent angle of 1.380. It is seen that NEPCM water performed better compared to pure water at low Reynolds number and lower heating inputs but was not found advantageous at higher Reynolds numbers. Ho et al. [23] have investigated the impact of synthesized Nano-Encapsulated Phase Change Material (NEPCM) nanoparticles of 250 -350 nm size on heat transfer and the index of performance in an eight rectangular micro channel heat sink. The results reveal that the heat transfer enhancement is notable at low Reynolds numbers and reduces at higher Reynolds numbers. Along with individual inserts, many researchers have also studied the combined use of different inserts to enhance heat transfer in compound insertion.

Pimsarn et al. [24] optimized the aero thermal performance factor for a tube fitted with inclined circular rings and twisted tape compound inserts. The authors evaluated the impact of various factors, including the number of twisted tapes and twist ratios of a circular ring on heat transfer and pressure drop performance. The heat transfer enhancement of 128.7% and 155.3% is reported for inclined circular ring and twisted tape over plain tube respectively. Thianpong et al. [25] examined the effectiveness of compound insertion utilizing a dimpled tube equipped with a twisted tape swirl generator for twist ratios of 3, 5, and 7 in the Reynolds number range of 12000 to 44000 with water serving as the working fluid. Results demonstrated that the compound enhancement approach greatly improved heat transmission and friction factor. Lower pitch and twist ratios were also associated with increased heat transfer and friction rates. Marzouk et al. [26] evaluated the performance of shell and tube heat exchangers using wired nails circular rod inserts and air bubble injection simultaneously and reported a significant enhancement in the range of 31 to 184% for various selected sizes of inserts.

Pourahmad et al. [27] studied the effect of the simultaneous use of active and passive techniques. Air bubbles were used as an active technique along with a double-twisted tape as a passive tool. Authors observed Nusselt number enhancement for simultaneous case in the range of 98–114%, 3–14%, and 20–39% over a plain tube case, a turbulator inserted case without bubble injection, and a plain tube with bubble insertion case respectively. Eiamsa-Ard et al. [28] experimentally studied the heat transfer of a tube using twisted tape and wire coils with constant as well as periodically varying pitch ratios and observed a better performance for coils with periodically varying pitch ratios over constant pitch ratios.

All compound inserts which are discussed in the literature have undergone consistent insert cross-sections along the length of the test section. This study suggests varied cross-section inserts along the length of the test section for heat transfer enhancement. In the present work, a new hemispherical cup with a threaded core rod and a helical screw tape having core rod inserts are proposed. The inserts are supposed to disturb the laminar-sub boundary layer due to sudden obstacles in the fluid flow field and impart additional swirl to the fluid.

2. Experimental setup and methodologies

2.1. Details of Insert

2.1.1. Cup cross-section with threaded core rod insert

For the proposed cup shape inserts, three hemispherical cups are mounted on a central core rod having threads for a locking nut. An arrangement helped to alter the pitch of the insert as per the requirement. Core rods having threads are prepared in three segments having an equal lengths of 1/3 of the overall test section length. With such an arrangement, the insert could be used with any other insert type for studying thermo hydraulic performance of compound insertion. Three aluminum cups with the outer diameter of 17.9 mm, thickness of 3 mm, and core rod diameter of 5.8 mm were prepared with a pitch of 0.15 m for cup 1, 0.2 m for cup 2, and 0.25 m for cup 3. The details of the cup inserts prepared are given in Table 1.

The geometric details of cup shape inserts along with pictures are shown in Fig. 1 and Fig. 2 respectively.

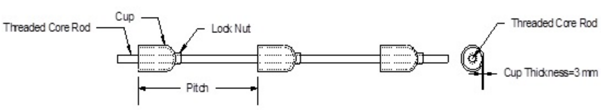

Fig. 1. Geometric details of Cup with threaded core rod insert

2.1.2. Helical screw tape with core rod inserts

Helical screw tape is prepared on wood turning lathe. The material used is Bakelite for its lesser thermal conductivity (1.4 W/m K). The outer diameter of the screw is 17.55 mm and core rod diameter is kept at 8 mm. The pitch of the screw is kept at 33.9 mm. These are also manufactured in three segments and threaded portions are kept at the end for intermediate connection. The geometric details of the selected helical screw tape along with the picture are shown in Fig. 3 and Fig. 4 respectively.

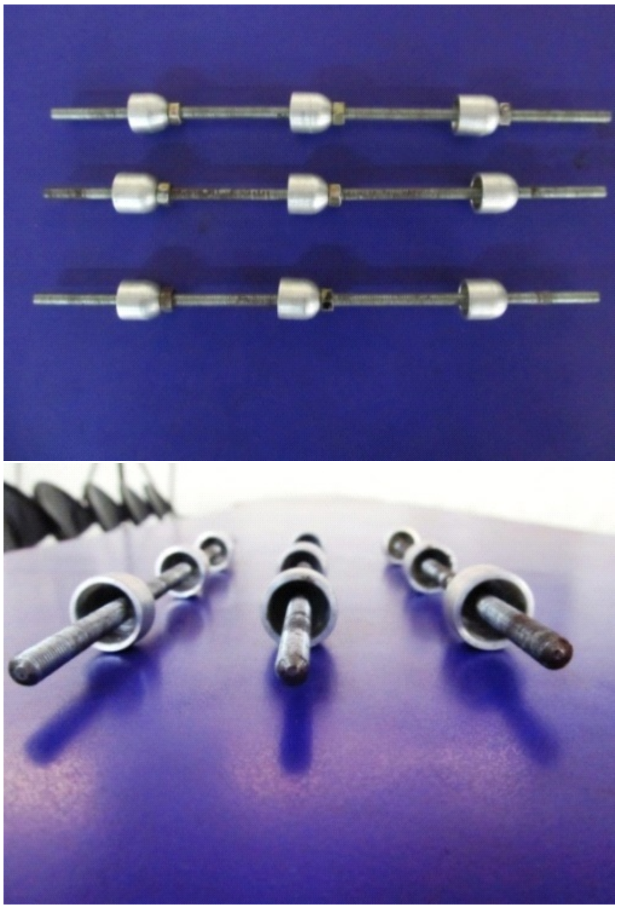

Fig. 2. Photographs of cup shape inserts

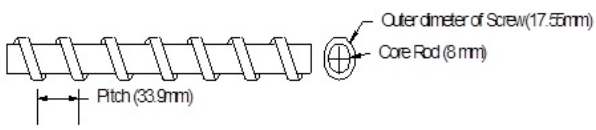

Fig. 3. Geometric details of helical screw tape with threaded core rod insert

2.1.3. Compound insertion cases

Two compound insertion cases are studied. In case 1 (C1), cup shape insert is placed in the initial and final 1/3 test section length and helical screw is kept in the middle section shown in Fig. 5. In case 2 (C2), a helical screw is placed in initial and final 1/3 test section length and a cup shape insert is kept in the middle section shown in Fig. 6.

Table 1. Details of hemispherical cup insertss

Insert	Material	Cup outer diameter	Cup thickness	Core rod diameter	Pitch
Cup 1	Aluminium	17.9 mm	3 mm	5.8 mm	0.15 m
Cup 2	Aluminium	17.9 mm	3 mm	5.8 mm	0.2 m
Cup 3	Aluminium	17.9 mm	3 mm	5.8 mm	0.25 m

Fig. 4. Photograph of helical screw tape inserts

Fig. 5. Picture of compound insert case 1 (C1)

Fig. 6. Picture of compound insert case 2 (C2)

2.2. Experimental test setup details

Fig. 7 shows the details of experimental setup. The details of setup and test conditions are also listed in Table 2.

Test runs were conducted on a circular double-pipe heat exchanger. Outer steel and inner copper pipes of thickness 5.85 mm and 3.65 mm with inner diameters of 48.3 mm and 21.4 mm respectively were selected for heat exchanger. A 100-lit hot water tank fitted with two heaters of 2 kW capacities each were used for hot water supply. Hot water at 75°C temperature was allowed to flow through the annulus using a circulation pump in a closed loop. A gate valve was used for maintaining the required flow rate of hot water. A

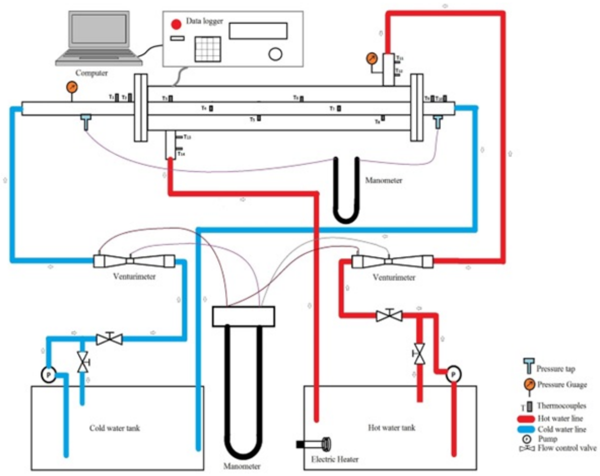

Fig. 7. Schematic of the Experimental setup [29]

Venturimeter was fitted in the circulation line for the measurement of a flow rate. A cold water circulation pump was used for the inner tube (test section) cold water circulation. For maintaining the accurate flow rate, two gate valves and a single bye-pass valve was used in the cold water circulation cycle. Flow rate measurement for cold water was also done using a Venturimeter located in the cold water line. To quantify the pressure drop over the test portion, a U-tube manometer filled with a carbon tetra-chloride (CCL 4) manometric fluid was utilized. A data acquisition system fitted with pre-calibrated T type Cu-Constantant thermocouples was used to collect temperature data. Two thermocouples were used for temperature data collection at the inlet and outlet of both hot and cold fluids. Six thermocouples placed equidistant along the length of the tube were used to calculate the average temperature of the test section surface wall.

2.3. Experimental Procedure

Before the conduction of trials with inserts, the experimental results for a plain tube were required to be validated with well-established equations to get confidence in the experimental work. Hence, the trials were conducted initially on a plain tube for validation of the experimental set up between the Reynolds number ranges of 8000 to 32000 with Reynolds number increasing by a step of 4000. After plain tube validation, trials were repeated for 4 individual

Table 2. Details of setup and test conditions

(A) Experimental Setup		
i)	Type of Heat Exchanger	Tube in tube heat exchanger
ii)	Inner tube inner diameter	21.4mm
iii)	Inner tube Thickness	3.65 mm
iv)	Outer tube inner diameter	48.3mm
v)	Outer tube thickness	5.85mm
vi)	Length of test section	For heat transfer 0.8m, For pressure drop 0.99m
vii)	Manometric fluid	Carbon Tetrachloride, CCL_4 (density 1577 $\mathrm{kg/m^3}$)
(B) Test Conditions		
i)	Cold water inlet temperature	25°C
ii)	Hot water inlet temperature	75°C
iii)	Range of Reynolds number, (Re)	8000 to 32000

inserts and 2 compound insert cases. During trials, mass flow rates of hot and cold fluids, temperatures and pressure drop for plain tube were measured after reaching a steady state. Thereafter, mass flow rates of hot and cold fluids, temperatures, and pressure drop measurements data was collected for all inserted tube cases.

The uncertainties associated with all measuring equipment's used in the experiments are listed in Table 3.

3. Data collection and analysis

The following methodology is used to calculate the average Nusselt number and the friction factor:

Cold water heat gain is estimated as:

$$Q_c = \dot{m}_c C_{p,w} \left(T_{c,out} - T_{c,in}\right) \tag{1}$$

Hot water heat lost in annulus is estimated as:

$$Q_h = \dot{m}_h C_{p,w} \left(T_{h,\,in} - T_{h,\,out}\right) \tag{2}$$

The average heat transfer rate (Qavg), is estimated as:

$$Q_{avg} = \frac{Q_c + Q_h}{2} \tag{3}$$

For the purpose of calculating the average inner side heat transfer coefficient, it is assumed that the temperature of the tube wall surface is constant. Therefore, the heat transfer coefficient (hi) can be calculated as follows by disregarding the thermal resistance of the wall of a copper tube [30]:

$$Q_{avg} = h_i A_i \Delta T_{lm} \tag{4}$$

Where

$$\Delta T_{lm} = \frac{\left(\widetilde{T}_s - T_{c,\,out}\right) - \left(\widetilde{T}_s - T_{c,\,in}\right)}{\ln\left[\left(\widetilde{T}_s - T_{c,\,out}\right) / \left(\widetilde{T}_s - T_{c,\,in}\right)\right]} \tag{5}$$

and

$$A_i = \pi D_i, L \tag{6}$$

The temperature of the outer surface of the tube wall (test section) $\left(\widetilde{T}_s\right)$ is determined by the average of the temperatures recorded by six thermocouples:

$$\widetilde{T}_s = \sum T_s/6 \tag{7}$$

The average heat transfer coefficient is used to calculate the average Nusselt number as follows: [30]:

$$Nu = \frac{h_i D_h}{k} \tag{8}$$

The average friction factor coefficient is calculated as:

$$f = \frac{\Delta P}{\left(\frac{L1}{D_i}\right)\left(\rho \frac{V^2}{2}\right)} \tag{9}$$

Where V is the inner tube's mean working fluid velocity. The bulk mean fluid temperature is used to measure the thermo physical properties.

To evaluate the effectiveness of passive inserts, numerous researchers have proposed the term "Average performance ratio" (R3) [31]. It is the ratio of the Nusselt number of the inserted tube to the Nusselt number of the plain tube at same pumping power that is required for the inserted tube to keep the flow going. The equivalent Reynolds number is estimated as [31]:

$$f_a \times Re_a^3 = f_p \times Re_c^3 \rightarrow Re_c^{2.75} = f_a \times \frac{R_a^3}{0.079} \tag{10}$$

Where f_a is inserted tube friction factor. The Dittus Boelter Equation [30] can be used to calculate the Nusselt number for the equivalent plain tube Reynolds number Rec as follows:

$$Nu_c = 0.023 Re_c^{0.8}\, Pr^{0.4} \tag{11}$$

The average performance ratio is given by [31]:

$$R3 = \frac{Nu_a}{Nu_c} \tag{12}$$

Table 3. Details of measurements and uncertainties

Measuring Parameter	Equipment	Uncertainty
Pressure	U-tube manometer	±1 mm of CCL4 column (density 1577 kg/m^3)
Mass flow rate	Venturimeter	±0.0125 kg/s
Temperature	T-type thermocouple	±0.4°C

4. Results and discussions

4.1. Plain tube validation

By comparing the Plain tube Nusselt number and friction factor values to the established equations of Dittus Boelter and Blasius for Nusselt number and friction factor, respectively, the experimental setup is confirmed. The results obtained agreed within 10% for the Nusselt number and 8% for friction factor. Experimental results are compared with standard equations in Fig. 8 and Fig. 9 for plain tube Nusselt number and friction factor respectively.

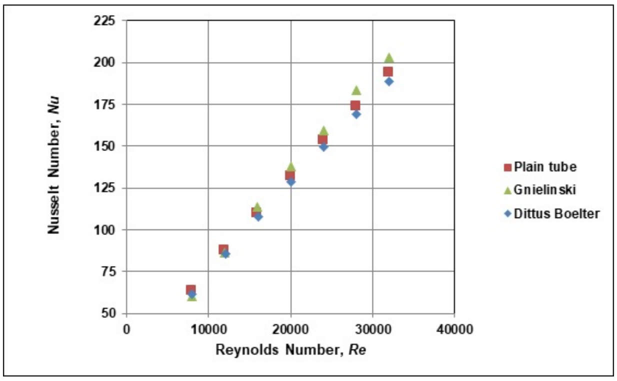

Fig. 8. Plain tube Nusselt Number [29]

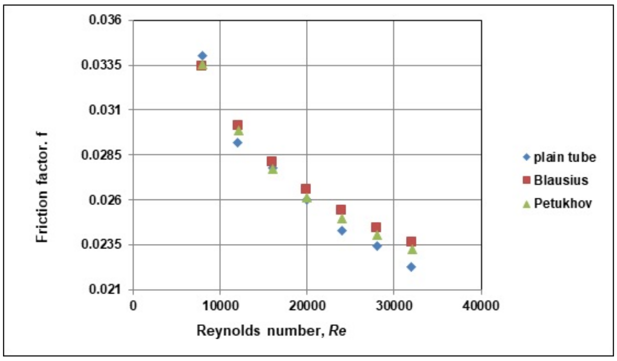

Fig. 9. Plain tube friction factor [29]

4.2. Heat transfer and pressure drop characteristics

Nusselt number and friction factor results for all inserts in the selected range of Reynolds number are shown in Fig. 10 and Fig. 11 respectively. Based on the experimental data, it can be seen that compound case 1 insert accounts to the highest heat transfer rates due to flow separation and swirl generated along the length of test section. Furthermore, a cup shape insert with 0.15 m pitch resulted into highest pressure drop in the selected range of Reynolds number.

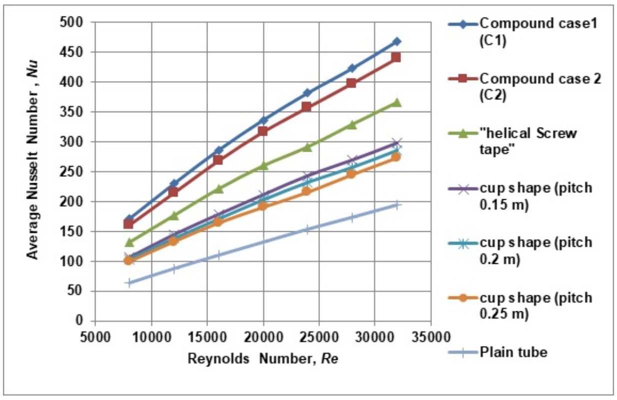

Fig. 10. Average Nusselt number vs. Reynolds number

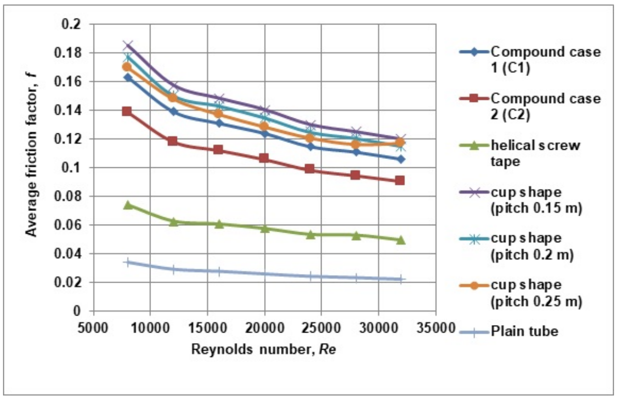

Fig. 11. Average Friction factor vs. Reynolds number

4.3. Effect of pitch length for cup shape inserts

It is reported that the Average Nusselt number ratio of an augmented case to plain tube case is affected considerably by varying pitch of the cup shape insert. It is reported that Nua/Nup ratio is decreased from 1.67 at Reynolds number 8000 to 1.53 at Reynolds number 32000 for a pitch length of 0.15 m. The corresponding values are 1.62 − 1.46 for a pitch length of 0.2 m and 1.55 − 1.38 for a pitch length of 0.25 m as shown in Fig. 12. This reveals that heat transfer performance is better at lower pitch lengths of insert. The lower pitch increases irregularity in the flow creating more turbulence and waviness reducing boundary layer thickness. Hence higher heat transfer rates are seen at lower pitch lengths.

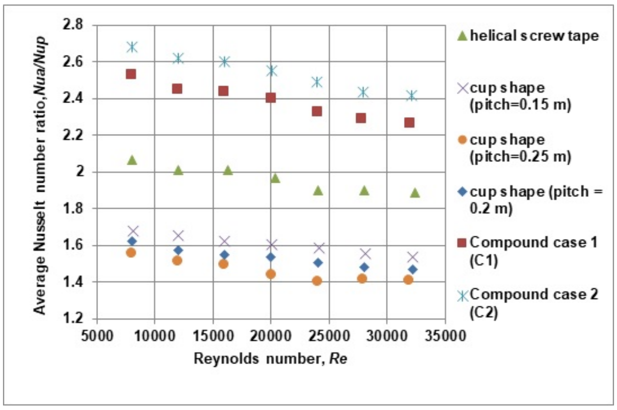

Fig. 12. Average Nusselt number ratio vs. Reynolds number

Friction factor ratios reported for cup shape inserts ranged between 5.44 at Reynolds number 8000 to 5.38 at Reynolds number 32000 for a pitch length of 0.15 m. The corresponding values are 5.22 at Reynolds number 8000 to 5.06 at Reynolds number 12000 for a pitch length of 0.2 m and 5.06 at Reynolds number 12000 to 4.94 at Reynolds number 16000 for a pitch length of 0.25 m as shown in Fig. 13. According to the findings, the sudden obstacle in the flow field causes the friction factors for all cup shape geometries to rise by approximately five times in comparison to plain tubes. Also, friction factor reduces with decreasing pitch as the frequency of obstacles reduces with respect to time.

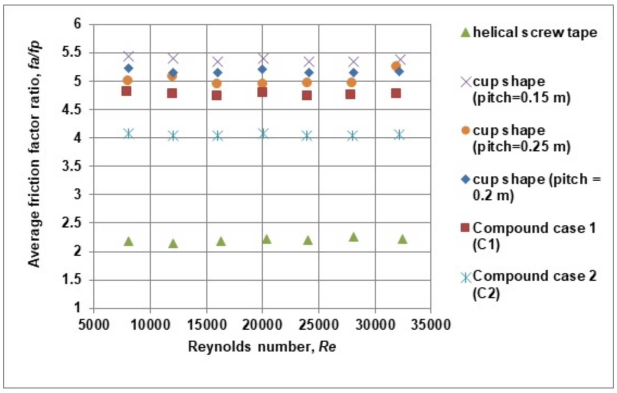

Fig. 13. Average friction factor ratio vs. Reynolds number

For cup shape individual insertion, it is apparent that the flow pattern is altered after striking the leading surfaces of cups. This might lead to increased turbulence and residence time of flow in the test section. It is evident that sudden obstacles in the flow field increase friction factors, but the heat transfer enhancement is insignificant in comparison to other inserts, resulting in lower average performance ratios ranging from 0.67 and 0.63 for Reynolds number 8000 and 32000 (corresponding Equivalent Reynolds numbers 17000 and 94000) respectively as shown in Fig. 14. This is because the sudden obstacle in the flow field causes increased pressure drop and residence time of flow in the test section but the flow turbulence is not increased comparatively as no swirl is imparted by the insert geometry.

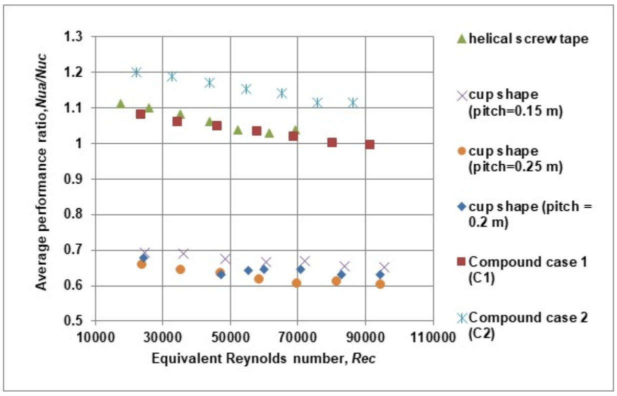

Fig. 14. Average performance ratio vs. Equivalent Reynolds number

4.4. Helical screw tape individual insertion

The average Nusselt number ratio of an augmented case to plain tube case for helical screw tape is reported in the range of 2.06 at a Reynolds number 8000 to 1.88 at a Reynolds number 32000. Thus it can be concluded that the heat transfer rate decreases with increasing Reynolds number. The helical geometry of the tool causes swirl in the flow. It is apparent that the swirling motion breaks the boundary layer near the tube wall resulting into better heat transfer rate between cold and hot fluid. Average friction factor ratio of augmented case to plain tube case ranged between 2.14 at a Reynolds number 12000 to 2.25 at a Reynolds number 28000. As the geometry of tool causes turbulence to the flow, frictional resistance increases. The effect becomes dominant at higher Reynolds number due to higher velocity of flow. The average performance ratio is reported and found in the range of 1.11 and 1.02 for Reynolds number 8000 and 28000 (corresponding Equivalent Reynolds numbers 17000 and 61400) respectively as shown in Fig. 14. As the performance ratio range is above unity for the selected range of Reynolds number, helical screw tape performance is found to be superior over cup shape inserts and plain tube case.

4.5. Compound insertion

Two different compound insertion cases C1 and C2 are studied. The average Nusselt number ratio of an augmented case to plain tube case for case C1 is reported in the range of 2.52 at a Reynolds number 8000 to 2.26 at a Reynolds number 32000. The corresponding values for case C2 are found to be 2.65 – 2.38 respectively. This can be explained by the swirl and pressure gradient caused by the geometry of the insert in the radial direction. The increased swirl and pressure in the radial direction would make the boundary layer along the length of the tube to be thinner which would result in a higher heat flow rate through the fluid. The average friction factor ratio of augmented case C1 to plain tube case ranged between 4.8 at a Reynolds number 8000 to 4.72 at a Reynolds number 16000. The corresponding values reported for case C2 are 4.08 at a Reynolds number 8000 to 4.03 at a Reynolds number 28000. The average performance ratio for case C1 is reported and found in the range of 1.11 and 0.99 for Reynolds number 8000 and 32000 (corresponding Equivalent Reynolds numbers 17000 and 94000) respectively. The average performance ratio for case C2 is reported and found in the range of 1.2 and 1.11 for Reynolds number 8000 and 32000 (corresponding Equivalent Reynolds numbers 17000 and 94000) respectively as shown in Fig. 14.

For compound insertion case C1, the heat transfer enhancement seems to be achieved through two different mechanisms i.e. increase in residence time by cup shape geometry and swirl action imparted by helical screw tape. In this combination, three cups (pitch = 0.25 m) are used hence flow obstacles are more resulting in an appreciable pressure drop. Thus despite of heat transfer enhancement, the performance ratio is almost unity. In comparison to cup-helical-cup (C1) insertion, higher heat transfer rates are observed for the helical-cup-helical (C2) combination due to more flow turbulence created by the helical screw in 2/3rd length of the test section. The friction factor ratios are also reduced as two cups (pitch = 0.25 m) are used. This can be attributed to a better physical intermingling of fluid molecules caused by swirl imparted to the flow by twisted edges of the helical screw tape resulting in higher heat transfer rates.

5. Conclusions

Heat transfer and pressure drop performance of a helical screw tape with core rod and hemispherical cup shape insert were experimentally investigated for turbulent flow. Average Nusselt number enhancement at constant Reynolds number for almost all geometries of individual and compound inserts was reported to be in the range of $38 - 179\%$ along with friction factor increments between 306 - 444%. Heat transfer rates were increased for reduced pitch lengths of cup shape inserts. Due to the high swirl action created by helical screw tape insert geometry the heat transfer performance was improved compared with cup shape insert. The maximum average Nusselt number ratio at equal pumping power (Nu_a/Nu_c) was reported for compound case C2 however corresponding values for case C1, helical screw tape individual insertion is also found to be above 1. Results reveal that helical screw tape individual insert, compound case C1 along with compound case C2 performed better in the selected range of Reynolds number and were found superior over a plain tube. The effect of varying pitch and diameter of helical screw tape insert on heat transfer and pressure drop can be tested further for optimization in both individual and compound insertion cases. As insert devices contribute to enhancing the heat exchanger effectiveness at minimum expenditure, the various insert geometries should be optimized for individual and compound insertion to save energy and reduce the sizes of heat exchangers.

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Nomenclature

Greek Symbols

μ	dynamic viscosity, $\left[kgms^{-1}\right]$
ρ	density of fluid, $\left[kgm^{-3}\right]$

Subscripts

a	augmented tube case
av	average
c	cold
h	average
i	Inner
in	inlet
o	Outer
out	Outlet
p	plain tube case
s	tube wall surface

Other Symbols

ΔP	Pressure drop of fluid, $\left[Nm^{-2}\right]$
$\dot{m}$	Mass flow rate of fluid, $\left[kgsec^{-1}\right]$
A	Area, $\left[m^2\right]$
$C_{P,W}$	Specific heat at constant pressure, $\left[kJkg^{-1}\,K^{-1}\right]$
D	Diameter, $[m]$
f	Average friction factor, $[-]$
H	Heat transfer coefficient, $\left[Wm^{-2}\,K^{-1}\right]$
K	Thermal conductivity, $\left[Wm^{-1}\,K^{-1}\right]$

L	Length of insert, [m]
L	Length of test section for heat transfer, [m]
L_1	Length of tube between pressure taps, [m]
Nu	Average Nusselt number, [−]
Pr	Prandtl number $\left(=\mu C_p/k\right)$, [−]
Q	Heat transfer rate, [kW]
$R3$	Average performance ratio, [−]
Re	Reynolds number $(=\rho VD/\mu)$, [−]
T	Temperature, [°C]
T_{lm}	Logarithmic mean temperature difference, [°C]
V	Mean fluid velocity, $\left[\mathrm{msec}^{-1}\right]$

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