Heat and Mass Transfer (2018) 54:3137–3150
https://doi.org/10.1007/s00231-018-2347-x

ORIGINAL

Enhanced heat transfer and frictional losses in heat exchanger tube with modified helical coiled inserts

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Received: 28 November 2017 / Accepted: 5 April 2018 / Published online: 28 April 2018

Abstract

The application of compact heat exchangers in any thermal system improves overall performance with a considerable reduction in size and weight. Inserts of different geometrical features have been used as turbulence promoting devices to increase the heat transfer rates. The present study deals with the experimental investigation of heat transfer and fluid flow characteristics of a tubular heat exchanger fitted with modified helical coiled inserts. Experiments have been carried out for a smooth tube without insert, tube fitted with helical coiled inserts, and modified helical coiled inserts. The helical coiled inserts are tested by varying the pitch ratio and wire diameter ratio from 0.5–1.5, and 0.063–0.125, respectively for the Reynolds number range of 1400 to 11,000. Experimental data have also been collected for the modified helical coiled inserts with gradually increasing pitch (GIP) and gradually decreasing pitch (GDP) configurations. The Nusselt number and friction factor values for helical coiled inserts are enhanced in the range of 1.42–2.62, 3.4–27.4, relative to smooth tube, respectively. The modified helical coiled insert showed enhancements in Nusselt number and friction factor values in the range of 1.49–3.14, 11.2–19.9, relative to smooth tube, respectively. The helical coiled and modified helical coiled inserts have thermo-hydraulic performance factor in the range of 0.59–1.29, 0.6–1.39, respectively. The empirical correlations of Nusselt number and friction factor for helical coiled inserts are proposed.

Nomenclature

A_c	Cross sectional area m^2
A_s	Surface area m^2
C_p	Specific heat at constant pressure J/kg K
d	Inner diameter of test section m
e	Wire diameter m
e/d	Wire diameter ratio -
f	Friction factor -
GIP	Gradually increasing pitch of helical coil -
GDP	Gradually decreasing pitch of helical coil -
h	Convective heat transfer coefficient W/m^2 K
k	Thermal conductivity of water W/m K
L	Length of the test section m
$\dot{m}$	Mass flow rate kg/s
Nu	Nusselt number -
p	Pitch of wire m
p/d	Pitch ratio -
Δp	Pressure difference across test section N/m^2
Q	Rate of heat transfer W
Re	Reynolds number of water -
T_s	Mean surface temperature of test section K
T_b	Bulk mean temperature of water K
T_i	Water temperature at inlet of test section K
T_o	Water temperature at outlet of test section K
U	Mean velocity of water m/s

Greek symbols

ν	Kinematic viscosity of water m^2/s
ρ	Density of water kg/m^3
η	Thermo-hydraulic performance factor −

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1 Introduction

A variety of heat exchangers have been developed for the use at different levels of technological complexity and sizes. Recuperative type heat exchanger is perhaps the simplest among widely used heat exchangers in which the two fluids are separated by a highly conductive solid wall. Heat transfer process occurs in a tubular heat exchanger due to the combined mode of conduction and convection from hot to cold fluids which are separated by a thin wall. The boundary layer formed over the hot wall offers a thermal resistance which lowers the heat transfer rate between the heated surface and the fluid stream. An efficient heat exchanger requires mainly the higher thermal performance with least pumping power for compact and economical design. Hence, the efforts must be directed towards improvement of heat duty of a convectional heat exchanger for the transport of a required energy at the optimal size and cost.

Researches are being carried out in order to improve the performance of heat exchangers by the application of heat transfer augmentation techniques. Several heat transfer augmentation methods have been tested so far, but the application of passive techniques showed considerable rise in the heat transfer rate with the moderate penalty of frictional losses. The passive techniques can be applied by way of incorporating modified geometries or modified surfaces such as treated surface, rough surface, extended surfaces, coiled tube, etc. Inserts are widely used as passive heat transfer augmentation devices and can be of any geometries like twisted tapes, delta winglet twisted tapes, conical rings, v-nozzles, swirling jets, helical coiled inserts, etc.

San et al. [1] performed the experiments for a smooth tube with coiled wire inserts and compared the heat transfer enhancement for different working fluids. It has been reported that the heat transfer enhancement considerably decreases with an increase in the values of Reynolds number if water is used as working fluid in place of air. An increase in the wire diameter ratio and decrease in the pitch ratio leads to increase in the Nusselt number at all values of Reynolds number. Keklikcioglu et al. [2] investigated the thermo-hydraulic performance of a circular tube with coiled wire inserts while a gap maintained between the inner wall of the tube and insert. The tube fitted with coiled wire inserts yielded a significantly higher heat transfer rates and friction factor than that of a smooth tube. The maximum thermo-hydraulic performance was observed for the insert having pitch ratio of 1 while the clearance of 1 mm is maintained between the tube and insert. The study concluded that the higher disturbances in the laminar boundary layer are induced by the presence of coiled wire inserts with clearance. Sheikholeslami et al. [3] experimentally investigated the performance of air to water double pipe heat exchanger with typical circular-ring and perforated circular-ring turbulators. The results revealed that the perforated circular-ring turbulator produces lower heat transfer enhancement than the typical circular-ring turbulator due to the considerable changes in the velocity and the temperature fields. In a study by Tu et al. [4], it was revealed that the pipe inserts can transfer greater heat energy for same pumping power as compared to the other type of inserts due to their unique structure.

Bhuiya et al. [5] evaluated the heat transfer and fluid flow performance in a tube with twisted wire brush inserts. It was observed that the application of twisted wire brush inserts led to a considerable increase in the heat transfer enhancement with corresponding increase in frictional losses over the plain tube. You et al. [6] investigated the thermo-hydraulic performance of conical strip inserts and found that the average Nusselt number and friction factor are function of the geometry angle and the strip pitch. Promvonge et al. [7] carried out experimental study to examine the heat transfer augmentation in a tube with inclined vortex rings. The vortex rings were found to produce counter-rotating vortices inside the tube that increases the turbulence intensity and fluid mixing. Kongkaitpaiboon et al. [8] experimentally investigated that effect of perforated conical-ring on the performance of heat exchanger and reported that the thickness of thermal boundary layer reduces and thereby enhances the heat transfer rate over the plain tube.

Promvonge et al. [9] performed experiments to show the effect of a conical-nozzle insert provided with free-spacing snail entry. It was reported that the conical-nozzle insert with free-spacing snail entry brought out considerable enhancements in heat transfer and friction over the plain tube. Eiamsa-ard et al. [10] used V-nozzle inserts in a tube to explore its effects on heat transfer and friction characteristics and found a considerable increase in the heat transfer rate over the plain tube. It was reported that the presence of V-nozzle produced the reverse flow that improved the heat transfer rate in the tube. In a study by Eiamsa-ard et al. [11], the effect of different types of delta-winglet twisted tapes are studied. The results showed that the Nusselt number and friction factor in a tube carrying the delta-winglet twisted tape are increased with the decrease in twisted ratio and increase in depth of wing cut ratio. Eiamsa-ard et al. [12] experimentally investigated tube flow with the combination of twisted tape and wire coil inserts. The empirical correlations of the Nusselt number and friction factor were also presented. Chang et al. [13] in an experimental study measured the axial heat transfer distributions and the pressure drop coefficients of the tube fitted with broken twisted tapes. The local Nusselt number and mean Fanning friction factor in the tube carrying the broken twisted tape were increased as the twist ratio was decreased. Eiamsa-ard et al. [14] performed experiments to evaluate the performance of tapered twisted tapes and found that the heat transfer and frictional losses were increased with decreasing twist ratio and taper angle. The thermal performance factor tends to increase with decreasing tape twist ratio while taper angle is increased. Nanan et al. [15] investigated that the presence of perforated helical twisted tapes leads to the reduction in friction losses as compared to that of helical twisted tapes. Bhuiya et al. [16] found in an experimental study that the tube fitted with perforated twisted tapes brought out significant augmentation in heat transfer rate and friction factor than that of the plain tube. A study by Eiamsa-ard et al. [17] revealed that the straight tape having delta wings on both sides with alternate axis has higher heat transfer rate and friction factor than that of the tape without alternate axis. Promvonge et al. [18] experimentally investigated that the application of wire coil insert with the twisted tapes leads to a two-fold enhancement in heat transfer as compared to the wire coil insert or twisted tape insert. Garcia et al. [19] observed in an experimental study that in the laminar flow regime, wire coil insert behaves as smooth tube, and in the transition region, higher heat transfer rates were obtained while keeping the constant pumping power.

The literature survey clearly reflects the progress made towards the exploration of heat exchanger characteristics with different classes of tube inserts. It is invariably observed in all cases that the heat transfer is augmented by the application of inserts with the simultaneous rise in pressure drop. It is therefore imperative to identify the insert geometry which can mitigate the augmentation in frictional losses while yielding the considerably higher heat transfer rates. It came into view from some recent studies that the application of wire coil inserts with uniform pitch brought out notable heat transfer enhancements at the expense of moderate pumping power in comparison to other swirl flow devices. However, the effects of gradually increasing and gradually decreasing pitch of helical coiled inserts are not reported in the literature and thus it is imperative to further explore the effect of helical coiled and modified helical coiled inserts. The present work deals with the investigation of heat transfer and fluid flow characteristics of tube fitted with helical coiled inserts having different pitch ratio (p/d) and wire diameter ratio (e/d). The performance of modified helical coiled inserts with gradually increasing and gradually decreasing pitch configurations are also discussed in this work. The experimental data for different geometrical parameters of inserts have been collected by varying the Reynolds number from 1400 to 11,000.

2 Experimental investigation

An open loop experimental test setup is designed and fabricated according to the guidelines laid down in the literature [20]. In order to validate the experimental setup, the data pertaining to heat transfer and friction are collected for a smooth tube by varying the flow rate. After ensuring the legitimacy of data collected from the experimental setup, the data pertaining to heat transfer and friction for different types of helical coiled inserts are collected under varied fluid flow conditions. The raw data collected during the test runs are utilized to compute the dimensionless key parameters representing the heat transfer, fluid flow, and friction.

2.1 Experimental test setup and procedure

The experiments are carried out by using water as the working fluid to study the heat transfer and friction factor characteristics of heat exchanger tube with different types of helical coiled inserts. The test facility includes a water tank, a circular pipe for supplying water to the test section, a rotameter to measure the flow rate, a claiming tube and the test section as shown in Fig. 1. A water tank of 200 l capacity is placed at 2.5 m height to store water for experimental test run. The inlet section of 1500 mm length is used to calm down the disturbances in the fluid stream and to ensure the fully developed flow. The water to the inlet section is directed by the ball valve through the calming tube. The volumetric flow rate of water is measured with the aid of a glass tube rotameter that has been calibrated before the experiments with the help of a measuring flask. A copper tube of 1000 mm length and 24 mm internal diameter is used as test section. The test section is heated by providing the uniform heat flux over the outer surface of the tube with the help of heating element fabricated by using Nichrome wire loops. The electrical input power is controlled with the help of a variac transformer by regulating the voltage across the heating element. The outer surface of the test section is well insulated with rubber foam insulation to minimize heat loss to the surroundings. The tube wall temperatures are measured by fixing six T-type thermocouples over the tube surface and are calibrated before being used. The inlet and outlet temperatures of fluid are also measured by insertion type thermocouples. The pressure drop across the test section is measured by using a digital micro-manometer.

The experiments are carried out by varying the fluid flow rate through the system while providing the uniform heat flux over the test section. For a fixed flow rate, the test section is heated by provided uniform heat flux during the experiments and the measurable parameters like pressure drop across test section, temperature, volumetric flow rate, and electrical energy input are recorded after the system approached to the steady state condition. The hot water coming out from the test section is drained out through the canal during the experiments. The Reynolds number, friction factor and Nusselt number are obtained by using the measured parameters. Table 1 shows the range of parameters considered in the present investigation.

2.2 Geometry of helical coiled inserts

During the experimentation process, helical coiled inserts having different pitch ratio (p/d) and wire diameter ratio (e/d) are inserted into the circular tube in order to collect the pertinent data. Galvanized iron (GI) wires having diameter (e) of 1.5, 2, 2.5, 3 mm are used to fabricate the helical coiled inserts by machining process on lathe. The pitch ratio (p/d) is defined as the ratio of pitch length of helical coiled insert to the inner diameter of pipe, while the wire diameter ratio (e/d) indicates the ratio of wire diameter to the inner diameter of the tube. The modified helical coiled inserts are manufactured by gradually varying the pitch length of helical coiled inserts. The modified

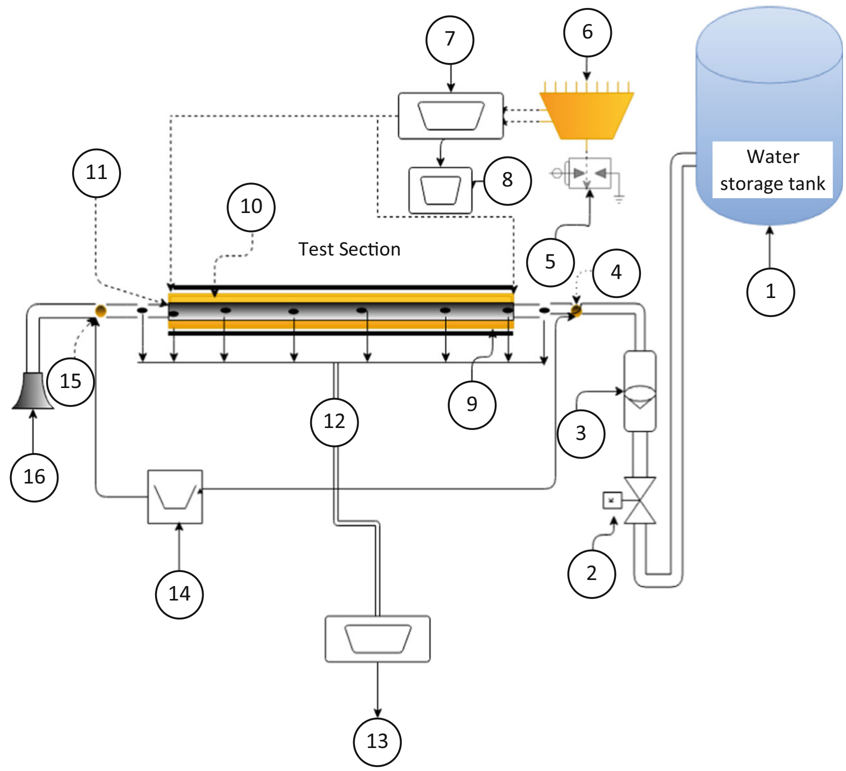


Fig. 1 Schematic view of experimental setup

1. Water storage tank
2. Ball valve
3. Rotameter
4. Pressure tap at inlet
5. Power supply
6. Variac
7. Ammeter
8. Voltmeter
9. Insulation
10. Heating element
11. Exit section
12. Thermocouples
13. Temperature indicator
14. Digital Micro-manometer
15. Pressure tap at exit
16. Discharge

helical coiled insert used in experimental work has two variants as gradually increasing pitch and gradually decreasing pitch. The details of geometry of helical coiled inserts and modified helical coiled inserts are shown in Fig. 2.

3 Data reduction

The experimental data of fluid and tube surface, pressure drop across test section have been recorded after reaching the steady state conditions for different flow rates. The measured data is used to evaluate the values of dimensionless parameters like friction factor, Nusselt number, and Reynolds number. The following method is used to obtain the dimensionless parameters representing the fluid flow, heat transfer, and friction.

The working fluid flows through the heated tube and thus the heat transfer by convection (Q_c) from the heated tube surface is equal to heat gain by water (Q_w), under adiabatic condition.

$$Q_w = Q_c \tag{1}$$

The heat received by water can be written as

$$Q_w = \dot{m}\, C_p\, (T_{out}–T_{in}) \tag{2}$$

The convective heat transfer rate from the heated surface of the tube is given by:

Table 1 Range of parameters

Parameters	Specification	Range
Pitch ratio (p/d)	Ratio of distance between two coil and diameter of spring (p/d)	0.5,1 and 1.5
Wire diameter ratio (e/d)	Diameter of wire to inner diameter of test section (e/d)	0.063–0.125
Reynolds Number (Re)	Flow parameter	1400–11,000

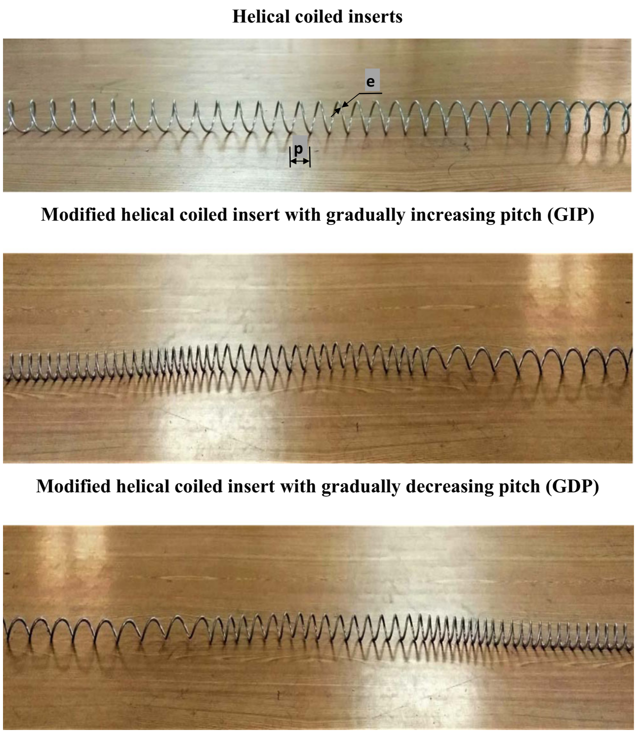


Fig. 2 Details of helical coiled insert geometry

$$Q_c = h\, A_s\, (T_s - T_b) \tag{3}$$

where A_s is the surface area of tube, h is the convective coefficient of heat transfer, T_s is the average temperature of the tube surface, T_b is the mean bulk temperatures of water.

The mean pipe surface temperature T_s is the arithmetic mean of all temperatures measured by thermocouples placed over the test section.

$$T_s = \frac{T_1 + T_2 + T_3 + T_4 + T_5 + T_6}{6} \tag{4}$$

The mean bulk temperature of water (T_b) is the arithmetic mean of water temperature measured at the inlet and exit of test section.

Thus

$$T_b = \frac{T_i + T_o}{2} \tag{5}$$

The mean velocity of water flowing through the tube can be determined as:

$$U = \frac{\dot{m}}{\rho . A_c} \tag{6}$$

Reynolds number for the mean velocity of water (U) can be expressed as:

$$\mathrm{Re} = \frac{UD}{v} \tag{7}$$

The friction factor is obtained from the pressure drop $(\Delta P)_d$ in the test section by using the Darcy equation as:

$$f = \frac{(\Delta P)}{\left\{ \left({}^{L}/_{d} \right) \left({}^{\rho U^2}/_{2} \right) \right\}} \tag{8}$$

The average heat transfer coefficient is calculated by the following expression:

$$h = \frac{Q_{water}}{A_s (T_s - T_b)} \tag{9}$$

The convective heat transfer coefficient (h) is used to calculate the average Nusselt number as follows:

$$Nu = \frac{hD}{k} \tag{10}$$

The value of thermo-hydraulic performance factor (η) is calculated by using the following expression:

$$\eta = \left[\frac{Nu/Nu_s}{(f/f_s)^{1/3}} \right] \tag{11}$$

Where Nu and Nu_s are the Nusselt number for tube with insert and smooth tube without insert, respectively. Similarly f and f_s are the friction factor for tube with insert and smooth tube, respectively.

An uncertainty analysis of experimental results has been carried out by following the guidelines proposed by Kline and McClintock [21]. This method is based on estimation of uncertainties in the primary experimental measurements. The uncertainty estimates for the key parameters obtained for heat exchanger tube with the modified helical coiled inserts are given in Table 2.

3.1 Validation test

A thorough check of the measuring instruments and the experimental set up is ensured before the experimentation on smooth tube without insert. The mean values of friction factor and Nusselt number are determined by using the data

Table 2 Uncertainty analysis

Parameters	% Uncertainty
Heat transfer coefficient (*h*)	± 3.54%
Mass flow rate ($\dot{m}$)	± 1.61%
Nusselt number (*Nu*)	± 5.31%
Friction factor (*f*)	± 4.5%
Reynolds number (*Re*)	± 1.7%
Useful heat gain (*Q*)	± 3.5%

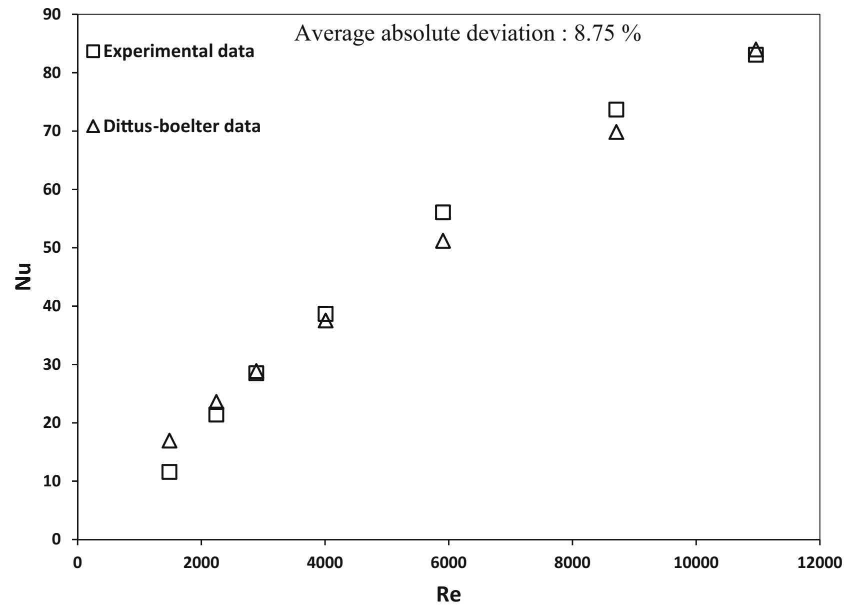


Fig. 3 Comparison of experimental values of Nusselt number from Dittus-Boelter data

reduction procedure. The experimental values of friction factor and Nusselt number have been compared with the values obtained from the suggested correlations such as Dittus-Boelter equation [22] for Nusselt number and modified Blasius equation [22] for friction factor for the smooth tube flow. The reference data for smooth tube are obtained from the following equations:

$$Nu = 0.023\ Re^{0.8}\ Pr^{0.4} \tag{12}$$

$$f = 0.316\ Re^{-0.25} \tag{13}$$

The experimental values of Nusselt number and friction factor are plotted along with the reference data for the entire range of Reynolds number in Figs. 3 and 4, respectively. These plots show that the experimental results obtained from the experimental set up are closely running with the standard data for all the values of Reynolds number. The experimental values of Nusselt number deviates from Dittus-Boelter data by 8.75% while the deviation of the experimental values friction factor from Blasius data is found to be 11%. The validation test reveals that the experimental data is in good agreement with that of the standard values and thus can be used for data collection in case of a smooth tube with inserts.

4 Results and discussion

The convective heat transfer from the tube fitted with different helical coil configurations is analyzed and the

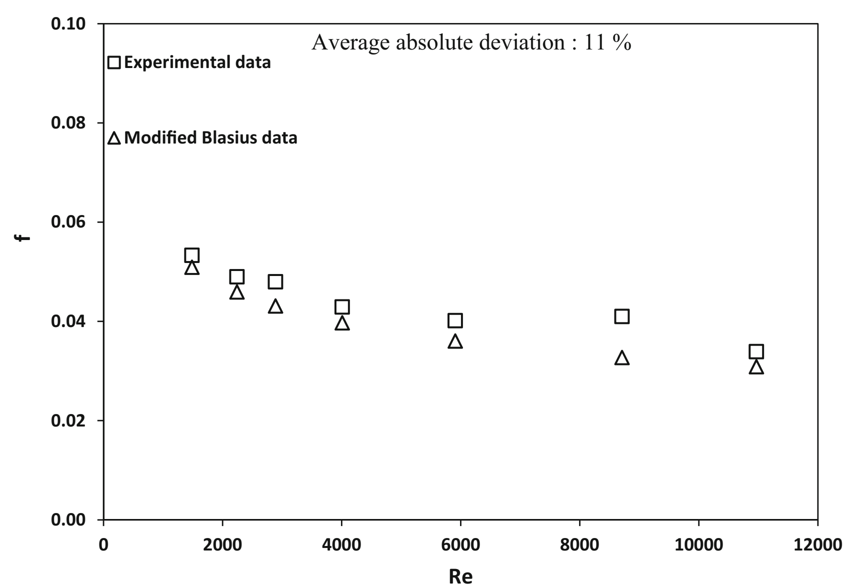


Fig. 4 Comparison of experimental values of friction factor from modified Blasius data

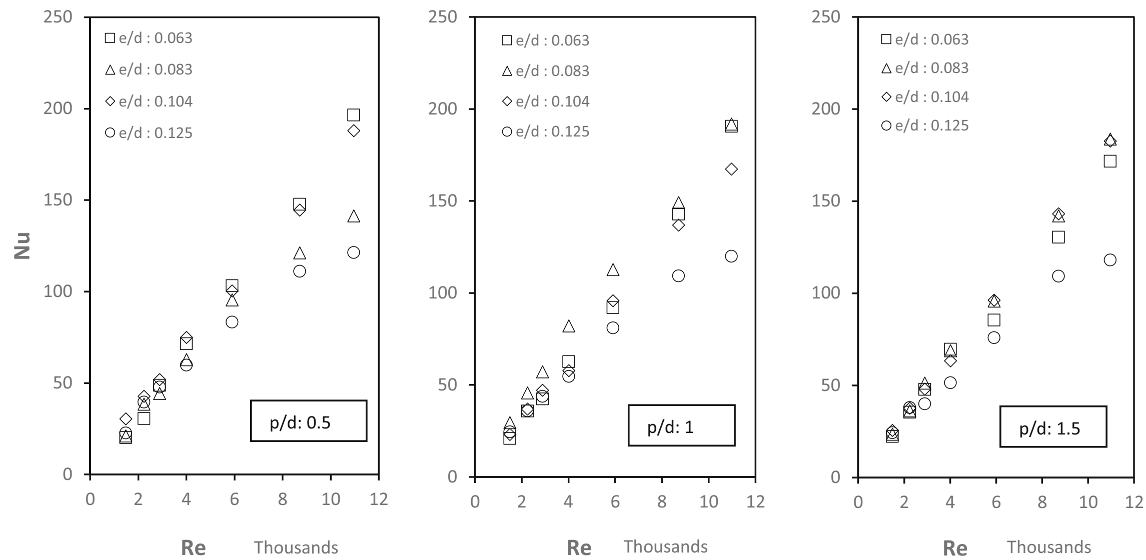


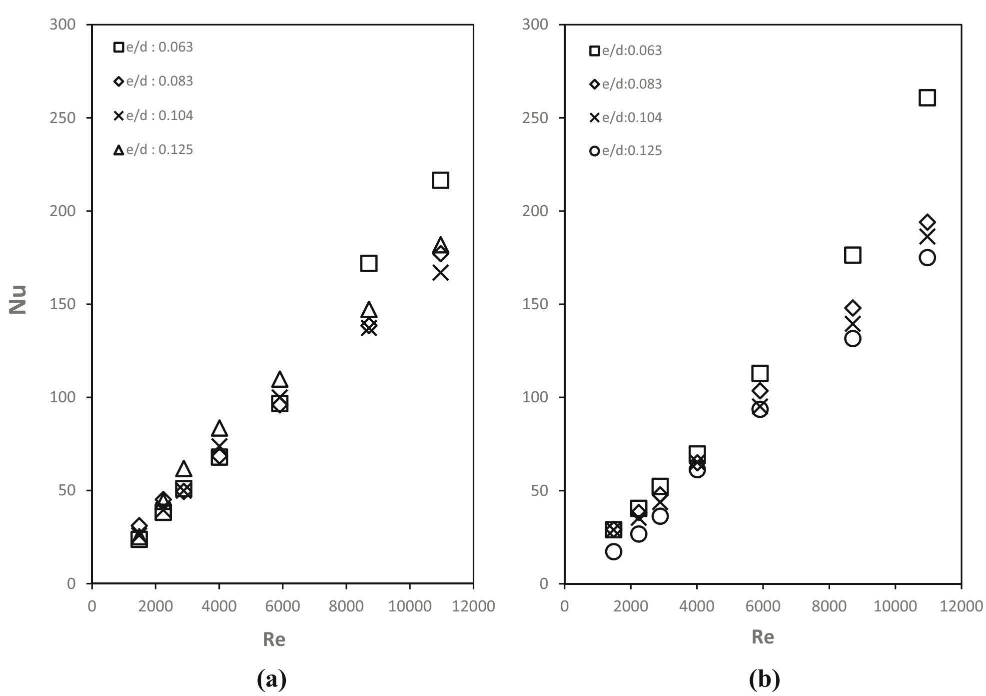
Fig. 5 Nusselt number as function of Reynolds number for different geometrical parameters of helical coiled inserts

corresponding enhancements in heat transfer with respect to smooth tube are discussed. It is acknowledged that the geometrical parameters, viz., pitch ratio (p/d) and wire diameter ratio (e/d) have significant impact on the heat transfer from the tube. The heat transfer characteristics of modified helical coiled inserts having wire diameter ratio (e/d) of 0.063, 0.083, 0.104, and 0.125 with the gradually increasing pitch (GIP) from 12 to 36 mm and with the gradually decreasing pitch (GDP) from 36 to 12 mm be also discussed in this section.

4.1 Nusselt number enhancement

Figure 5 shows the Nusselt number variation as function of Reynolds number for the pitch ratio (p/d) varied in the range of 0.5 to 1.5 and the wire diameter ratio (e/d) varied in the range of 0.063 to 0.125. The plot shows that the Nusselt number increases with the increase in Reynolds number for all the geometrical parameters of helical coiled inserts. The minimum value of Nusselt number corresponds to the (p/d) of 1.5 for all the values of (e/d), while the maximum values of Nusselt number are found for the combination of (p/d) of 0.5 and

Fig. 6 Variation of Nusselt number for modified helical coiled inserts with (**a**) gradually increasing pitch and (**b**) gradually decreasing pitch



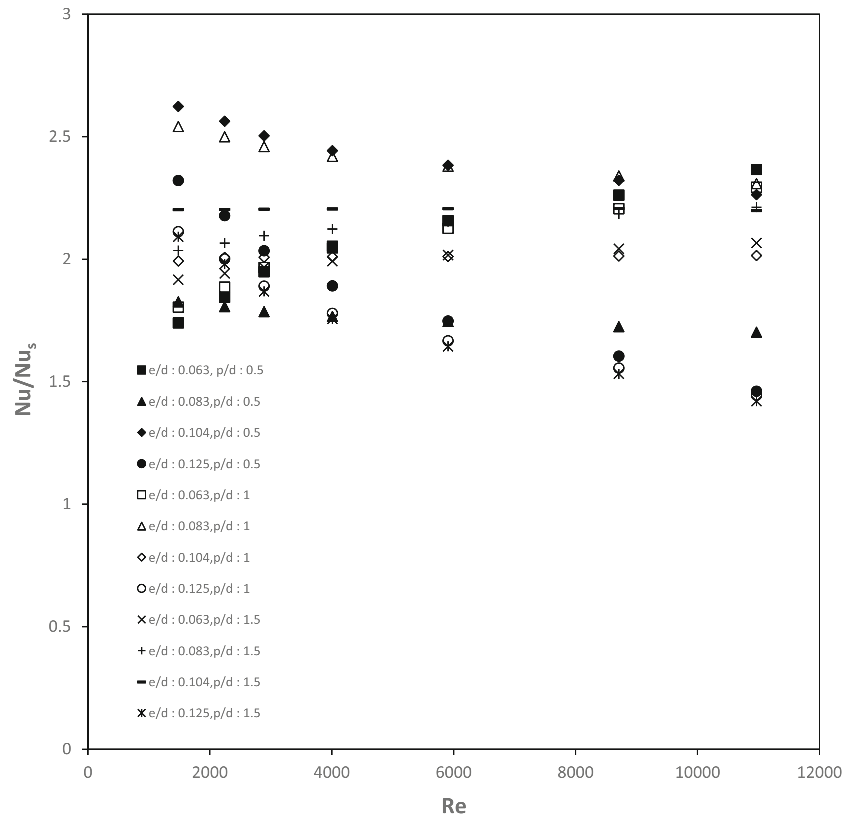


Fig. 7 Variation of Nusselt number enhancement ratio for different geometrical parameters of helical coiled inserts

(e/d) of 0.063. It is observed that as pitch ratio (p/d) approaches to higher value, there is a notable decrease in the heat transfer rate.

Figure 6a exhibits the Nusselt number variation with the change in the Reynolds number for wire diameter ratio (e/d) in case of modified helical coiled inserts with gradually increasing pitch. The plot depicts that the maximum heat transfer rate corresponds to the (e/d) of 0.125, for Reynolds number less than 6000 whereas the (e/d) of 0.063 yields maximum heat transfer rate when the Reynolds number approaches to higher values. The effect of Reynolds number on the Nusselt number for different values of (e/d) in case of modified helical coiled insert with gradually decreasing pitch can be seen in Fig. 6b. The plot clearly shows that the maximum and minimum values of Nusselt number correspond to (e/d) of 0.063 and 0.125, respectively, irrespective of the change in Reynolds number. It is seen that the increase in the Nusselt number is aggravated as Reynolds number approaches to the higher values, especially in case of the wire diameter ratio (e/d) of 0.063.

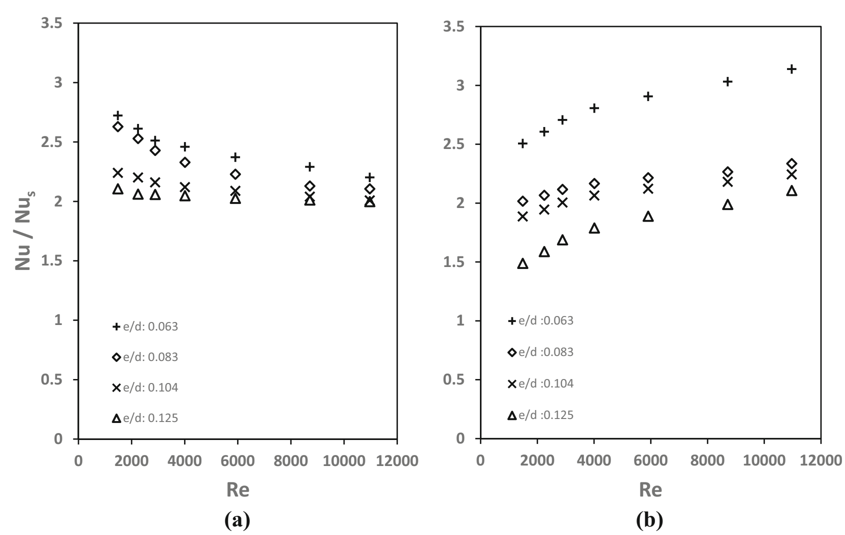


Fig. 8 Variation of Nusselt number enhancement ratio for modified helical coiled inserts with (**a**) gradually increasing pitch and (**b**) gradually decreasing pitch

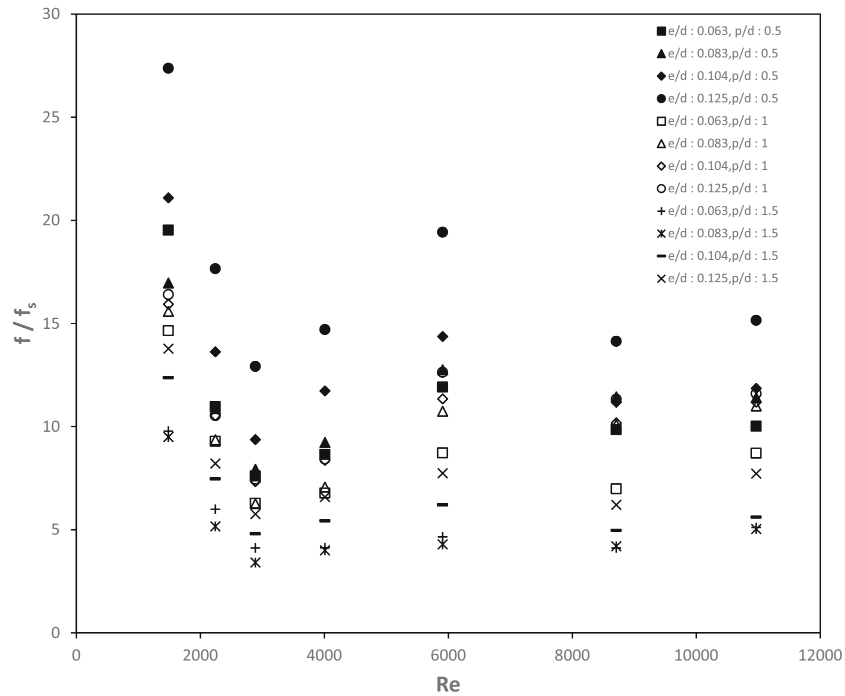


Fig. 9 Friction factor enhancement ratio as function of Reynolds number for different geometrical parameters of helical coiled inserts

Figure 7 shows the effect of Reynolds number on the Nusselt number enhancement ratio (Nu/Nu_s) for the pitch ratio (p/d) varied in the range of 0.5 to 1.5 and the wire diameter ratio (e/d) varied in the range of 0.063 to 0.125. The minimum value of Nusselt number enhancement ratio (Nu/Nu_s) corresponds to the (p/d) of 1.5 and (e/d) of 0.125, while the maximum values of Nusselt number enhancement (Nu/Nu_s) are obtained from the combination of (p/d) of 0.5 and (e/d) of 0.104. It is observed that the Nusselt number enhancement ratio (Nu/Nu_s) lie in the range of 1.42–2.62.

Figure 8a exhibits the effect of Reynolds number on the Nusselt number enhancement ratio (Nu/Nu_s) for different values of the wire diameter ratio (e/d) in case of modified helical coiled insert with gradually increasing pitch. The plot depicts that the maximum Nusselt number enhancement ratio (Nu/Nu_s) corresponds to the (e/d) of 0.063 for all values of the Reynolds number, whereas the (e/d) of 0.125 has minimum Nusselt number enhancement ratio (Nu/Nu_s) for the entire range of Reynolds number. The variation of Nusselt number enhancement ratio (Nu/Nu_s) for different values of (e/d) in case of modified helical coiled insert with gradually decreasing pitch can be seen in Fig. 8b. The plot clearly shows that the maximum and minimum values of the

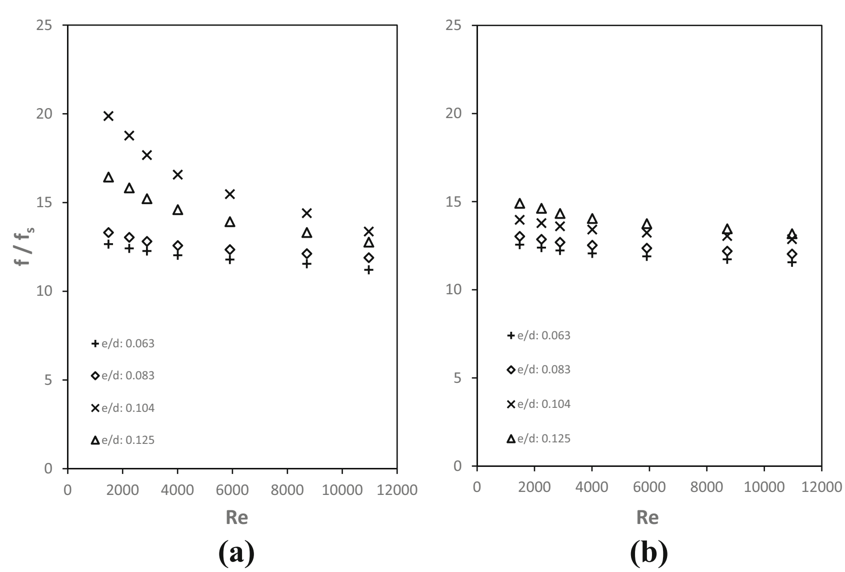


Fig. 10 Effect of Reynolds number on friction factor enhancement ratio for modified helical coiled inserts with (**a**) gradually increasing pitch and (**b**) gradually decreasing pitch

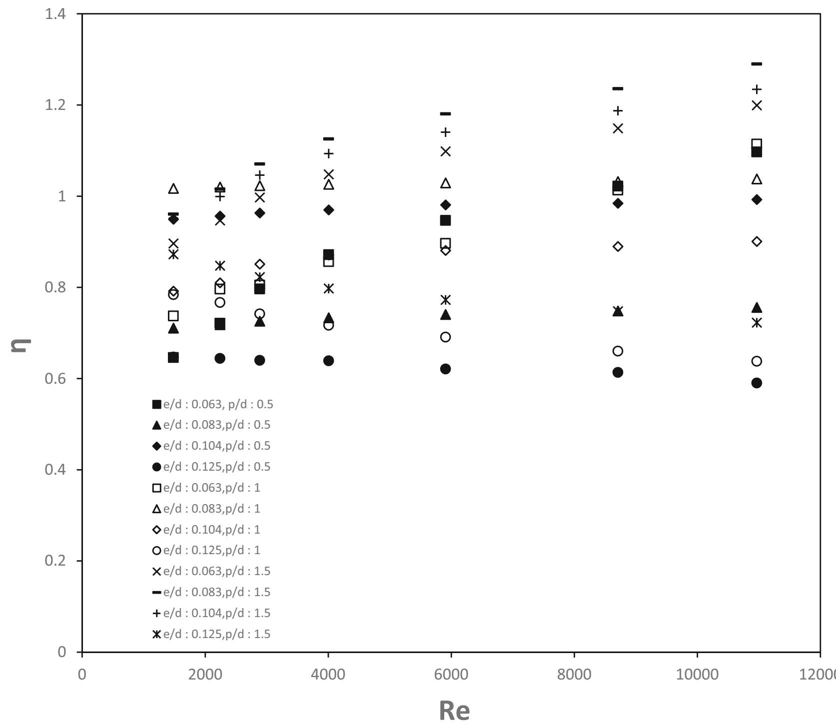


Fig. 11 Thermo-hydraulic performance as function of Reynolds number for different geometrical parameters of helical coiled inserts

Nusselt number enhancement ratio (Nu/Nu_s) are obtained for the (e/d) of 0.063 and 0.125, respectively, irrespective of the change in Reynolds number. It can be noticed that the Nusselt number enhancement ratio (Nu/Nu_s) is increased as Reynolds number approaches to the higher values in all cases. The plot reveals that the modified helical coiled insert with (e/d) of 0.063 has significantly higher enhancements in Nusselt number in comparison to other values of (e/d).

A careful survey of Fig. 8a, b show that the gradual increase in pitch of modified helical coiled insert leads to a decrement in the Nusselt number enhancements when the Reynolds number is increased. However, Nusselt number enhancement ratio (Nu/Nu_s) ascends by using gradually decreasing pitch geometry with an increase in the Reynolds number. It may be due to the fact that the turbulence intensity is highly sensitive to the change in pitch length of modified insert under varied fluid flow rates.

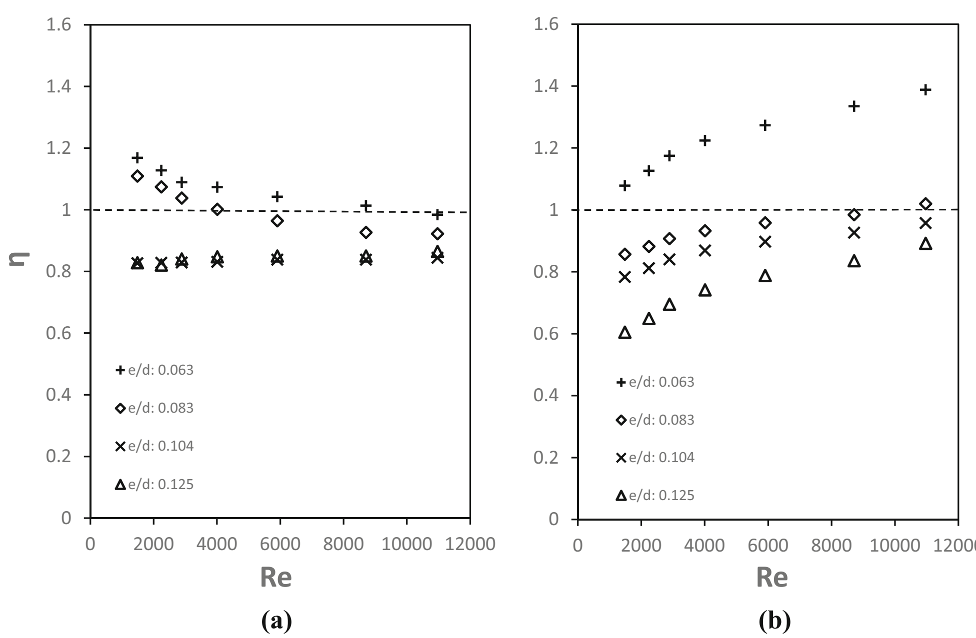


Fig. 12 Thermo-hydraulic performance for modified helical coiled inserts with (**a**) gradually increasing pitch and (**b**) gradually decreasing pitch

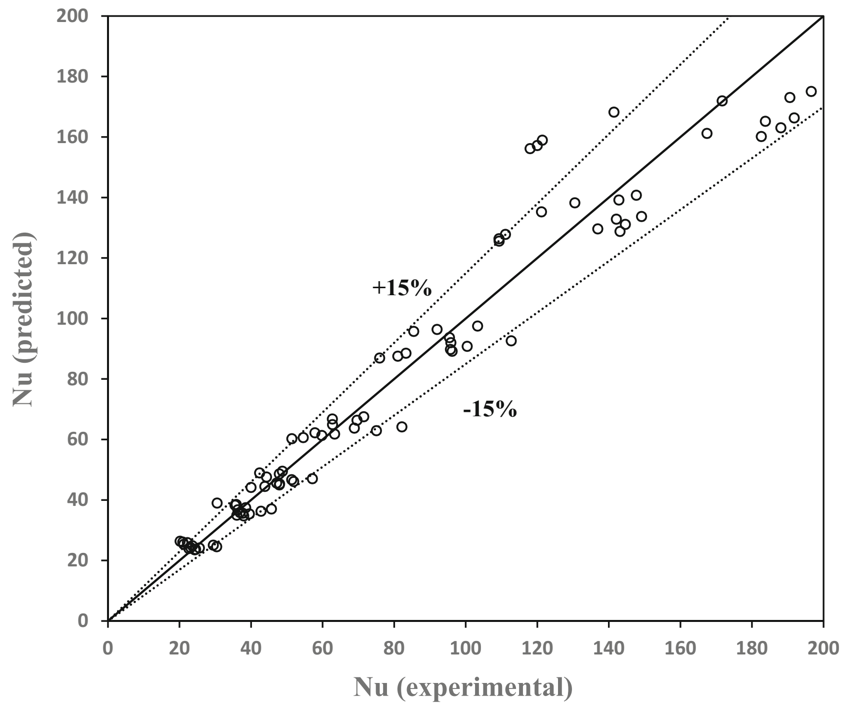


Fig. 13 Comparison of experimental and observed values of Nusselt number

4.2 Friction factor enhancement

The determination of friction factor (f) is necessary to gauge the net gain in the performance while a passive technique of heat transfer enhancement is used. The friction factor is directly proportional to the pressure drop across the test section and inversely proportional to the square of fluid velocity. Thus, a lower value of friction factor corresponds to the higher Reynolds number and vice-versa. The application of inserts in a heat exchanger tube brings out heat transfer enhancement with simultaneous rise in the pressure drop due to higher disturbances in the path of fluid flow. Consequently, the overall performance of heat exchanger degrades to a large extent. In view of this observation, the heat transfer enhancements are to be brought out in such a manner by which the frictional losses can be kept under prescribed limits. With this objective, the

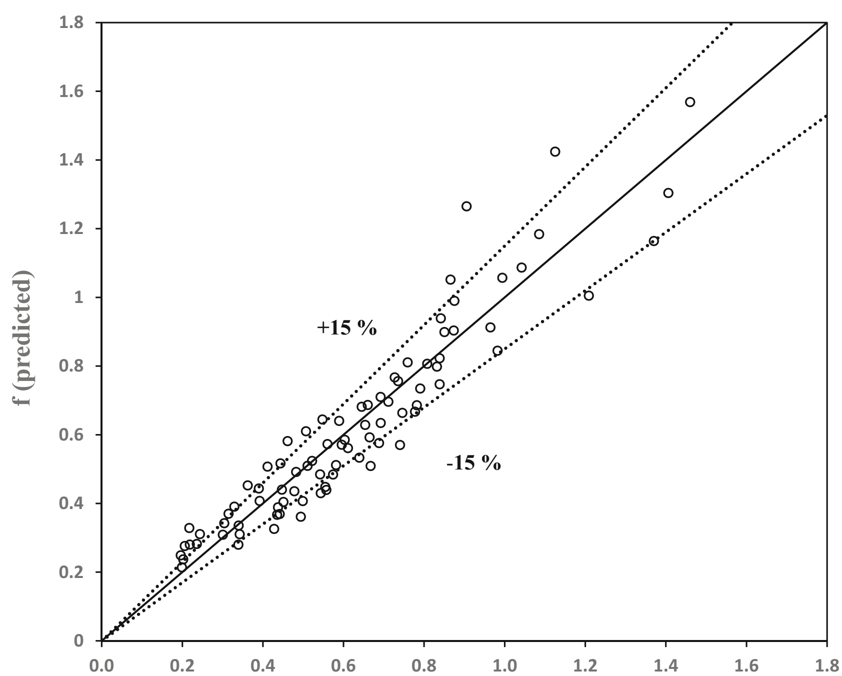


Fig. 14 Comparison of experimental and observed values of friction factor

effects of geometrical parameters of different types of helical coiled inserts on the friction factor are explored for the broad range of flow Reynolds number.

The plot depicting the effect of Reynolds number on the friction factor enhancement ratio (f/f_s) for different geometrical parameters of helical coiled insert can be seen in Fig. 9. The friction factor in case of tube with helical coiled insert having a pitch ratio (p/d) of 0.5 and the wire diameter ratio (e/d) of 0.125 is augmented to around 27 times than that of a smooth tube without insert. An increase in the pitch ratio (p/d) values causes a decrease in the friction factor enhancements. The maximum values of friction factor enhancements ratio (f/f_s) corresponds to the insert having pitch ratio (p/d) of 0.5 and wire diameter ratio (e/d) of 0.125, while the minimum friction factor enhancements ratio (f/f_s) corresponds to the insert having pitch ratio (p/d) of 1.5 and wire diameter ratio (e/d) of 0.083. It is evident from the above discussion that in order to reduce the enhancement in friction, there should be a minimum flow blockage, which happens in case of (p/d) equal to 1.5 and (e/d) equal to 0.083.

Figure 10a shows the plot of friction factor enhancement ratio (f/f_s) as function of Reynolds number for modified helical coiled insert with gradually increasing pitch. The friction factor enhancement rose to a value of around 20 when an insert having wire diameter ratio (e/d) 0.104 is used. The maximum and minimum values of friction factor enhancement corresponds to wire diameter ratio (e/d) of 0.104 and 0.063, respectively for all values of the Reynolds number. A plot of friction factor enhancement ratio (f/f_s) as a function of Reynolds number for modified helical coiled insert with gradually decreasing pitch is exhibited in Fig. 10b. The friction factor enhancement ratio reaches around 15 when an insert having wire diameter ratio (e/d) 0.125 is used. An increase in the wire diameter ratio (e/d) leads to the increment in the friction factor enhancement ratio (f/f_s). The maximum and minimum values of friction factor enhancement corresponds to wire diameter ratio (e/d) of 0.125 and 0.063, respectively for all the values of Reynolds number.

4.3 Thermo-hydraulic performance

Thermo-hydraulic performance factor (η) is the important parameter which signifies the actual enhancement attained by the application of inserts in a heat exchanger. Since there is a substantial enhancement observed in the value of heat transfer and friction for helical coiled and modified helical coiled insert geometries, therefore it is imperative to identify the geometry, which can exhibit thermo-hydraulic performance factor preferably more than unity. Han et al. [23] and Webb et al. [24] proposed a method for evaluating the relative performance of an enhanced heat transfer surface in which the enhancement in heat transfer brought out by the application of passive technique is compared to that of the smooth surface under fixed pumping power consumption.

A plot shown in Fig. 11 portrays the variation of Thermo-hydraulic performance factor (η) for helical coiled inserts as a function of Reynolds number for pitch ratio (p/d) varied in the range of 0.5 to 1.5 and wire diameter ratio (e/d) varied in the range of 0.063 to 0.125. The plot shows that the Thermo-hydraulic performance factor escalated to a value of around 1.3 when an insert having pitch ratio (p/d) of 1.5 and wire diameter ratio (e/d) of 0.083 is used. The maximum value of thermo-hydraulic performance factor is yielded by the combination of (p/d) of 1.5 and (e/d) of 0.083, while the minimum value corresponds to (p/d) of 0.5 and (e/d) of 0.125.

The influence of Reynolds number on thermo-hydraulic performance factor for modified helical coiled insert with gradually increasing pitch can be seen in Fig. 12a. The thermo-hydraulic performance factor decreases with an increase in Reynolds number for (e/d) of 0.063 and 0.083 while it marginally increases for the rest of (e/d) values. The thermo-hydraulic performance factor remains more than unity for (e/d) of 0.063 and 0.083 while the Reynolds number value is kept within 4000. A graph of thermo-hydraulic performance factor as a function of Reynolds number for modified helical coiled insert with gradually decreasing pitch is shown in Fig. 12b. The maximum thermo-hydraulic performance of 1.4 corresponds to the wire diameter ratio (e/d) of 0.063. It can be observed that the increase in wire diameter ratio (e/d) leads to decrement in the thermo-hydraulic performance factor irrespective of the values of Reynolds number. It is interesting to note that the insert having wire diameter ratio (e/d) of 0.063 is the only insert geometry which yields thermo-hydraulic performance factor of greater than unity for the entire range of Reynolds number.

5 Empirical correlations of Nusselt number and friction factor

The empirical correlations of Nusselt number and friction factor as function of flow and geometrical parameters of inserts are developed by using regression analysis. These correlations are helpful for the investigators and designers to predict the heat transfer and frictional loss in heat exchanger with helical coiled inserts. The functional relationship of Nusselt number and friction factor with the system and operating parameters can be written as:

$$Nu = A\,[(p/d), (e/d), Re] \tag{14}$$

$$f = B\,[(p/d), (e/d), Re] \tag{15}$$

The predicted and experimental values of Nusselt number are plotted in Fig. 13 for the entire range of parameters of helical coiled inserts. The deviation of predicted values of

Nusselt number from those obtained by the experimentation is lying within ±15% of range.

The Nusselt number correlation is obtained as:

$$Nu = 0.0177 \quad Re^{0.946} \quad (p/d)^{-0.016} \quad (e/d)^{-0.14} \tag{16}$$

A plot of experimental and observed values of friction factor for helical coiled inserts is shown in Fig. 14. The deviation of predicted values of friction factor from those obtained during experimentation is lying within range of ±15%.

The correlation of friction factor is obtained as:

$$f = 78.316\, Re^{-0.448} \ (p/d)^{-0.664} \ (e/d)^{0.53} \tag{17}$$

6 Conclusions

The experimental results with regard to heat transfer and friction in a circular tube fitted with helical coiled and modified helical coiled inserts have been reported for water as a working fluid. The effect of different helical coiled configurations on the heat transfer and frictional losses have been examined for the pitch ratio (p/d) from 0.5 to 1.5 and wire diameter ratio (e/d) from 0.063 to 0.125 by varying the Reynolds number in the range of 1400 to 11,000. The experimental results have been presented in the form of Nusselt number and the friction factor with regard to the change in fluid flow and geometrical parameters of inserts in order to understand their effects on heat transfer and friction factor characteristics. The enhancements in Nusselt number and friction factor brought out by the application of inserts are discussed in detail. Moreover, the enhancements in heat transfer and friction are simultaneously evaluated by examining the thermo-hydraulic performance factor.

The following conclusions can be drawn from the present work:

- The Nusselt number of the helical coiled and modified helical coiled inserts increases with the increase in Reynolds number for all geometrical sets of parameters. The maximum enhancement in Nusselt number, relative to smooth tube, is found to be 2.62 corresponding to the pitch ratio (p/d) of 0.5 and wire diameter ratio (e/d) of 0.104. The Nusselt number enhancement ratio (Nu/Nu_s) lies in the range of 1.42–2.62 for tube fitted with helical coiled inserts. The enhancements in Nusselt number for modified helical coil inserts with gradually increasing pitch (GIP) and gradually decreasing pitch (GDP) are found to be in the range of 2–2.72 and 1.49–3.14, respectively. It is observed that the modified helical coiled inserts with gradually decreasing pitch performed better than other types of inserts considered in this study.
- The friction factor decreases as Reynolds number approaches to higher values in all cases. The increase in the values of pitch ratio (p/d) and wire diameter ratio (e/d) leads to the decrement in the values of friction factor irrespective of the Reynolds number. The friction factor enhancement ratio lie in the range of 3.4–27.4, 11.2–19.9, and 11.6–14.9 for the heat exchanger tube fitted with helical coiled inserts, modified helical coiled inserts with gradually increasing pitch, and modified helical coiled inserts with gradually decreasing pitch, respectively.
- The thermo-hydraulic performance factor of the heat exchanger tube fitted with helical coiled inserts, modified helical coiled inserts with gradually increasing pitch, and modified helical coiled inserts with gradually decreasing pitch is found in the range of 0.59–1.29, 0.82–1.17, and 0.6–1.39, respectively. It is evident from the results that the performance of modified helical coiled inserts with gradually decreasing pitch surpasses all the geometries of inserts considered in this study.

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