# HEAT TRANSFER AND FRICTION FACTOR CHARACTERISTICS OF PIPE-IN-PIPE HEAT EXCHANGER FITTED WITH VARIANT PLAIN TAPE INSERT

by

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*The heat exchanger is used to transfer heat between the fluids without mixing them for both cooling and heating processes. Normally the fluids are separated by a solid wall or tube that is made of different materials in order to avoid mixing. The performance of a heat exchanger is predicted on the basis of heat transfer rate. Many new techniques are being explored by industries to improve the heat transfer rate of heat exchangers. In this work, a double tube heat exchanger is used for studying the variation in heat exchange by inserting a flat tape with different geometries: plain tape, plain tape step cut arc, and plain tape step cut rectangle. Experiments are carried out by varying pressure of hot water and the evaluation is done for different mass-flow rates with inlet temperatures of hot and cold water at 53 °C and 30 °C, respectively, under counter-flow arrangements. The experimental results revealed that the plain tape inserts in counter-flow arrangements enhance heat transfer rate substantially thereby increasing the effectiveness of the system for a marginal increase in pressure drop.*

Key words: *double tube heat exchanger, plain tape, plain tape-step cut arc, plain tape-step cut rectangle, Nusselt number, Reynolds number*

## Introduction

Heat exchanger interchanges the heat among fluids of various temperatures, which are separated by a solid wall or any other medium. In the utilization of thermal energy, it is very important to control the temperature of the incoming and outgoing fluid streams. The temperature gradient or the differences in temperature facilitate the transfer of heat. Transfer of heat could occur through either of the three heat transfer principles: radiation, conduction, and convection. Conduction takes place as the heat from the fluid at elevated temperature passes through the solid wall. The wall must be thin and fabricated from a completely conductive material to increase the heat transfer rate. Convection plays the most important role in the

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heat transfer of a heat exchanger. The forced convection mode is comparatively a better way to transfer the heat of one moving fluid to another moving fluid in a heat exchanger.

## Literature review

Dittus and Boelter [1] studied the fundamentals involved in the heat transfer from water to atmosphere through tubular radiator. Hasim *et al.* [2] have investigated the enhancement of heat transfer by tube using equal pumping power evaluation method. Helically ribbed tube with twisted tube inserts were used to enhance the heat transfer. An experimental investigation work to predict heat transfer coefficient and friction factors by gathering non-isothermal correlations with twisted tape insert was proposed and presented by Naphon [3]. Twisted tapes were inserted in the test tube in different orientations such as: full-length typical twisted tape at different twisted ratios ($y$ = 6.0 and 8.0) and twisted tape with various free space ratios ($S$ = 1.0, 2.0, and 3.0). Increased friction factor and heat transfer in free space ratios were studied [4].

Vivekanadan *et al*. [4] have used FEA to analyze the thermal changes and analytical equations. Kumar *et al.* [5] investigated the improvement in heat exchanger using spiral coiled sheet plates. Eiamsa-ard [6] has presented an experimental work on dimpled tube fitted with twisted tape swirl generator to predict friction and compound heat transfer. The experiments were performed with different pitch (Pr = 0.7 and 1.0) and twist ratios ($y/w$ = 3, 5, and 7). Increased heat transfer and friction factor were observed in dimpled tube with twisted tape. Investigations were carried out on flow friction, heat transfer and thermal performance characteristics of tube fitted with delta winglet twisted tape. Using water as working fluid, oblique delta winglet twisted (O-DWT) tape and straight delta winglet twisted (S-DWT) tapes were studied. From the results, O-DWT was found to be more efficient than S-DWT, as per the investigation by Eiamsa-ard *et al.* [7]. Experiments were done to investigate the friction loss and thermal performance factor of round tube. By increasing depth ratio and decreasing width, heat transfer was found to increase [8].

Pethykool *et al.* [9] reported that single phase turbulent flow heat transfer is increased by using helically corrugated tube. When compared to smooth tubes, the corrugated tubes heat transfer and thermal performance were better. Godwin *et al.* [10] have investigated flat tape and twisted tape for heat transfer enhancement by comparing the results of Nusselt number and overall heat transfer coefficient. Sivaprakash *et al.* [11] have studied the heat transfer variations with different types of profile inserts. The analytical and experimental results were compared with heat transfer coefficients and heat transfer effectiveness. The report has concluded that the trapezoidal cut twisted tape can be used in place of plain twisted tape to reduce the size of heat exchanger in industrial applications.

Double pipe counter water flow heat exchanger with an insert of semicircular disc baffles was used by Sakthivel *et al.* [12] to study the heat transfer and turbulent flow behavior. While comparing smooth tube friction factor, the heat transfer coefficient got increased when the space of baffle is decreased. The concentric tube heat exchanger was analyzed using ANSYS for steel and the work was carried out by Sarmad [13]. Annulus which has semicircular and quarter circular baffles was used in double pipe heat exchanger (DPHE) and analyzed. In this simulation work, increase in heat transfer rate and temperature variations were studied [13]. Sathish *et al.* [14] have used machine learning technique to predict heat transfer rate. Kanade Rahul *et al.* [15] have investigated on proper geometry of baffles required to improve heat transfer rate.

Simhadri *et al.* [16] have experimented with hot fluid pipe is fitted with twisted inserts. When the length of the twist was increased, effectiveness also increased. Qasim and Hussein, [17] investigated DPHE fitted with various types of twisted tapes to predict heat transfer and friction factor. Heat transfer rate increased in V-TT. Singh [18] has reported that twisted tapes were used to enhance heat transfer and friction factor and this technique is used for reducing heat exchanger size. Kaderi and Suredar [19] have analyzed concentric tube heat exchanger and found coefficient of heat transfer being increased while increasing the fluid-flow rate. The increase of turbulent flow heat transfer in DPHE was investigated using six types of twisted tapes (normal, regularly spaced, triangular-cut, rectangular-cut, semicircular-cut, and drilled).

Based on the study of available literature, it has been observed that the modifications in inserts such as small cut on the tape, twisted tapes, coil wire, trapezoidal, mesh inserts, brush inserts, *etc.*, are making higher impact on heat transfer rate of the DPHE. The reason behind the high thermal enhancement factor (TEF) is that those small gaps bring pressure drop in the system to a reasonable level. The present work reports the experimental work on heat transfer and friction factor characteristics of a DPHE fitted with plain tape, plain tape step cut arc (PT-SCA), and plain tape step cut rectangle (PT-SCR) with Reynolds number 2000 and 14000. The plain tapes with stepped arc and rectangular profiles in the peripheral region of the tape are used. This type of tapes could perform in the same manner as that of twisted tape, trapezoidal, square, triangular peripherally cut tapes used by other researchers. The experimental results obtained for the tube fitted with PT-SCA and PT-SCR were also compared with those of the tube with plain tape and plain tube.

## Experimental set-up

Schematic diagram of the experimental set-up is shown in fig. 1. The outer side of steel is insulated by glass wool to avoid heat loss to the atmosphere. A 3 kW heater is molded into the tank 1 to heat the fluid (water) and the temperature of the fluid is maintained at 53±1 °C through the temperature controller. The hot water is pumped to the annulus between two pipes from tank to outer pipe. The hot water is circulated from tank-1 and cold water circulated from tank 2.

Both hot and the cold water outlet pipes are coupled to the individual crystal rotameter that has the range of 20 L per minute with ±0.1 L per minute accuracy and used to measure the flow rates of hot and cold water. The temperature of cold fluid is maintained at the range of 30±1 °C through ice cube. The ice cubes are stored separately in tank 3. If the cold fluid temperature increases, the ice cubes are driven into the cold-water tank (tank 2). The thermocouples (eight RTD Pt 100) are used to measure the temperatures of both the tanks (T7 and T8) and also the inlet (T3, T4, and T5) and outlet (T1, T2, and T6) temperatures of the hot and cold water. Simultaneously the same

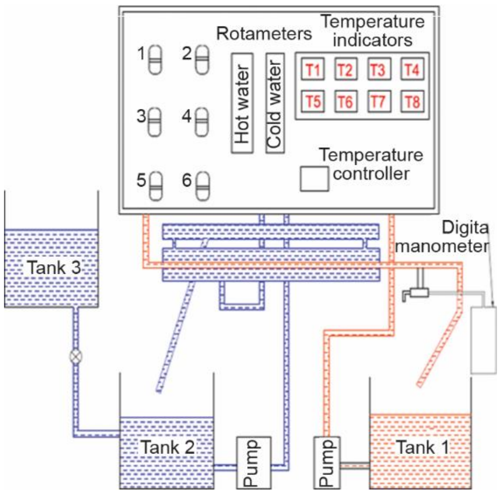


**Figure 1. Schematic diagram for counter flow double tube heat exchanger;** *1 – temperature controller, 2 – temperature meter, 3 – hot, 4 – digital meter, 5 – main, 6 – cold*

set of readings are taken for plain tape, PT-SCA, and PT-SCR with flow rate range of 2-8 L per minute. Pressure drop is measured by using digital manometer which is coupled with copper tube outlet. Plain tapes are made up of aluminum strips of thickness 3 mm and width 23.5 mm. The PT-SCA and PT-SCR profiles are cut on the surface of the plain tube with a pitch of 165 mm. Geometries of plain tape, PT-SCA, and PT-SCR are shown in figs. 2-4.

**Figure 2. Geometry of plain tape**

**Figure 3. Geometry of PT-SCA**

**Figure 4. Geometry of PT-SCR**

*Data reduction*

Heat transfer from the hot water in the test section:

$$Q_h = m_h Cp_h (T_{h1} - T_{h2}) \tag{1}$$

Heat received by the cold water in the test section:

$$Q_h = (T_{c2} - T_{c1}) \tag{2}$$

The percent of loss of heat among the hot and cold waterside in the current heat exchanger are denoted as:

$$\varepsilon = \frac{Q_h - Q_c}{Q_c} 100\% \tag{3}$$

The mean rate of transferred heat for a hot and cold waterside was considered for the computation of the internal convective coefficient of transferred heat:

$$Q_{avg} = \frac{Q_c + Q_h}{2} \tag{4}$$

The entire coefficient of transferred heat is:

$$Q_{avg} = UA_i \, \Delta T_m \tag{5}$$

where $A_i = \pi d_i l$.

$$Q_{avg} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\left(\dfrac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}\right)}$$

The annulus side Reynolds number was identified by the follow equation:

$$Q_{avg} = \frac{u_a D_h}{a v_c} \tag{6}$$

The tube side coefficient of transferred heat, $h_i$, established through neglecting the thermal coefficient resistance of the wall of the copper tube:

$$\frac{1}{U}=\frac{1}{h_\mathrm{i}}+\frac{1}{h_\mathrm{o}} \tag{7}$$

where the annulus side coefficient of transferred heat, $h_\mathrm{a}$, was assessed using the association of equations of Dittus and Boelter:

$$\mathrm{Nu_a}=\frac{h_\mathrm{a}\,D_\mathrm{h}}{k_\mathrm{c}}=0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{8}$$

Inner tube side Nusselt number, $\mathrm{Nu_i}$:

$$\mathrm{Nu_i}=\frac{h_\mathrm{i}\,D_\mathrm{i}}{k_\mathrm{h}} \tag{9}$$

Inlet Reynolds number, $\mathrm{Re_i}$:

$$\mathrm{Re_i}=\frac{u_\mathrm{i}\,D_\mathrm{i}}{\mu_\mathrm{h}} \tag{10}$$

The $f$ is computed by:

$$f=\frac{\Delta P}{\left(\dfrac{L}{d_\mathrm{i}}\right)\left(\dfrac{\rho u^2}{2}\right)_\mathrm{h}} \tag{11}$$

## Results and discussion

In this section, the heat transfer and friction factor characteristics and TEF of a DPHE fitted with plain tape, PT-SCA, and PT-SCR are presented. The experiments are performed using plain tapes with two section stepped arc and rectangular profiles (PT-SCA and PT-SCR) in the range of Re 2000 ≤ 14000.

### *Validation of plain tube experimental results*

Equations (12) and (13) show, the experimental data for turbulent flow that is found matching with the plain tube forced convection correlation of Dittus-Boelter [1] equation and Gnielinski [20] equation, Cengel *et al.* [21], with discrepancy of 20%, 20%, and 6% for Nusselt number as shown in fig. 5:

$$\mathrm{Nu}=0.023\ \mathrm{Re}^{0.8}\mathrm{Pr}^{0.3} \tag{12}$$

$$\mathrm{Nu}=\frac{\dfrac{f}{8}(\mathrm{Re}-1000)\,\mathrm{Pr}}{1+12.7\left(\dfrac{f}{8}\right)^{0.5}(\mathrm{Pr}^{2/3}-1)} \tag{13}$$

The variation of friction factor for the plain tube is shown in fig. 6. The data obtained in this work are compared with Blasius equation, eq. (14), Petukhov [22] equation, Cengel *et al.* [21], eq. (15) and it yields the deviation of 16.25%, 13.5%, and 18%, respectively, for friction factor.

$$f = 0.316\mathrm{Re}^{-0.25} \tag{14}$$

$$f = (0.790 \ln \mathrm{Re} - 164)^{-2} \tag{15}$$

The experimental results of the plain tube are correlated for Nusselt number and friction factor, respectively, through eqs. (16) and (17).

$$\mathrm{Nu} = 0.01266\mathrm{Re}^{0.879}\mathrm{Pr}^{0.33} \tag{16}$$

$$f = 0.999\mathrm{Re}^{-0.356} \tag{17}$$

Equations (16) and (17) are found to represent the experimental data within 6% for Nusselt number and 18% for friction factor also shown in figs. 5 and 6.

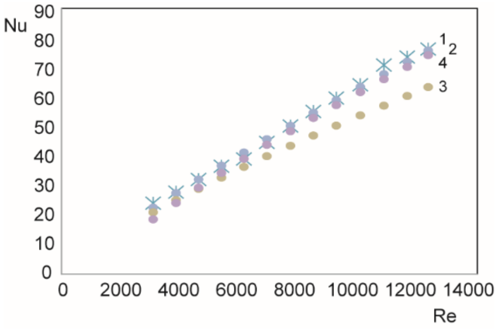


**Figure 5. Data verification of Nusselt number for plain tube under turbulent flow;**
*1 – plain tube experimental data, 2 – plain tube correlation, 3 – Dittus-Boelter equation [1], 4 – Gnielinski equation [20]*

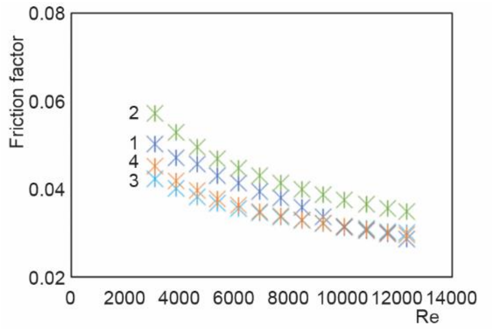


**Figure 6. Data verification of friction factor for plain tube under turbulent flow;**
*1 – plain tube experimental data, 2 – plain tube correlation, 3 – Blasius equation, 4 – Petukhov equation [22]*

### *Validation of plain tape experimental results*

The experimental results of the tube fitted with plain tape are compared with the plain tube and its results are validated using the correlations of available experimental data for the turbulent flows at the inlet test section. The variations of Nusselt number and Reynolds number for the tube fitted with plain tape are presented in fig. 7.

It is observed that for all cases, the Nusselt number increases with the increase in Reynolds number. As expected, plain tape heat transfer rates are higher than those from the plain tube results. Mean Nusselt number of turbulent flow for the tube fitted with plain tape is 1.056 times greater than that of the plain tube. Figure 8 shows the friction factor decreasing continuously with Reynolds number. From the result, the mean friction factor for the plain tape is found to be 1.7423 times higher than that of plain tube.

### *Effect of PT-SCA on heat transfer*

Research works from the literature studies showed that the heat transfer enhancement effect can be increased by modifying the geometry of the tape *i. e.* small cut on the plain tape which is shown in fig. 9.

The experimental results of the tube fitted with PT-SCA are compared with the plain tape and plain tube for the turbulent flows at the inlet of test section and empirical correlations

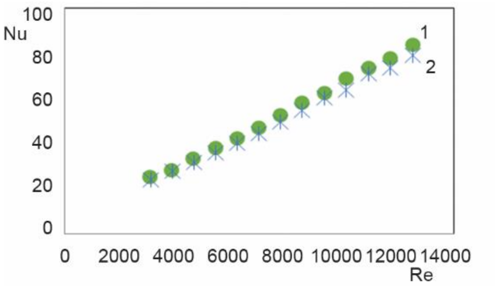


**Figure 7. The Nu *vs.* Re for plain tape and plain tube under turbulent flow;** *1 – plain tape experimental data, 2 – plain tube*

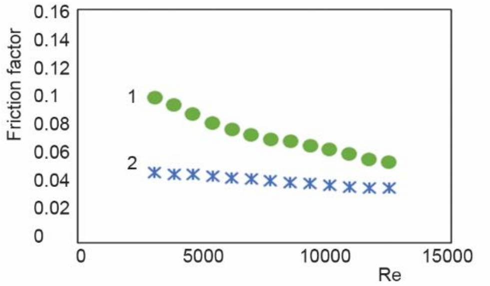


**Figure 8. Friction factor *vs.* Re for plain tape and plain tube under turbulent flow;** *1 – plain tape experimental data, 2 – plain tube*

are developed for Nusselt number and friction factor. The variation of the Nusselt number *vs.* Reynolds number in the tube fitted with PT-SCA, plain tape and also the plain tube are presented in fig. 10. The PT-SCA provides an additional disturbance to the fluid in the vicinity of the tube wall and leads to a higher heat transfer enhancement when compared with plain tube and plain tape.

From the present experiments, Nusselt number for the tube equipped with plain tape is 1.217 times greater than that of the plain tape and 1.288 times greater than that of the plain tube. The correlations for the Nusselt number, eq. (18), and friction factor, eq. (19) is developed to the present experimental results for the tube fitted with PT-SCA.

$$\mathrm{Nu}_{\mathrm{SCA}} = 0.0068\mathrm{Re}^{0.721}\mathrm{Pr}^{0.33} \qquad (18)$$

$$f_{\mathrm{SCA}} = 0.999\mathrm{Re}^{-0.2516} \qquad (19)$$

The predicted values agree with the experimental data within 6% and 5% variations, respectively, for Nusselt number and friction factor as shown in figs. 11 and 12.

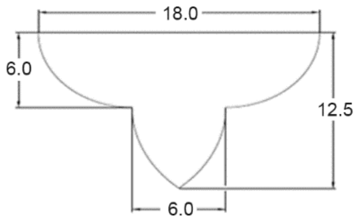


**Figure 9. Shape and size of the geometry on the PT-SCA**

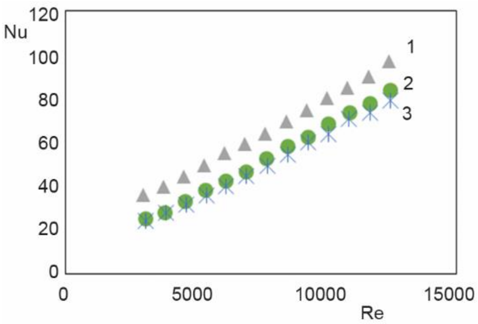


**Figure 10. The Nu *vs.* Re for PT-SCA, plain tape, and plain tube;** *1 – PT-SCA experimental data, 2 – plain tape experimental data, 3 – plain tube*

The heat transfer and friction factor characteristics for the tube fitted with PT-SCR is investigated experimentally under the turbulent flow condition at the test section. The geometry, size and shape are shown in fig. 13.

Nusselt number, fig. 14, in the tube with PT-SCR is higher than those in the plain tube and tube with plain tape, and PT-SCA insert over the range of Re = 2000 ≤ 14000. The PT-SCR provides an additional disturbance to the fluid in the vicinity of the tube wall and vortices behind the cuts and thus leads to a higher heat transfer enhancement in comparison with plain tube, plain tape and PT-SCA. The variation of Nusselt number with Reynolds

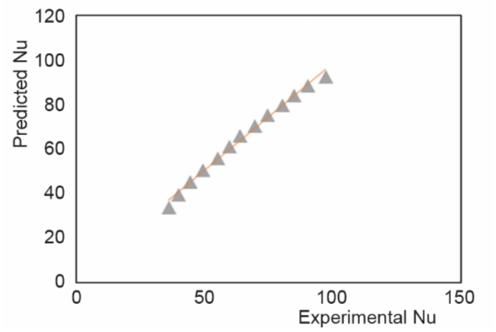


**Figure 11. The $Nu_{pre}$ *vs.* $Nu_{exp}$ for PT-SCA under turbulent flow**

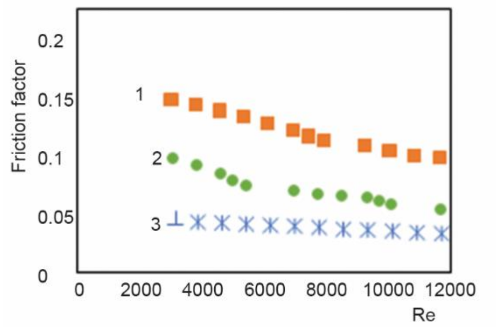


**Figure 12. Friction factor *vs.* Re for PT-SCR, plain tape and plain tube under turbulent flow;**
*1 – PT-SCR experimental data, 2 – plain tape, 3 – plain tube*

number in the tube fitted with PT-SCR, plain tape, and plain tube are presented in fig. 14. Mean Nusselt number for the tube equipped with PT-SCR is 1.38 times higher than that of plain tape and 1.46 times higher than that of plain tape.

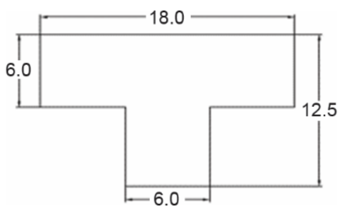


**Figure 13. Shape and size of the geometry on PT-SCR**

### *Effect of PT-SCR on heat transfer*

The correlations for Nusselt number, eq. 20, and friction factor, eq. 21, are developed to the present experimental results for the tube fitted with PT-SCR. The predicted values that settle with the experimental data within 9% and 10% for Nusselt number are shown in fig. 15:

$$\mathrm{Nu_{SCR}} = 0.0378\mathrm{Re}^{0.8}\mathrm{Pr}^{0.33} \tag{20}$$

$$f_{\mathrm{SCR}} = 1.0001\mathrm{Re}^{-0.238} \tag{21}$$

The predicted data agree with the experimental results for plain tape, PT-SCA, and plain tube. The predicted experimental values for frictional factor and Reynolds numbers are shown in fig. 16.

An experimental friction factor and the predicted frictional factor for plain tape step cut arc insert are shown in fig. 17.

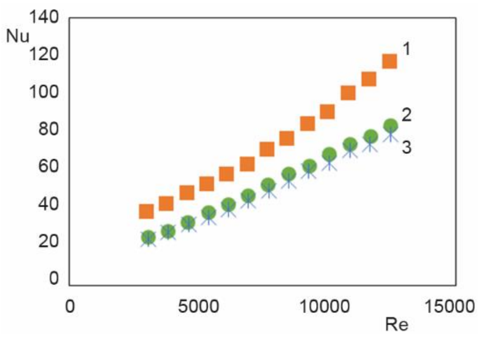


**Figure 14. The Nu *vs.* Re for PT-SCR, plain tape and plain tube under turbulent flow;**
*1 – PT-SCR experimental data, 2 – plain tape, 3 – plain tube*

### *Thermal enhancement factor for plain tape, PT-SCA, and PT-SCR*

The TEF, $\eta$, at equal pumping power is defined:

$$\eta = \left|\frac{h_{\mathrm{t}}}{h_{\mathrm{p}}}\right| \tag{22}$$

The TEF is evaluated since it is an important parameter indicating the potential of a plain tape for practical applications. Heat transfer enhancement is obtained at the expense of increased pressure drop caused by the tube insertions. Therefore, a performance analysis is

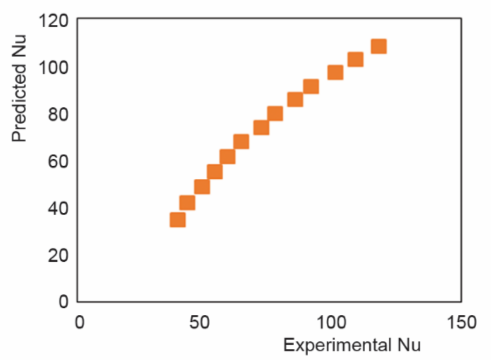


**Figure 15. The Nu$_{pre}$ *vs.* Nu$_{exp}$ for PT-SCR under turbulent flow**

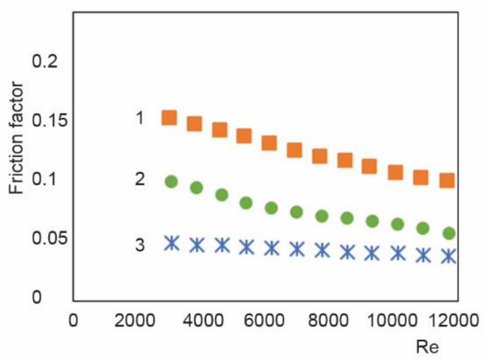


**Figure 16. Friction factor *vs.* Re for PT-SCA, plain tape, and plain tube;** *1 – PT-SCR experimental data, 2 – plain tape, 3 – plain tube*

important for the evaluation of the net energy gain to determine if the geometry of the plain tape employed to increase the heat transfer is effective from the energy point of view.

The overall TEF is shown in fig. 18, and the turbulent flow correlation equation of TEF for the plain tape, PT-SCA, and PT-SCR are expressed in eqs. (23)-(25).

$$\eta_{PT} = 2.0844\text{Re}^{-0.076} \tag{23}$$

$$\eta_{SCA} = 2.046\text{Re}^{-0.05} \tag{24}$$

$$\eta_{SCR} = 2.027\text{Re}^{-0.0362} \tag{25}$$

The experimental friction factor and the predicted frictional factor for plain tape step cut rectangle insert are shown in fig. 19.

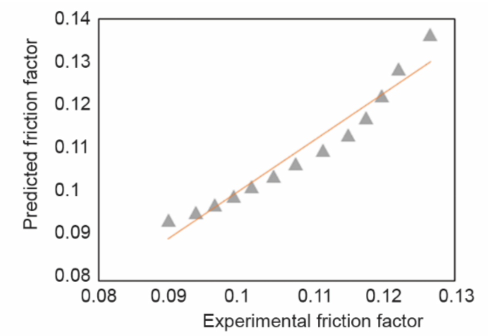


**Figure 17. The $f_{pre}$ *vs.* $f_{exp}$ for PT-SCA under turbulent flow**

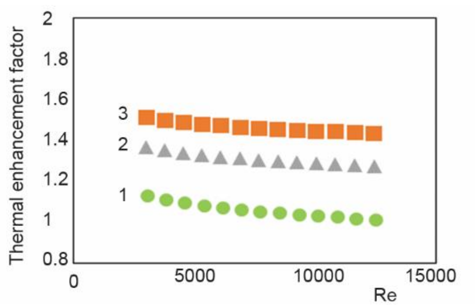


**Figure 18. The TEF *vs.* Re for plain tape, PT-SCA, and PT-SCR under turbulent flow;** *1 – plain tape, 2 – PT-SCA, 3 – PT-SCR*

The experimental investigations on heat transfer and friction factor characteristics for variants of plain tape inserts (PT-SCA and PT-SCR) fitted in the DPHE have been studied and presented. It is observed that the modifications on the plain tape *i. e.* small cut on the tape provides good results on the enhancement of both, heat transfer rate and thermal enhancement. Figure 20 shows the heat exchanger operating under turbulent flow condition where the flow rate is high.

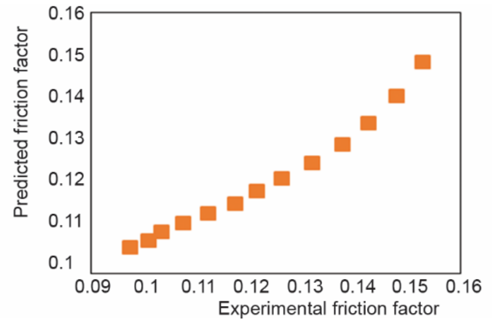


**Figure 19. The $f_{pre}$ *vs.* $f_{exp}$ for PT-SCR under turbulent flow**

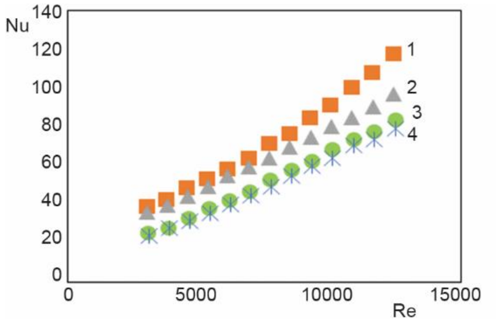


**Figure 20. The Nu *vs.* Re for plain tube, plain tape, PT-SCA, and PT-SCR under turbulent flow;** *1 – PT-SCR, 2 – PT-SCA, 3 – plain tape, 4 – plain tube*

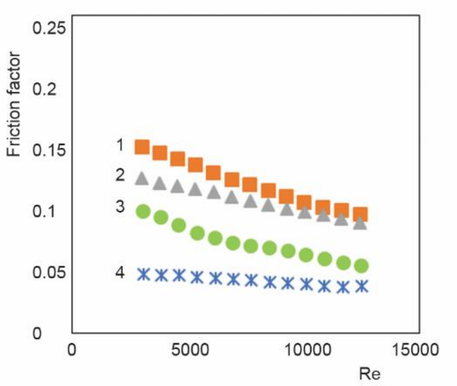


**Figure 21. Friction factor *vs.* Re for plain tube, plain tape, PT-SCA, and PT-SCR under turbulent flow;** *1 – PT-SCR, 2 – PT-SCA, 3 – plain tape, 4 – plain tube*

The plain tube fitted with plain tape inserts (PT-SCA and PT-SCR) led to increase the Nusselt number and decrease the friction factor with the increase in Reynolds number are shown in the fig. 21. It is the additional disturbance that influenced the improved thermal performance of the group of plain tape, PT-SCA, and PT-SCR which enhances turbulent intensity in the flow path that consequently leads to the enhancement in heat transfer.

## Conclusion

Nusselt numbers for the plain tube fitted with plain tape, PT-SCA, and PT-SCR are 1.056, 1.288, 1.460 times greater than that of plain tube, respectively.

Friction factors in the tube with plain tape, PT-SCA, and PT-SCR are 1.742, 2.537, 2.884 times higher than plain tube, respectively. The TEF for plain tape, PT-SCA, and PT-SCR are 1.063, 1.236, 1.382 times greater than that of plain tube, respectively. From the experimental results, it can be concluded that the TEF for the PT-SCR is 23.08% higher than that of plain tube and the PT-SCR insert gives the most improved performance in the DPHE compared to other profiles.

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