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The experimental investigation of heat transfer and pressure drop in a tube with coiled wire inserts placed separately from the tube wall

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ABSTRACT

The paper presents the experimental investigation of heat transfer and pressure drop in a tube with coiled wire inserts placed separately from the tube wall in turbulent flow regime. The experiments were performed with a constant wire thickness of $a = 6$ mm, three different pitch ratios ($P/D = 1$, $P/D = 2$ and $P/D = 3$) and two different distances ($s = 1$ mm, $s = 2$ mm) at which the coiled wire inserts were placed separately from the tube wall. Uniform heat flux was applied to the external surface of the tube and Reynolds numbers varied from 4105 to 26 400 in the experiments. The experimental results obtained from a smooth tube were compared with those from the studies in literature for validation of experimental set up. The use of coiled wire inserts leads to a considerable increase in heat transfer and pressure drop over the smooth tube. The Nusselt number and friction factor increase with decreasing pitch ratio (P/D) and distance (s) for coiled wire inserts. The highest overall enhancement efficiency of 50% was achieved for the coiled wire with $P/D = 1$ and $s = 1$ mm at Reynolds number of 4220. As a result, the experimental results reveal that using these coiled wire inserts are thermodynamically advantageous at all Reynolds numbers.

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1. Introduction

Many efforts have been made on heat transfer enhancement according to the progress of thermal systems. The recent researches in heat transfer enhancement lead to the development of currently used heat transfer techniques. These techniques can be classified into active and passive techniques. The active techniques require additional external power such as fluid injection and suction, fluid vibration and the usage of electrostatic fields. The passive techniques include extended surfaces, rough surfaces and swirl flow devices, etc. and no additional external power is needed. The coiled wire insert is a swirling flow device that is extensively used in various heat transfer applications such as, air conditioning and refrigeration systems, heat recovery processes, food and dairy processes, chemical process plants. The wire coils have some advantages over the other enhancement techniques such as simple manufacturing, low cost, easy to insert and remove from the tube.

The coiled wire inserted into a flow provide swirling flow and improved fluid mixing between the tube core and the near wall region, thus the heat transfer enhances by rapid fluid mixing. However, the swirl stimulated heat transfer enhancement causes an unavoidable shear stress and pressure drop in coiled wire inserted tube. To date, numerous investigations are performed in order to determine the effect of coiled wire inserts on heat transfer and pressure drop [1–7]. Promvonge [8], experimentally investigated the heat transfer enhancement in a circular tube with twisted tape and wire coil inserts. He reported that using the wire coils together with twisted tapes provides more heat transfer over the use of wire coil/twisted tape alone. The influence of several coiled wire geometries on pressure drop during condensation of R-134a vapor inside a horizontal tube was experimentally studied by Akhavan-Behabadi et al. [9]. Consequently, a new correlation based on the experimental data for predicting the pressure loss in coiled wire inserted tubes was developed and in another experimental study, Akhavan-Behabadi et al. [10] presented the results of the increasing heat transfer enhancement and pressure drop during flow boiling of R-134a in a coiled wire inserted horizontal evaporator while the heat transfer enhancement by using coiled wire inserts during forced convection condensation of R-22 inside a horizontal tube was experimentally investigated by Agrawal et al. [11]. Prasad and Shen [12] studied the heat transfer enhancement

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by using different coiled wire inserts based on exergy analysis. The effect of coil pitch and other related parameters on heat transfer enhancement and pressure loss in horizontal concentric tubes with coiled wire inserts are introduced by Naphon [13] and the results indicated that the coiled wire inserts are especially effective on laminar flow regime in the meaning of heat transfer enhancement. An experimental study was conducted related with the investigation of thermal performance of a tube with square cross-sectioned coiled wire by Promvonge [14]. The compared experimental results with the results obtained from circular cross-sectioned wire reveal that the square coiled wire insert yields better performance over the circular one under the same conditions. The effect of snail entry on thermal enhancement of a tube fitted with circular and square cross-sectioned coiled wire was also studied by Promvonge [15]. It is seen that the coiled wire inserts with snail entry provide more heat transfer and pressure drop than the inserts without snail entry.

In the above literature survey, most researchers are focused on investigating the effect of coil pitch, wire thickness or wire geometry on heat transfer and pressure drop in a tube with coiled wire inserts. These coiled wire inserts are attached on the tube wall for enhancing the heat transfer by increasing the effective heat transfer area and disturbing the laminar sublayer. The present experimental study differs from those available in the literature, that the coiled wire inserts are placed inside the tube separately from the tube wall in order to improve the heat transfer by only disturbing the laminar sublayer. The experiments were conducted with constant wire thickness of $a = 6$ mm in the range of Reynolds number from 4105 to 26 420. Three different pitch ratios ($P/D = 1$, $P/D = 2$ and $P/D = 3$) and two different distances ($s = 1$ mm and $s = 2$ mm) from the tube wall were considered. The effect of pitch ratio (P/D) and distance (s) from the tube wall on Nusselt number and friction factor were investigated and finally the overall enhancement results were also presented for all coiled wire inserts.

2. Experimental set up

The schematic view of experimental set up used in the experiments is shown in Fig. 1. The set up consists of a blower, a nozzle for air entering the tube uniformly, a venturimeter to measure the flow rate, a calming tube (6000 mm) and the heat transfer test tube (3100 mm) with a coiled wire insert. The SS304 seamless steel test tube has 56 mm inner diameter (D), 60 mm outer diameter (D_o), and 2 mm thickness (t). The equilateral triangular cross-sectioned coiled wire used in experiments was manufactured from aluminum by extrusion method. Three different pitch ratios ($P/D = 1$, $P/D = 2$ and $P/D = 3$) and two different distances ($s = 1$ mm and $s = 2$ mm) from the tube wall are considered in the experimental study. The test tube with coiled wire insert and the details are indicated in Fig. 2. In the experiments, the Teflon rings are used for placing the coiled wire separated from the tube wall. These rings were densely attached onto the inserts, thus the contact of inserts with tube inner wall was prevented. The coiled wire inserts with Teflon rings used in the experimental study are shown in Fig. 3.

The test tube was heated by electrical cable in order to provide uniform heat flux along the entire length of the test section. A variac transformer was used to control the electrical output. The outer surface of the test tube was well insulated with glass wool to reduce the convective heat loss to the surroundings. The surface temperatures of the tube wall were measured by 28 K-type thermocouples, which were placed on the local wall of the tube and calibrated within ±0.2 °C deviation by thermostat before being used. The inner and outer temperatures of the bulk air were also measured by K-type thermocouples at certain points.

The inlet bulk air is firstly vacuumed through the calming tube and later the air is directed through the test tube by 3 kW blower. After passing the test tube, the air enters the venturimeter, which is used for flow rate measurements. The venturimeter was calibrated before the experiments and the calibration procedure was given in detail in Ref. [16]. The volumetric air flow rates from the blower were adjusted by varying motor speed through an inverter. The pressure drop across the test tube is measured by using a differential pressure transmitter. The experiments were carried out by varying the flow rate and electrical power. The test tube was heated during the experiments and data of temperature, volumetric flow rate and pressure drop of the bulk air and electrical output were recorded after the system was allowed to approach the steady state condition. The data except for the volumetric flow rate were recorded by a multi-channel data logger with high resolution. The Reynolds number of the bulk air varied from 4105 to 26 400. The Nusselt number, Reynolds number and the various characteristics of the flow were calculated based on the average tube wall temperatures and the inlet and outlet air temperatures by a computer programme written in FORTRAN language. The local wall temperature, inlet and outlet air temperatures, pressure drop across the test section and volumetric flow rate were measured in order to investigate the heat transfer of the heated test tube.

The uncertainties of experimental measurements were determined by using the method introduced by Kline and McClintock [17]. The uncertainty calculation method used includes calculating derivatives of the desired variable with respect to the individual experimental quantities and applying known uncertainties. Consequently, the maximum uncertainties of the non-dimensional parameters were found ±3.2% for Reynolds number, ±8.6% for Nusselt number and ±6.5% for friction factor.

3. Data deduction

The heat flux applied to the test tube cause an increase in the outer surface temperature (T_{ow}) of the test tube in axial direction. Therefore, the heat loss is calculated for each part of the test tube in which the thermocouples exist. The total heat loss is taken as the sum of these 28 parts. The heat loss Q_{loss} is the heat transfers from the outer tube wall to the surroundings and calculated as follows:

$$Q_{loss} = F(T_{ow} - T_{\infty}) \tag{1}$$

$$\frac{1}{F} = \frac{1}{h_o A_{ins}} + \frac{1}{k_{ins} A_{12}} \tag{2}$$

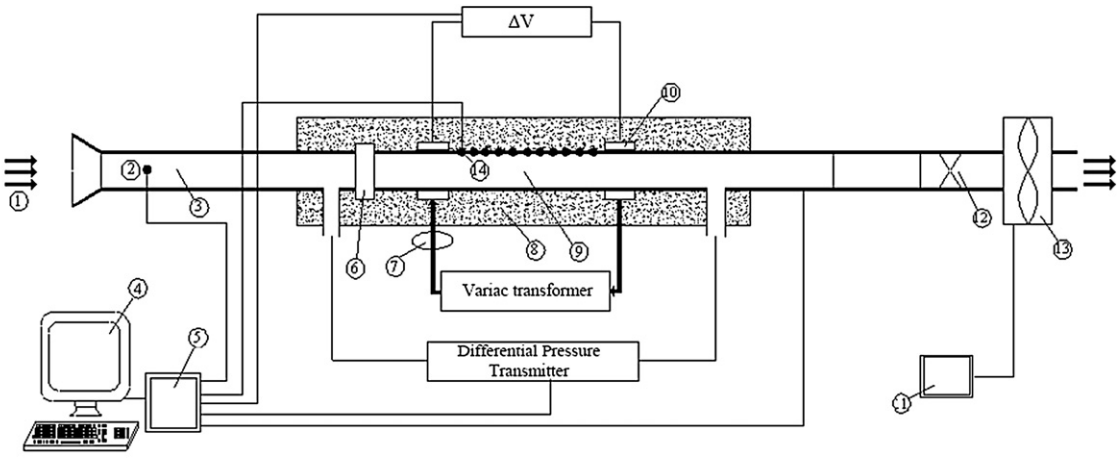
In Eq. (2), h_o indicates the heat transfer coefficient of the natural convection occurs between the outer surface of the insulated test tube and the surroundings and determined by iterative computation via FORTRAN computer code. A_{ins} and k_{ins} represent the surface area and conductive heat transfer coefficient of the insulated test tube, respectively. A_{12} is the logarithmic mean average of the insulated test tube surface area and the test tube outer surface area.

The steady state heat transfer rate and heat flux applied to the test tube can be written as:

$$Q_{air} = Q_{conv} \tag{3}$$

$$Q_{air} = \dot{m} C_{p,air}(T_0 - T_i) = \Delta V I - Q_{loss} \tag{4}$$

The heat provided by the electrical cable in the test tube is about 3–4% higher than the heat absorbed by the air for the thermal equilibrium test because of the convection and radiation heat losses (Q_{loss}) from the test section to the surroundings. Therefore, only the heat transfer rate absorbed by the air is taken into consideration for the convective heat transfer coefficient calculation.



1. Inflow
2. Inlet temperature measurement point
3. Calming tube
4. PC
5. Data logger
6. Coupling unit
7. Current transformer
8. Insulation
9. Test tube
10. Current valve
11. Inverter
12. Venturimeter
13. Blower
14. Thermocouples

Fig. 1. Schematic view of experimental set up.

$$q = \frac{Q_{air}}{\pi D_o L} \tag{5}$$

The local convective heat transfer coefficient through the heated test tube in any axial x direction is defined as

$$h(x) = \frac{q}{T_{iw}(x) - T_b(x)} \tag{6}$$

Here, $T_{iw}(x)$ and $T_b(x)$represent the local inner wall temperature of the heated test tube and local bulk temperature of the fluid, respectively. The determination of these temperatures is given in detail in Ref. [18]. The local convective heat transfer coefficients were calculated iteratively for each part of the test tube in which the thermocouples exist. All of the thermophysical properties of air were determined at the overall bulk mean temperature.

Consequently, the local Nusselt number can be calculated from:

$$Nu(x) = \frac{h(x)D}{k} \tag{7}$$

where k is the conductive heat transfer coefficient of fluid.

There is almost no difference between the local and mean Nusselt numbers for fully developed flow. Therefore, the average of last three local Nusselt numbers at the end of the test tube were taken into consideration for presenting the heat transfer results in Section 4.

The Reynolds number is defined by

$$\mathrm{Re} = \frac{UD}{\nu} \tag{8}$$

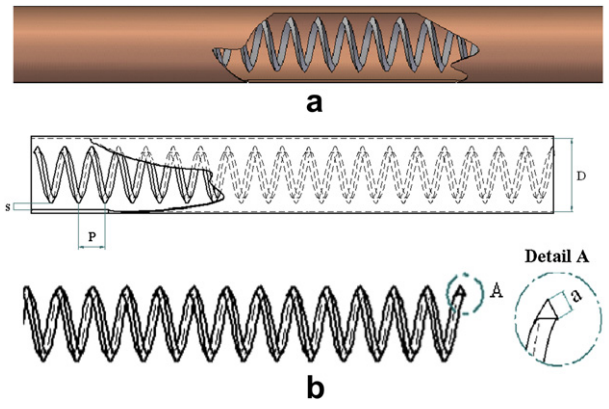


Fig. 2. (a) The coiled wire inserts placed separately from the tube wall, (b) the details of coiled wire.

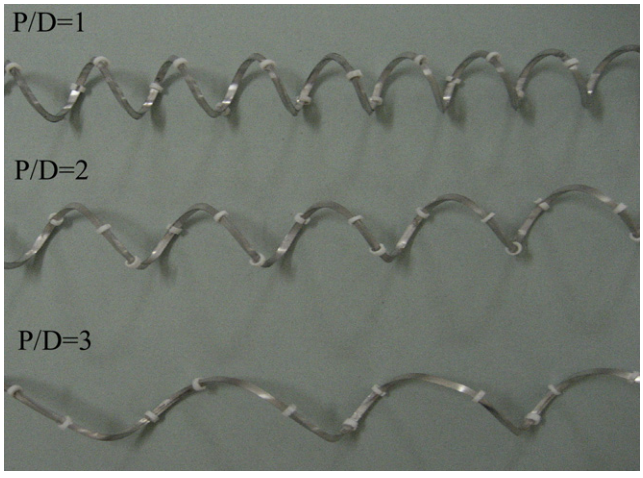


Fig. 3. The coiled wire inserts with Teflon rings (s = 2 mm).

The friction factor, f is calculated as follows:

$$f = \frac{\Delta P}{\frac{1}{2}\rho \cdot U^2 \frac{L}{D}} \tag{9}$$

where U indicates the mean fluid velocity in the tube.

According to constant pumping evaluation criteria [19]:

$$(\dot{V}\Delta P)_s = (\dot{V}\Delta P)_c \tag{10}$$

and the relationship between the friction factor and Reynolds number can be given as below:

$$\left(f\mathrm{Re}^3\right)_s = \left(f\mathrm{Re}^3\right)_c \tag{11}$$

$$\mathrm{Re}_s = \mathrm{Re}_c (f_c/f_s)^{1/3} \tag{12}$$

The overall enhancement efficiency, η is expressed as the ratio of the, h_c of an enhanced tube with coiled wire insert to that of a smooth tube, h_s, at a constant pumping power is introduced by Webb [19]:

$$\eta = \frac{h_c}{h_s}|_{pp} = \frac{Nu_c}{Nu_s}|_{pp} = \left(\frac{Nu_s}{Nu_s}\right)\left(\frac{f_s}{f_c}\right)^{1/3} \tag{13}$$

4. Results and discussion

The present heat transfer and friction factor results for a smooth tube were first validated in terms of Nusselt number and friction factor before the experiments with coiled wire inserts. The obtained experimental results of Nusselt number and friction factor for smooth tube were compared with the results obtained from the well-known steady state flow correlations of Gnielinski [20] and Petukhov [21], for the fully developed turbulent flow in circular tubes.

Gnielinski [20] correlation as given in Eq. (14) is used to find out heat transfer in smooth tube:

$$Nu = \frac{(f/8)(\mathrm{Re_D} - 1000)\mathrm{Pr}}{1 + 12.7(f/8)^{1/2}\left(\mathrm{Pr}^{2/3} - 1\right)} \quad 3000 \le \mathrm{Re} \le 5 \times 10^6 \tag{14}$$

Correlation of Petukhov [21] is given as follows:

$$f = (0.790\, In\, \mathrm{Re} - 1.64)^{-2} \quad 3000 \le \mathrm{Re} \le 5 \times 10^6 \tag{15}$$

The comparison between the results of the present smooth tube and the correlations of Eqs. (14) and (15) are shown in Figs. 4 and 5, respectively. As seen in these figures, there is a good agreement between the results for the present smooth tube and the correlations available in the literature. These results reveal the accuracy of the experimental set up and used measurement technique.

The correlations obtained from present smooth tube results for Nusselt number and friction factor are given as follows:

$$Nu = 0.0213\,\mathrm{Re}^{0.807}\mathrm{Pr}^{0.41} \tag{16}$$

$$f = 0.413344\,\mathrm{Re}^{-0.274713} \tag{17}$$

The present experimental study deals with the investigation of the heat and flow friction characteristics of an equilateral triangular cross-sectioned coiled wire inserted tube under uniform heat flux boundary condition. The experiments were performed

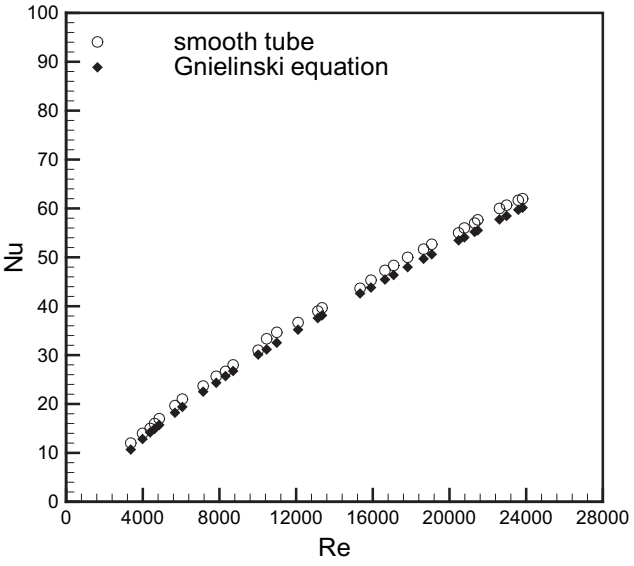


Fig. 4. Validation of smooth tube tests for Nusselt number.

with coiled wire having 6 mm thickness with three different pitch ratios ($P/D = 1$, $P/D = 2$ and $P/D = 3$) and two different distances ($s = 1$ mm and $s = 2$ mm) from the tube wall. Figs. 6 and 7 show the effect of pitch ratio (P/D) and the distance (s) at which the coiled wire inserts are placed separately from the tube wall on heat transfer and friction factor under turbulent flow conditions, respectively. It is obvious from these figures that the tube with coiled wire introduces both higher heat transfer and pressure drop than the smooth tube.

The variation of Nusselt number with Reynolds number for coiled wire inserts with three different pitch ratios ($P/D = 1$, $P/D = 2$ and $P/D = 3$) and two different distances ($s = 1$ mm and $s = 2$ mm) is presented in Fig. 6. It is evident from this figure heat transfer increases significantly with the reduction of pitch ratio, this is because the turbulence intensity and flow path become greater and longer for small pitch ratios. Heat transfer also increases with decreasing distance at which the coiled wire inserts are placed

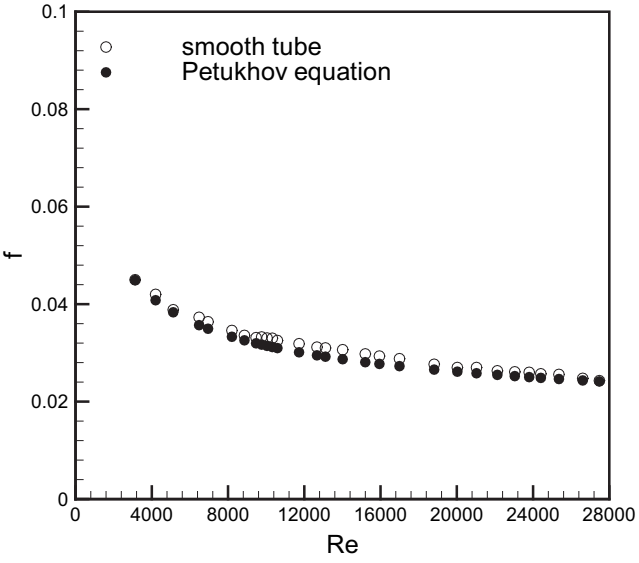


Fig. 5. Validation of smooth tube tests for friction factor.

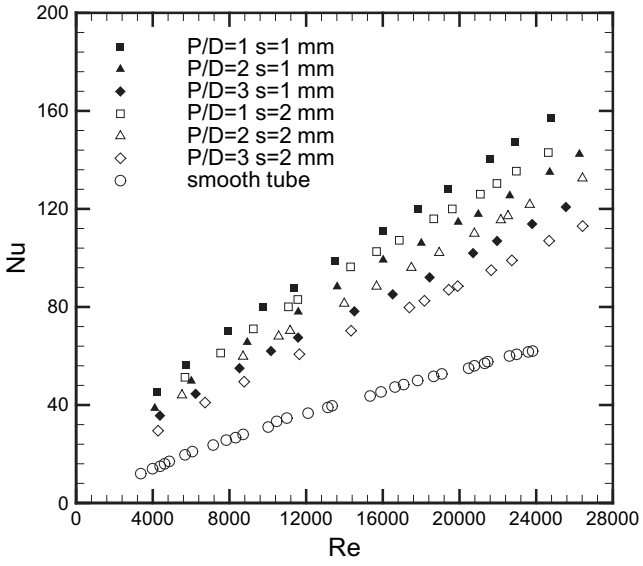


Fig. 6. Variation of Nusselt number with Reynolds number for different coiled wire inserts.

separately from the tube wall. The clearance between the wall and the coiled wire inserts causes higher velocities occur in the clearance gap, and therefore heat transfer increases. The coiled wire inserts, which were placed at a distance of 1 mm from the tube wall, provide higher heat transfer than the ones at a distance of 2 mm from the tube wall. Because the coiled wire inserts which were close to the near wall region, interrupts the development of the boundary layer of the fluid flow and increases the turbulence intensity in the flow field better than the ones far from the tube wall. As a result, the highest and lowest heat transfer are achieved for the coiled wire inserts with ($P/D = 1$, $s = 1$ mm) and ($P/D = 3$, $s = 2$ mm), respectively. The increase in Nusselt number for the cases ($P/D = 1$, $s = 1$ mm) and ($P/D = 3$, $s = 2$ mm) is in the range 140–200% and 64–94% over that of the smooth tube depending on Reynolds numbers, respectively.

The variation of friction factor with Reynolds number for all investigated coiled wire inserts is also presented in Fig. 7. The friction factor tends to decrease with increasing Reynolds number. The coiled wire inserts cause a remarkable fluid friction as well, in comparison with the smooth tube because of the dissipation of the dynamic pressure of the fluid due to higher surface area and flow blockage of the coiled wire through the flow field. The fluid friction increases with the decrease in pitch ratio (P/D) and distance (s). The number of coils increases by reducing the P/D ratio. Therefore, the highest pressure drop was observed for the coiled wire insert with ($P/D = 1$, $s = 1$ mm) due to the fact that increasing the number of coils disturbs the entire flow field and causes more friction, while the lowest pressure drop was obtained for the case of ($P/D = 3$, $s = 2$ mm). Flow passes through the clearance between the wall and coiled wire inserts, and fluid friction results with an increase because of the flow disturbance near the wall region due to the coiled wire inserts. The friction factor for the coiled wire insert with ($P/D = 1$, $s = 1$ mm) and ($P/D = 3$, $s = 2$ mm) is averagely 7.4 and 4.45 times more than the smooth tube, respectively.

Consequently, the experimental results of Nusselt number and friction factor for coiled wire inserts are correlated as given in below:

$$Nu = 0.077156\,\mathrm{Re}^{0.716692}(P/D)^{-0.253417}(s/D)^{-0.124382}\mathrm{Pr}^{0.4} \quad (18)$$

$$f = 3.970492\,\mathrm{Re}^{-0.367485}(P/D)^{-0.31182}(s/D)^{-0.157719} \quad (19)$$

Fig. 8 illustrates the heat transfer ratio (Nu_c/Nu_s) versus Reynolds number for all investigated cases. The heat transfer ratio must be greater than unity for a net energy gain. The heat transfer increase provided by coiled wire inserts is remarkable and tends to decrease with the increase of Reynolds number, as indicated in Fig. 8. The reverse flow provided by coiled wire inserts improves the convection of the tube wall by rising the mean velocity and temperature gradient and reducing the cross section of flow field. The highest heat transfer ratio of 3.0 is achieved by using the wire with $P/D = 1$, $s = 1$ mm, which interrupts the development of the boundary layer of the fluid flow and increases the turbulence intensity in the flow field better than the others. The coiled wire with $P/D = 1$ and $s = 1$ mm produces higher heat fluxes and momentum transfers not only depending on the large effective driving potential force but also on the higher pressure drop in comparison with the other cases.

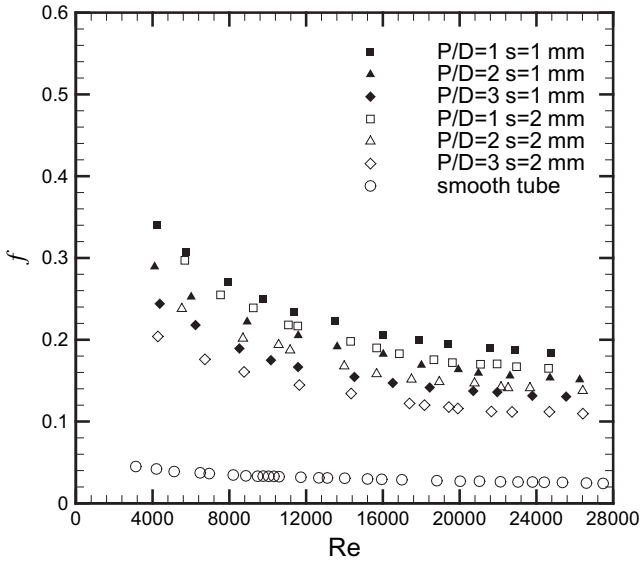


Fig. 7. Variation of friction factor with Reynolds number for different coiled wire inserts.

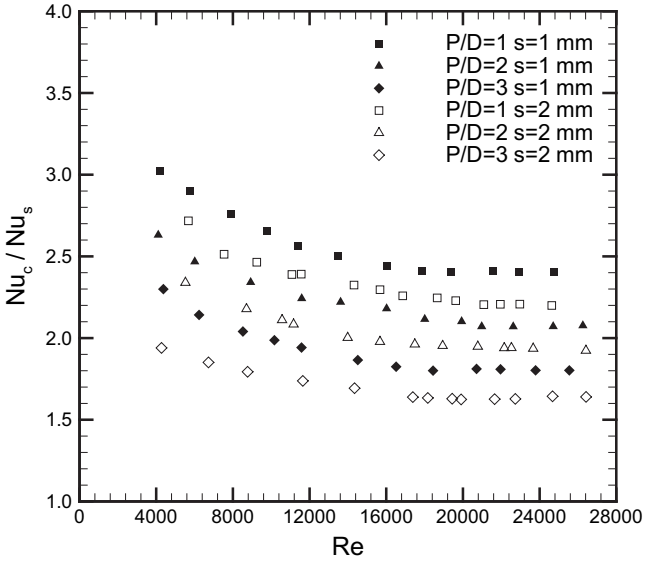


Fig. 8. Variation of heat transfer ratio (Nu_c/Nu_s) with Reynolds number.

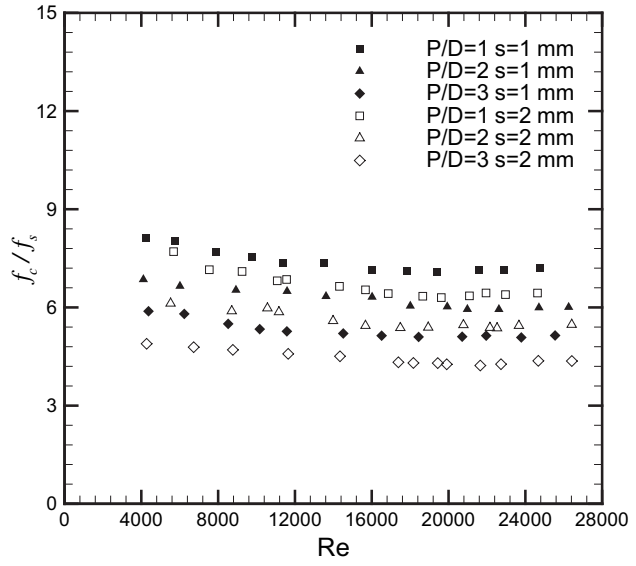


Fig. 9. Variation of friction factor ratio (f_c/f_s) with Reynolds number.

The variation of friction factor ratio (f_c/f_s) with Reynolds number is presented in Fig. 9. The friction factor ratio increases with the decrease of pitch ratio (P/D) and distance (s) similar with heat transfer rate. The coiled wire inserts which are close to the tube wall exhibit more flow resistance than the far ones. The highest friction factor ratio of 8.13 is observed for the coiled wire with ($P/D = 1$, $s = 1$ mm), in which there is higher surface area and flow blockage of the coiled wire through the flow field.

The variation of overall enhancement efficiency (η) is plotted against Reynolds number for all cases in Fig. 10. Heat transfer enhancement is provided with the expense of increasing pressure drop caused by coiled wire inserts. Therefore, a performance analysis is necessary for the evaluation of the net energy gain to make a decision if the technique applied to increase the heat transfer is efficient from energy point of view. The comparison is made based on the same pumping power in order to determine the net final gain.

There will be a net energy gain only if η is greater than unity. So, it is clear from this figure that using coiled wire inserts is advantageous over the smooth tube. As seen from Fig. 10 that using these inserts at low Reynolds numbers provide higher thermal efficiency. The overall enhancement efficiency tends to decrease with the increase in Reynolds number, however η does not change considerably after Reynolds number is about 16 000. Consequently, all of the inserts are thermodynamically advantageous ($\eta > 1$) at all Reynolds numbers because the heat transfer increase dominates the increase in friction loss. The overall enhancement efficiency increases with the decrease of pitch ratio (P/D) and distance (s) for all investigated cases. The highest overall enhancement efficiency of 1.50 was obtained for the coiled wire with $P/D = 1$ and $s = 1$ mm.

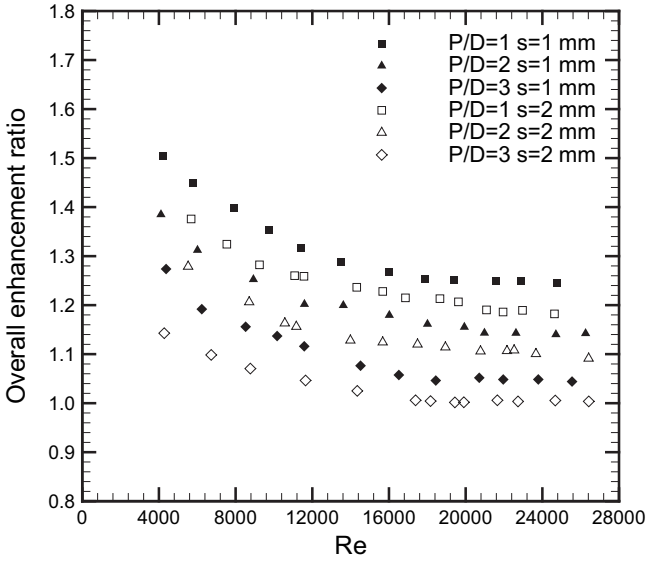


Fig. 10. Variation of overall enhancement efficiency (η) with Reynolds number.

5. Conclusions

It is known that attaching the inserts onto the tube wall may cause contamination over time, so there will occur an additional resistance to heat transfer. Therefore, placing the inserts separated from the wall will be convenient in terms of preventing the formation of contamination. The present experimental study was carried out in order to investigate heat transfer enhancement by means of coiled wire inserts placed separately from the tube wall. The secondary flows of the fluid cause swirling flow in coiled wire inserted tube, therefore the tube with coiled wire introduces both higher heat transfer and pressure drop than the smooth tube depending on pitch ratio (P/D) and distance (s). The experimental results show that the Nusselt number and friction factor increase with decreasing of pitch ratio (P/D) and distance (s). The highest overall enhancement efficiency of 50% was achieved for the coiled wire with $P/D = 1$ and $s = 1$ mm at Reynolds number of 4220. In conclusion, the experimental results reveal that using these coiled wire inserts separately from the tube wall is advantageous in order to provide higher thermal performance especially at low Reynolds numbers and the correlations for the Nusselt number and friction factor obtained from the present experimental data are given for practical uses.

Acknowledgements

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Nomenclature

a: wire thickness (m)
$C_{p,air}$: specific heat capacity of air (J/kg K)
D: inner diameter of the tube (m)
F: total heat transfer coefficient (W/m^2 K)
f: friction factor
h: convective heat transfer coefficient (W/m^2 K)
I: current (A)
k: thermal conductivity (W/m K)
L: length of the test tube (m)
$\dot{m}$: air mass flow rate (kg/s)
Nu: Nusselt number (hD/k)
ΔP: pressure drop (N/m^2)
P: pitch of coiled wire (m)
Pr: Prandtl number
Re: Reynolds number (UD/ν)
s: distance between coiled wire and test tube inner wall (m)
Q: heat transfer (W)
q: heat flux (W/m^2)
T: steady state temperature (K)
t: thickness of the test tube (m)
U: mean fluid velocity (m/s)
V: voltage (V)
$\dot{V}$: volumetric flow rate (m^3/s)

Greek letters

ρ: fluid density (kg/m^3)
η: overall enhancement efficiency
υ: kinematic viscosity (m^2/s)

Subscripts

b: bulk
c: coiled wire inserted tube
conv: convection
i: inlet
ins: insulated test tube
iw: inner wall of test tube
o: outlet
ow: outer wall of test tube
pp: pumping power
s: smooth tube
w: tube wall media
∞: ambient