THERMOHYDRAULIC PERFORMANCE COMPARISON OF COMPOUND INSERTS FOR A TURBULENT FLOW THROUGH A CIRCULAR TUBE

by

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Heat transfer and pressure drop characteristics of three different passive inserts are experimentally investigated for individual and compound insertion. Insert cross-section is altered along the length of test section for compound insertion. Test runs were conducted in a concentric circular tube in tube heat exchanger in the Reynolds number range of 8000 to 32000 with water as a working fluid. Enhancements in Nusselt number and friction factors are reported to be in the range of 38-234% and 55-524%, respectively, over plain tube. The average performance ratios based on equal pumping power are also reported and found in the range of 0.63-1.53. Based on experimental results, optimum combination for compound insertion is proposed.

Key words: *turbulent flow, compound insertion, heat transfer*

Introduction

Heat transfer augmentation of heat exchangers is possible by application of different passive augmentation devices like twisted tapes, coiled wire inserts, *etc.* Many researchers through experimental results have suggested the individual or combined use of these devices depending upon the range of Reynolds number. A brief review of work done in the field is reported in this section.

Kumar and Prasad [1] experimentally investigated the effect of twisted tape inserts on heat transfer and pressure drop in the Reynolds number range of 4000 to 21000 with two twist ratios of 3 and 12. They reported increments in heat transfer and pressure drop with decreasing values of twist-pitch to tube diameter ratio. Chen *et al.* [2] provided correlations for heat transfer and pressure drop based on experimental data of different geometries of dimpled tube in turbulent flow and also concluded that with the application of dimpled tube, the weight and size of heat exchangers can be reduced to almost half.

Yilmaz *et al.* [3] performed experiments to see the influence of swirl generators with conical and spherical deflecting elements on heat transfer in the Reynolds number range of 32000 to 110000 using air as a working fluid and observed that deflecting elements for swirl generators are not advantageous over plain swirl generators. Paisarn [4] investigated the effect of coil wire insert on heat transfer and pressure drop in horizontal concentric tube heat exchang-

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er with cold and hot water as working fluids and reported that coil wire inserts have significant effect on the enhancement of heat transfer especially on laminar flow regimes. Also effect of coil wire insert on the enhancement of heat transfer tends to decrease as Reynolds number increases. Naphon *et al.* [5] reported significant enhancement in heat transfer and friction factor of horizontal tube fitted with helical rib inserts with water as a working fluid in the Reynolds number range of 15000 to 60000. Li *et al*. [6] tested discrete double inclined ribs in the Reynolds number range of 15000 to 60000 with water as a working fluid and concluded that the rear vortex and the main vortex contribute much to the heat transfer enhancement in these tubes. Seemawute and Eiamsa-ard [7] studied the effect of peripherally cut twisted tape with alternate axis in a uniform heat flux circular tube for Reynolds number between 5000 to 20000 with water as a working medium and reported maximum thermal performance (based on constant pumping power) of 1.25, 1.11, and 1.02 for twisted tape with alternate axis, peripherally cut tape and regular twisted tape, respectively. Eiamsa-ard *et al.* [8] experimentally compared the performance of oblique delta winglet twisted tape with straight delta winglet twisted tape in the Reynolds number range of 3000 to 27000 with water as a working fluid and observed higher Nusselt number and friction factors for oblique delta winglet twisted tape than those for plain tube and tube fitted with typical twisted tape. Eiamsa-ard *et al.* [9] experimentally investigated the effects of nine different peripherally cut twisted tape inserts on heat transfer and friction factor with constant twist ratio ($y/w = 3$) and different tape depth ratio ($d/w = 0.11$, 0.22, and 0.33) in the Reynolds number range of 1000 to 20000 using water as a working fluid. The reported maximum enhancement in Nusselt number was 2.6 for turbulent regime and 12.8 for laminar regime resulting in maximum performance factors of 1.29 for turbulent and 4.88 for laminar regime, respectively. Kongkaitpaiboon *et al.* [10] tested the heat transfer and pressure friction factor performance of a tube fitted with perforated conical ring with pitch ratios of 4, 6, and 12 along with 4, 6, and 8 perforated holes in the Reynolds number range of 4000 to 20000 with air as a working medium. The results revealed that the thermal performance factor increased with increasing number of perforated holes and decreasing pitch ratio though heat transfer and friction factor increased with decreasing pitch ratios and number of perforated holes. Liu and Sakr [11] reviewed the experimental and numerical work done on passive heat transfer techniques by different researchers and mentioned that twisted tape inserts perform better in laminar flow than turbulent flow. However, techniques like conical rings, nozzles, *etc.* perform better in turbulent flow.

Along with individual inserts, many researchers have also studied the combined use of different inserts to enhance the heat transfer. Promvonge and Eiamsa-ard [12] studied the effect of combination of conical nozzle inserts and swirl generators on heat transfer in the Reynolds number range of 8000 to 18000 with air as a working fluid in uniform heat flux condition and reported higher heat transfer rates compared to individual use of inserts. Promvonge and Eiamsa-ard [13] investigated the combined use of conical ring turbulators and twisted tape swirl generator (twist ratio $y = 3.75$, and 7.5) in the Reynolds number range of 6000 to 26000 with air as a working medium. They observed decrement in enhancement efficiency with increasing Reynolds number and stable enhancement efficiency over Reynolds number of 16000 and recorded the maximum enhancement to be 3.7 times over that of a plain tube. Promvonge [14] investigated the effect of combining wire coils and twisted tapes in the Reynolds number range of 3000 to 18000 with air being a working medium and found that the use of combined turbulators is more effective at low Reynolds number. Thianpong *et al.* [15] investigated the performance of compound insertion using a dimpled tube fitted with twisted tape swirl generator for twist ratios of 3, 5, and 7 in the Reynolds number range of 12000 to 44000 with water being a working fluid. Results revealed that compound enhancement technique significantly affected heat transfer and friction factor. Also higher heat transfer and friction factors were recorded for decreasing pitch and twist ratios. Promvonge *et al.* [16] used combined wedge ribs and winglet type vortex generators in a solar air heater channel in the Reynolds number range of 5000 to 22000 with three different angles of attack (60°, 45°, and 30°). They observed decrement in Nusselt number with increasing Reynolds number along with higher heat transfer rates at lower angles of attack and low Reynolds numbers. Saha [17] tested full and short length twisted tapes with and without oblique teeth for rectangular and square ducts with axial corrugations for turbulent flow with Reynolds number ranging from 10000 to 100000 using air as a working fluid. He observed a minute improvement in thermo-hydraulic performance for twisted tapes with oblique teeth compared to twisted tape without oblique teeth. Garcia *et al.* [18] analyzed and compared corrugated tubes, dimpled tubes and wire coils based on experimental results and concluded that the roughness shape has more impact on pressure drop rather than on heat transfer. Based on results, they have recommended using a smooth tube for Reynolds number less than 200, wire coil for Reynolds number range of 200 to 2000 and corrugated or dimpled tube for Reynolds number above 2000.

All compound inserts that have been reported in literature are tested with uniform cross-section. In this study, variable cross-section of insert along the length of test section is proposed for heat transfer augmentation. Three different passive inserts are considered for individual and compound insertion and experimental data for heat transfer and pressure drop is presented in this article.

Experimental set-up and methodologies

Details of inserts

The geometric details of three different inserts are listed in tab. 1. Taking into consideration the machinability and durability, inserts were manufactured from aluminum. Five point star shape cross sectioned twisted rod insert was manufactured by twisting the splined rod (with star cross-section) prepared on milling machine up to the required twist angle to achieve the required pitch. For cup shape insert, cups with central hole were prepared separately and mounted on threaded rod with locking nut so as to alter the distance between the cups as per the requirement. Coiled wire insert was prepared manually by wounding the selected wire over rod of suitable diameter to maintain the required size of coiled wire. Star shape and cup shape inserts were manufactured in three different segments with each segment length equal to 1/3rd of total test section length. All segments were provided with end threaded portions for intermediate connection of different segments. Thus the requirement of variable cross-section of insert along the length of test section was satisfied to form different passive compound inserts. Details are shown in figs. 1 and 2.

Table 1. Geometric details of individual inserts

(A)	5 Point star shape twisted rod	
1	Cross-section	5 point star shape
2	Segment length	0.267m
3	Full length	0.8m
4	Diameter	12 mm
5	Pitch	40 mm (full length)
(B)	Cup shape with threaded core rod	
1	Cross-section	Hemispherical cup
2	Segment length	0.267 m
3	Full length	0.8 m
4	Outer diameter	17.9 mm
5	Cup thickness	3 mm
6	Threaded core rod diameter	5.8 mm
(C)	Coiled Wire	
1	Coil diameter	18.2 mm
2	Wire diameter	0.8 mm

Experimental set-up details

Figure 1. Picture of individual inserts

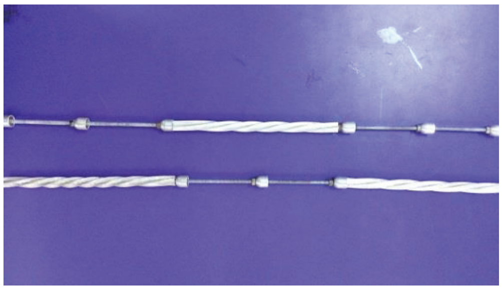

Figure 2. Picture of compound inserts

The details of experimental set-up are shown in fig. 3. The set-up consisted of a tube in tube heat exchanger with counter flow arrangement wherein the cold water at 25 °C from a large capacity underground water tank was circulated through 21.4 mm diameter inner Cu tube (test section) of length 0.8 m and hot water through 48.3 mm diameter outer steel tube (annulus) in a closed loop. The calming length of the test section was kept 450 mm. The required flow rates for cold water corresponding to selected range of Reynolds number (8000 to 32000) at inlet of test section were maintained through two gate valves and one by pass valve fitted in cold water circulation line. For accurate measurements of flow rates, a pre calibrated venturimeter was placed upstream of test section with U-tube manometer using carbon tetrachloride (CCl_4) as a manometric fluid for pressure drop measurements across inlet and throat of venturimeter. The cold water after circulation through test section was recollected in underground tank. Due

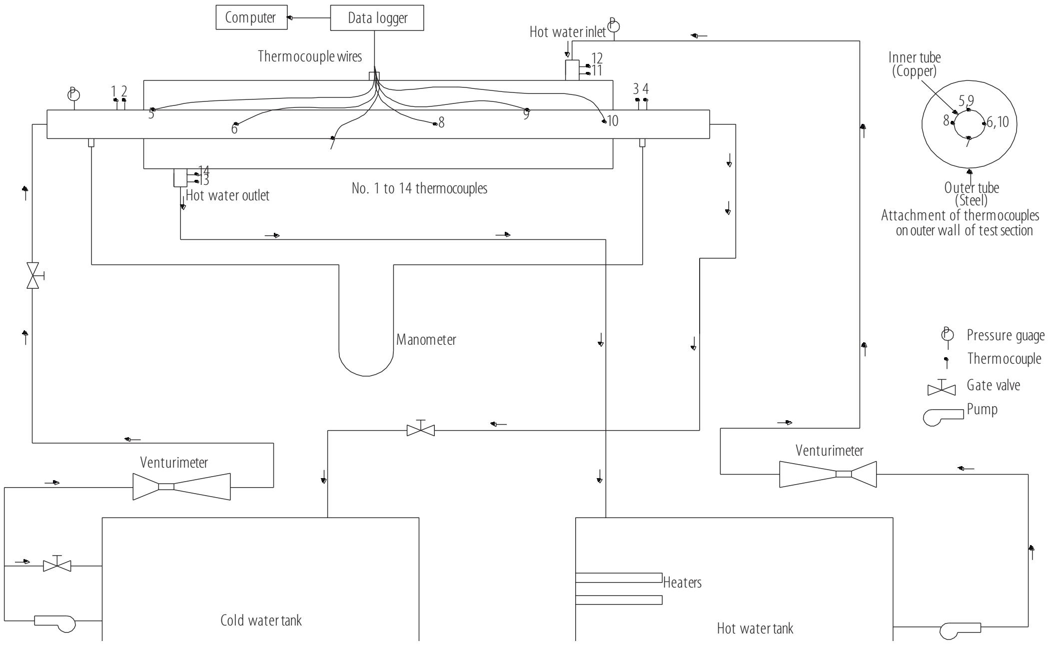

Figure 3. Details of experimental set-up

to abundant quantity of cold water available in underground tank, it acted as a sink and further arrangement for cooling of cold water after passing through test section was not required. For pressure drop measurements across test section, a separate U-tube manometer using CCl_4 as a manometric fluid was used. A well-insulated water tank of 100 liter capacity fitted with two heaters of 2 kW capacities was used for heating of water. The temperature of water in hot water tank was maintained at 75 °C through thermostat.

For plain tube test runs, once the hot water temperature reached 75 °C, it was circulated through annulus in a closed loop. Thereafter, the cold water was circulated through test section and system was allowed to reach a steady-state at which all temperature and pressure readings were taken. Thereafter, the procedure was repeated for tube with different individual and compound inserts as per the combinations mentioned in tab. 2.

The inlet and outlet temperatures of cold and hot water along with test section outer surface wall temperatures were measured by using pre-calibrated T-type thermocouples connected to a data logger. For each inlet and outlet location of cold and hot water, two thermocouples were placed for accurate measurements. Also for test section outer surface wall temperatures, six thermocouples were placed circumferentially and equidistant along the length. All measuring devices used in experimentation along with their uncertainties are listed in tab. 3.

Table 2. Details of combinations for compound inserts

Inserts	Cross-section for upstream 1/3rd length	Cross-section for middle 1/3rd length	Cross-section for downstream 1/3rd length	Type
C1	5 point star shape	5 point star shape	5 point star shape	Single
C2	Cup with core rod	Cup with core rod	Cup with core rod	Single
C3	Coiled wire	Coiled wire	Coiled wire	Single
C4	5 point star shape	5 point star shape	Cup with core rod	Compound
C5	5 point star shape	Cup with core rod	5 point star shape	Compound
C6	Cup with core rod	Cup with core rod	5 point star shape	Compound
C7	Cup with core rod	5 point star shape	Cup with core rod	Compound
C8	Star and coiled wire	Star with coiled wire	Star with coiled wire	Compound
C9	Cup with coiled wire	Cup with coiled wire	Cup with coiled wire	Compound

Table 3. Details of measurements and uncertainties

Measurement	Instrument	Uncertainty
Pressure	U-tube manometer	±1 mm of CCl_4 column (density 1577 kg/m^3)
Mass flow rate	Venturimeter	±0.0125 kg/s
Temperature	T-type thermocouple	±0.4 °C

Data collection and analysis

Based on inner diameter of test section, the average Nusselt number and friction factor are calculated:

– the heat gain by cold water is obtained by

$$Q_\mathrm{c} = \dot{m}_\mathrm{c} C_{p,w} (T_\mathrm{c,out} - T_\mathrm{c,in}) \tag{1}$$

– the heat lost by hot water in annulus is calculated

$$Q_\mathrm{h} = \dot{m}_\mathrm{h} C_{p,w} (T_\mathrm{h,in} - T_\mathrm{h,out}) \tag{2}$$

Due to convection and radiation losses, the difference between the heat gain by cold water and heat given by hot water was found to be in the range of 3 to 10%. Thus average inner side heat transfer coefficient is calculated based on average heat transfer rate, Q_{avg}:

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

A constant tube wall surface temperature is assumed for average inner side heat transfer coefficient calculations. Thus by neglecting the thermal resistance in Cu tube wall, heat transfer coefficient (h_i) is calculated [19]:

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

where

$$\Delta T_{lm} = \frac{(\tilde{T}_s - T_{c,out}) - (\tilde{T}_s - T_{c,in})}{\ln \dfrac{\tilde{T}_s - T_{c,out}}{\tilde{T}_s - T_{c,in}}} \tag{5}$$

and

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

The average of temperatures recorded by six thermocouples placed on tube wall (test section) outer surface is considered as tube wall surface temperature, $\tilde{T}_s$:

$$\tilde{T}_s = \sum \frac{T_s}{6} \tag{7}$$

The average Nusselt number is obtained from average heat transfer coefficient:

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

The average friction factor coefficient is calculated:

$$f = \frac{\Delta P}{\left(\dfrac{L_1}{D_i}\right)\left(\rho \dfrac{V^2}{2}\right)} \tag{9}$$

where V is the mean velocity of working fluid in inner tube. Thermo-physical properties are taken at overall bulk mean fluid temperature.

Many researchers have proposed the term *average performance ratio* (*R3*) to judge the performance of passive inserts [20]. It is the ratio of Nusselt number of inserted tube to the Nusselt number of plain tube corresponding to equal pumping power as it is required for inserted tube to maintain the flow. The equivalent Reynolds number is given by:

$$f_a \text{Re}_a^3 = f_p \text{Re}_c^3 \rightarrow \text{Re}_c^{2.75} = f_a \frac{\text{Re}_a^3}{0.079} \tag{10}$$

where f_a is friction factor of inserted tube. The Nusselt number for equivalent plain tube Reynolds number Re_c is calculated from Dittus Boelter equation:

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

The average performance ratio is calculated:

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

Results and discussions

Verification of plain tube

The experimental values for Nusselt number and friction factor for plain tube are compared with following equations:

$$\mathrm{Nu} = 0.023\,\mathrm{Re_p^{0.8}}\,\mathrm{Pr^{0.4}} \qquad \text{(Dittus Boelter)} \tag{13}$$

$$\mathrm{Nu} = \frac{\frac{f}{8}(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\left(\mathrm{Pr}^{2/3}-1\right)\sqrt{\frac{f}{8}}} \qquad \text{(Gnielinski)} \tag{14}$$

$$f = 0.079\mathrm{Re}^{-0.25} \qquad \text{(Blasius)} \tag{15}$$

The experimental results agree well within 10% for Nusselt number and 8% for friction factor with previous equations. The results are compared in figs. 4 and 5. Thus plain tube is verified. In addition, as some new geometries are proposed in the experimentation, the inserted tube Nusselt number and friction factors are validated by testing the identical geometry (conical strips) used by Guo *et al.* [21] in their research work. The comparison of their published data and present results is shown in figs. 6 and 7.

Thermo-hydraulic performance of inserts

The experimental results for Nusselt number and friction factor are represented in figs. 4 and 5. Also the enhancements in Nusselt number and friction factor over plain tube are

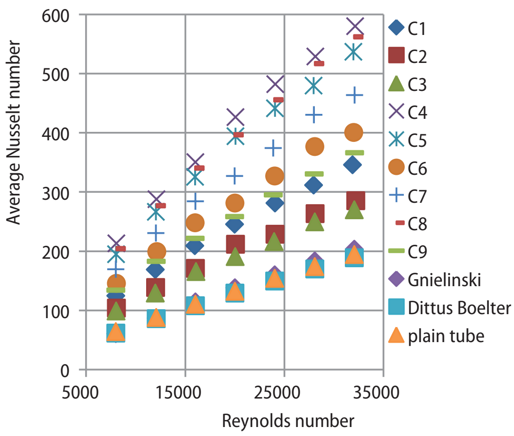

Figure 4. Variation of average Nu *vs.* Re

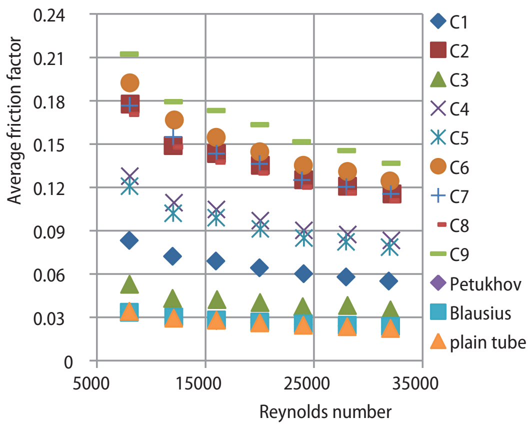

Figure 5. Variation of average *f vs.* Re

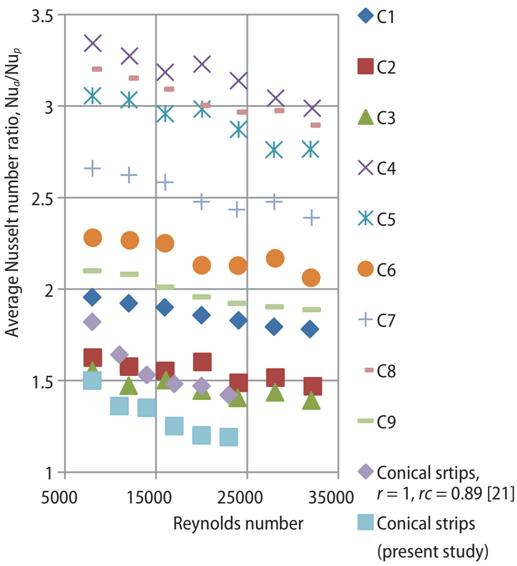

Figure 6. Average Nu ratio (Nu_a/Nu_p) *vs.* Re

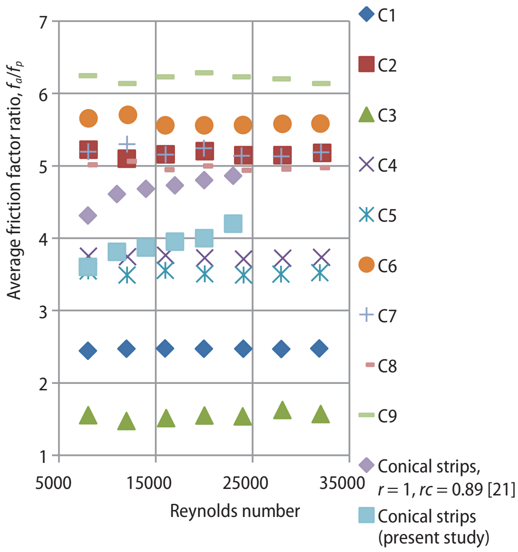

Figure 7. Average *f* ratio (f_a/f_p) *vs.* Re

shown in figs. 6 and 7. For star cross-section full length insertion (C1), it is seen that the heat transfer rates are increased compared to plain tube. This can be explained by the swirl and pressure gradient caused by geometry of insert in the radial direction. The increased swirl and pressure in the radial direction would make the boundary layer along the length of tube to be thinner which would result in higher heat flow rate through the fluid. In case of cup shape individual insertion (C2), the transfer enhancement is not appreciable compared to other inserts which leads to lower average performance ratios ranging from 0.67 to 0.63 for Reynolds number 8000 to 32000, respectively. 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. For coiled wire individual insertion (C3), the laminar sub layer breaking is expected to be promoted by wire surface near the tube wall resulting in higher heat transfer rates. It is apparent from the results that coiled wires create less pressure drop compared to other inserts in the selected range of Reynolds number.

In case of star-star-cup (C4) insertion, three cups were added in last 1/3rd length. The average performance ratio varied from 1.53 to 1.41. This can be attributed to better physical intermingling of fluid molecules caused by swirl imparted to the flow by twisted edges of the star shape tool resulting in higher heat transfer rates. However, due to solid cross-section obstacle, the insert causes higher pressure drop resulting in higher pumping power requirements.

For star-cup-star (C5) insertion, the average performance ratio was found to be ranging from 1.42 to 1.32 for Reynolds number 8000 to 32000, respectively. Due to increased residence time caused by cup shape and thin boundary layer caused by swirl due to star shape, higher heat transfer rates are seen compared to full length cup shape insertion. In case of cup-cup-star (C6) insertion, the flow residence time is expected to be increased by cup shape obstacle and turbulence by star cross-section placed in the last 1/3rd length of test section. The flow turbulence naturally increases along the length thus swirl device downstream of test section is not effective as it would have been in upstream 2/3rd length of test section.

For cup-star-cup (C7) insertion, higher momentum imparted by cup along with swirl generated by star shape increases heat transfer rate along with friction factors compared to full length cup shape insert. Nusselt number ratio at equal pumping power is found to be in the range of 1.11 to 1.02 as shown in fig. 8. In case of star and coiled wire combined insertion (C8), coiled wire would increase the waviness near tube wall promoting laminar sub-layer breaking. Star shape increases the swirl. The proper mixing of stream mass and mass near the tube wall would result in increased heat transfer rate. Coiled wire increases the pressure drop but the overall enhancement is found to be considerable among the selected inserts. For cup and coiled wire combined insertion (C9), performance lies between the full length cup and full length coiled wire insert. The pressure drop increases due to cup insert but the heat transfer also increases due to increased turbulence by possible laminar sub-layer breaking by coiled wire.

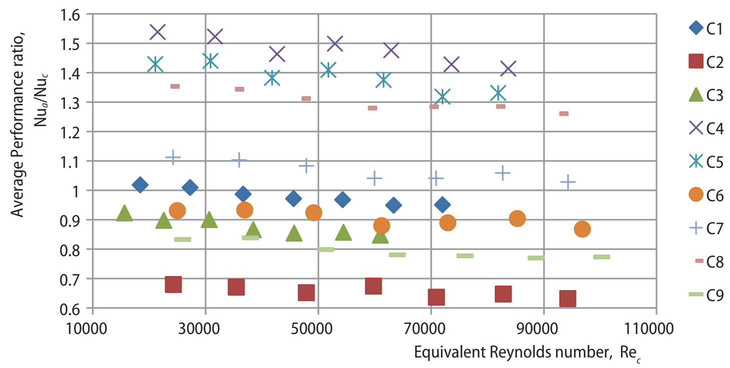

Figure 8. Variation of (Nu$_a$/Nu$_c$) *vs.* Re$_c$

Conclusions

Comparative experimental study of heat transfer and flow friction behavior of three different inserts is performed for individual and compound insertion. The cross-section of inserts along the length of test section was altered to check the heat transfer enhancement possibility. The performance of individual insert in combination with coiled wire is also studied. The results reveal that, the performance ratios for compound inserts C4, C5, C6, and C8 are above 1 hence are found suitable for heat transfer augmentation. Also the performance ratios compound inserts C7 and C9 are less than 1 hence are not found suitable for heat transfer augmentation. The performance ratios for star cross-section in combination with coiled wire are above full length inserts and plain tube. Also the four varying cross-section area of inserts along length, the maximum average performance ratios (range 1.53 to 1.41) are obtained for star-star-cup cross-section (C4) compound insertion and found optimum among the selected inserts.

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Nomenclature

A – area, [m^2]
$C_{p,w}$ – specific heat at constant pressure, [kJkg^{-1}K^{-1}]
D – diameter, [m]
f – average friction factor, [–]
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
k – thermal conductivity, [Wm^{-1}K^{-1}]
L – length of test section for heat transfer, [m]
L_1 – length of tube between pressure taps, [m]
$\dot{m}$ – mass flow rate of fluid, [kgs^{-1}]
Nu – average Nusselt number, [–]
ΔP – pressure drop of fluid, [Nm^{-2}]
Pr – Prandtl number (= $\mu C_p/k$), [–]
Q – heat transfer rate, [kW]
Re – Reynolds number (= $\rho VD/\mu$), [–]
$R3$ – average performance ratio, [–]
T – temperature, [°C]

T_{lm} – logarithmic mean temperature difference, [°C]
V – mean fluid velocity, [ms^{-1}]

Greek symbols

μ – dynamic viscosity, [kgms^{-1}]
ρ – density of fluid, [kgm^{-3}]

Subscripts

a – augmented tube case
avg – average
c – cold
h – hot
i – inner
in – inlet
out – outlet
p – plain tube case
s – tube wall surface

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