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Heat transfer enhancement in a tube with equilateral triangle cross sectioned coiled wire inserts

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ABSTRACT

The heat transfer and pressure drop were experimentally investigated in a coiled wire inserted tube in turbulent flow regime. The coiled wire has equilateral triangular cross section and was inserted separately from the tube wall. The experiments were carried out with three different pitch ratios (P/D = 1, 2 and 3) and two different ratio of equilateral triangle length side to tube diameter (a/D = 0.0714 and 0.0892) at a distance (s) of 1 mm from the tube wall in the range of Reynolds number from 3500 to 27,000. Uniform heat flux was applied to the external surface of the tube and air was selected as fluid. 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 rises with the increase of Reynolds number and wire thickness and the decrease of pitch ratio. The highest overall enhancement efficiency of 36.5% is achieved for the wire with a/D = 0.0892 and P/D = 1 at Reynolds number of 3858. Consequently, the experimental results reveal that the best operating regime of all coiled wire inserts is detected at low Reynolds number, leading to more compact heat exchanger.

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1. Introduction

Several heat transfer enhancement techniques are utilized in order to improve the heat transfer or thermal performance of heat exchangers. Up to now, many attempts have been made to reduce the size and cost of heat exchangers. The goal of the heat transfer enhancement is to provide high heat fluxes or heat transfer coefficient. The second law efficiency increases and the entropy generation decreases by reducing the temperature driving force which is caused with the effect of heat transfer coefficient increase. Generally, heat transfer enhancement techniques can be divided into two groups (passive and active techniques). The passive techniques have been usually preferred by many researchers since no additional external power is required as extended surfaces, rough surfaces and swirl flow devices. Coiled wire insert is one of the passive heat transfer enhancement techniques, which 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 coil is quite simple to manufacture, to insert and remove from the tube which, therefore, justifies its usage in heat transfer enhancement.

Coiled wire, twisted tape or other swirl flow devices inserted into a flow provide swirling flow and periodic redevelopment of the boundary layer, increase the effective heat transfer area and the turbulence intensity. The swirl induced tangential flow velocity component causes improved fluid mixing between the tube core and the wall region nearby. Thus enhancing the heat transfer by rapid fluid mixing. On the other hand, the swirl induced heat transfer enhancement brings along inevitable shear stress and pressure loss in coiled wire or twisted tape inserted tube. For a long time, many researches have been carried out concerning the effect of wire coil inserts on the heat transfer and pressure drop [1–4]. Wang and Sunden [5] introduced a comparison of the thermal and hydraulic performances between twisted tape and coiled wire inserts for both laminar and turbulent flow regimes. They reported that coiled wire inserts provided better overall enhancement than the twisted tape inserts. The thermal augmentation in circular tube with twisted tape and wire coil inserts was experimentally investigated by Promvonge [6]. The results indicated that the existence of wire coils together with twisted tapes yielded more heat transfer over the use of wire coil/twisted tape alone. The effect of the wire coils having different pitches on the thermal stress and conjugate heat transfer under uniform heat flux was investigated by Ozceyhan [7]. It was found that the maximum thermal stress ratio occurred in the case of $p = 2d$ for 3 m/s mean water velocity. Prasad and Shen [8] evaluated the performance of several coiled wire inserts in heat

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Nomenclature

a	side length of equilateral triangle in the wire cross section
$C_{p,air}$	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 (Ampere)
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 from the tube 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

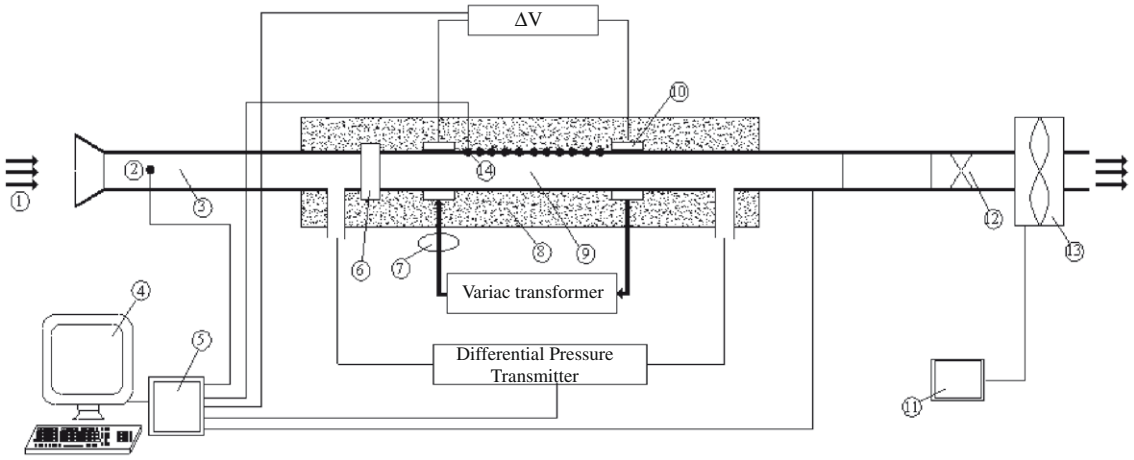
transfer enhancement, based on exergy analysis. The effect of different coiled wire geometry on pressure drop during condensation of R-134 a vapour inside a horizontal tube was experimentally investigated by Akhavan-Behabadi et al. [9]. Finally, they developed a new correlation based on the experimental data for predicting the pressure loss in coiled wire inserted tubes. 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 Agrawal et al. [11] experimentally studied the heat transfer enhancement by using coiled wire inserts during forced convection condensation of R-22 inside a horizontal tube. Naphon and Sriromruln [12] experimentally studied the heat transfer characteristics and the pressure drop of the micro-fin tubes with coiled wire insert. It was shown that the coil wire inserts increased both heat transfer and friction factor in the tube. An experimental study was conducted related with the investigation of the heat transfer and friction loss in a coiled wire inserted tube besides the properties of vortices generated by coiled wire inserts and the relation between entropy generation and vortex characteristics by Yakut and Sahin [13]. Naphon [14] investigated the effect of coil pitch and other corresponding parameters on heat transfer enhancement and pressure drop of the horizontal concentric tubes with coiled wire insert, and finally it is reported that the coiled wire inserts are particularly effective on laminar flow region in the meaning of heat transfer enhancement. Promvonge [15] studied the thermal performance of a tube with square cross sectioned coiled wire, and compared the experimental results with the results obtained from circular cross sectioned wire. The results reveal that the square coiled wire insert provides better overall enhancement than the circular one under the same conditions. In another study, Promvonge [16] additionally examined the effect of snail entry on thermal enhancement of a tube fitted with circular and square cross sectioned coiled wire.

The literature survey on investigations of coiled wire inserts shows that these inserts are generally attached onto the tube walls in order to improve the heat transfer by not only disturbing the laminar sublayer but also increasing the effective heat transfer area. In this experimental study, unlike the studies available in the literature, the coiled wire inserts with different pitch ratios (P/D) were placed inside the tube separated from the tube wall, thus the effective heat transfer area was not increased and heat transfer enhancement was achieved by only disturbing the laminar sublayer and discontinuing the development of boundary layer. Moreover, 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 [17].

The studies on coiled wire inserts available in literature are generally performed with circular or square cross sectioned coiled wire. Therefore, the equilateral triangular cross sectioned coiled wire was preferred to make the study different from those available in literature. The experiments were carried out with three different pitch ratios (P/D = 1, 2 and 3) and two different ratio of triangle length side to tube diameter (a/D = 0.0714 and 0.0892) at a distance (s) of 1 mm from the tube wall in the range of Reynolds number from 3500 to 27,000. Consequently, the effect of P/D and a/D on heat transfer and pressure drop were experimentally investigated and the overall enhancement results were also presented.

2. Experimental set-up

The open loop experimental facility used in the experiments is schematically shown in Fig. 1. The facility includes 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 coiled wire used in experiments was fabricated from aluminium by extrusion method in equilateral triangular cross section with side length of (a). Two different a/D ratios of 0.0714 and 0.0892, and three different pitch ratios (P/D = 1, 2 and 3) were considered in the experimental study. The test



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.

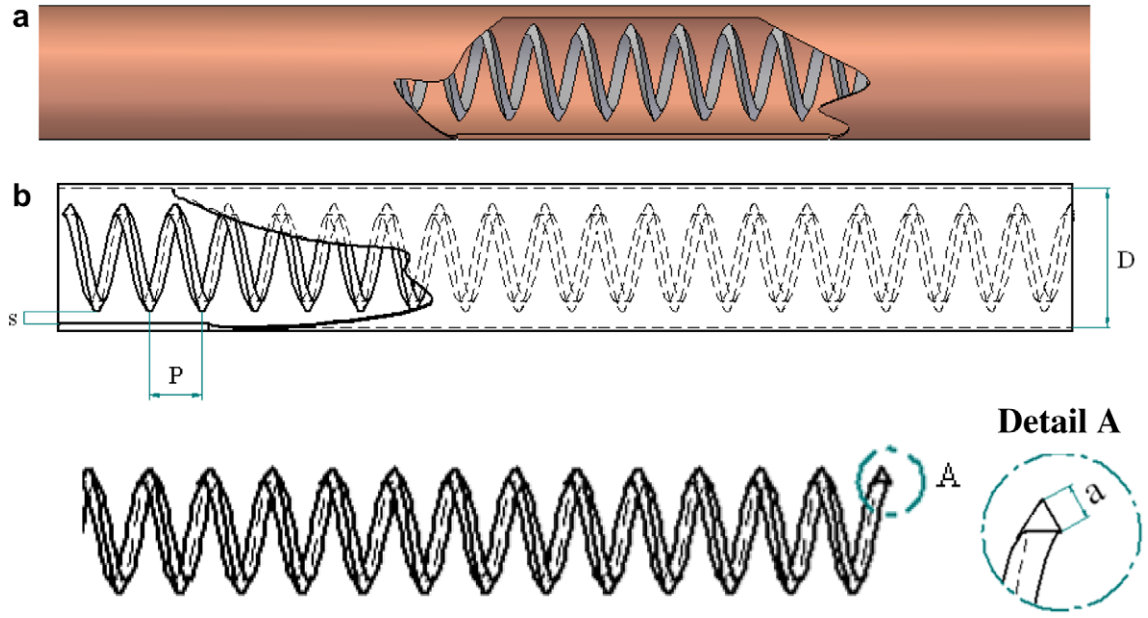


Fig. 2. (a) The equilateral triangle cross sectioned coiled wire inserted in a tube separated from tube wall and (b) the details of coiled wire.

tube with coiled wire insert and the details are given in Fig. 2. The Teflon rings were manufactured according to the wire thickness in order to fix 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.

Uniform heat flux was applied to the test tube by heating it with electrical cable. The electrical output power was controlled by a variac transformer to provide constant heat flux along the entire length of the test section. 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 heated air in the test tube was removed from the experimental field to the atmosphere by a piping system during the experiments. Therefore, the temperature of experimental field (T_∞) was not changed considerably, it is negligible.

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

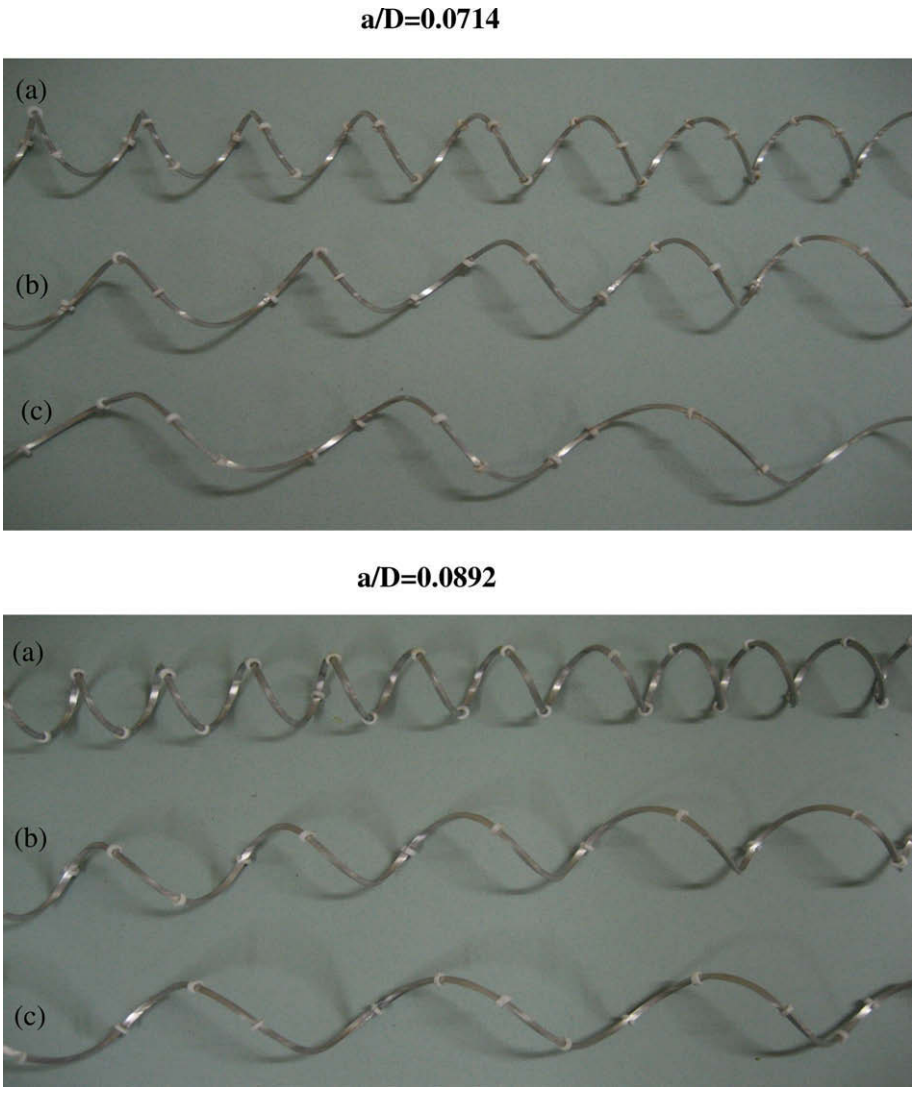


Fig. 3. The coiled wire inserts with Teflon rings (a) P/D = 1, (b) P/D = 2, and (c) P/D = 3.

given in detail in Ref. [18]. 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 3500 to 27,000. 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 [19]. 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 are can be written as;

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

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

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.

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

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. [20]. All of the thermophysical properties of air were determined at the overall bulk mean temperature.

Consequently, the local Nusselt Number can be calculated from;

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

where k is the conductive heat transfer coefficient of fluid.

The Reynolds number is defined by

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

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 [21];

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

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

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

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

The overall enhancement efficiency (η) is expressed as the ratio of the, hc of an enhanced tube with coiled wire insert to that of a smooth tube, hs at a constant pumping power is introduced by Webb [21]:

$$\eta = \frac{h_c}{h_s}\bigg|_{pp} = \frac{\mathrm{Nu}_c}{\mathrm{Nu}_s}\bigg|_{pp} = \left(\frac{\mathrm{Nu}_c}{\mathrm{Nu}_s}\right)\left(\frac{f_s}{f_c}\right)^{1/3} \tag{13}$$

4. Results and discussion

4.1. Validation of smooth tube tests

The Nusselt number and the friction factor are obtained for a smooth tube to validate the experimental procedure used before the experiments with coiled wire inserts. The results of Nusselt number and friction factor for smooth tube are compared with the results obtained from the well-known steady state flow correlations of Gnielinski [22] and Petukhov [23], for the fully developed turbulent flow in circular tubes.

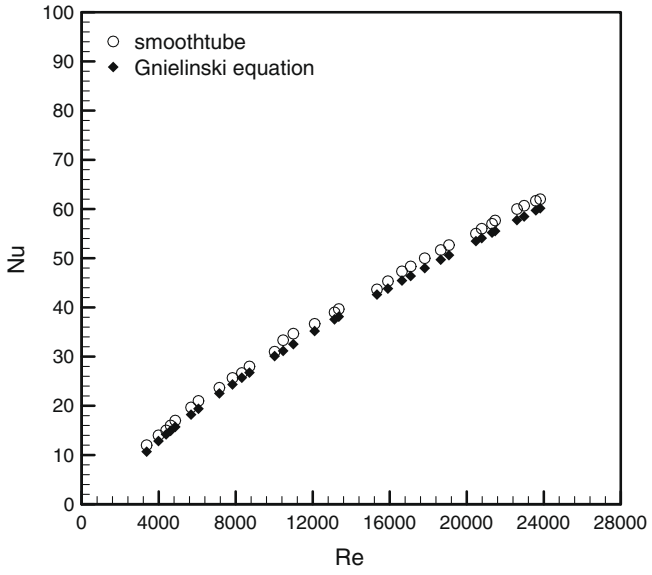


Fig. 4. Verification of Nusselt number for smooth tube.

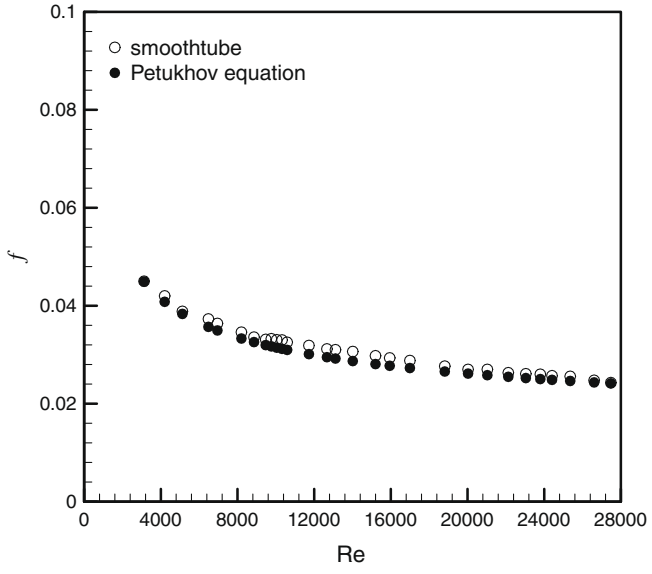


Fig. 5. Verification of friction factor for smooth tube.

Figs. 4 and 5 show the comparison between the results of the present smooth tube and the correlations of Gnielinski [22] and Petukhov [23], 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 indicate that the experimental set-up and measurement technique are accurate.

4.2. Effect of a/D

The present experimental study is related with the investigation of the heat and flow friction characteristics of a equilateral triangular cross sectioned coiled -wire inserted tube under uniform

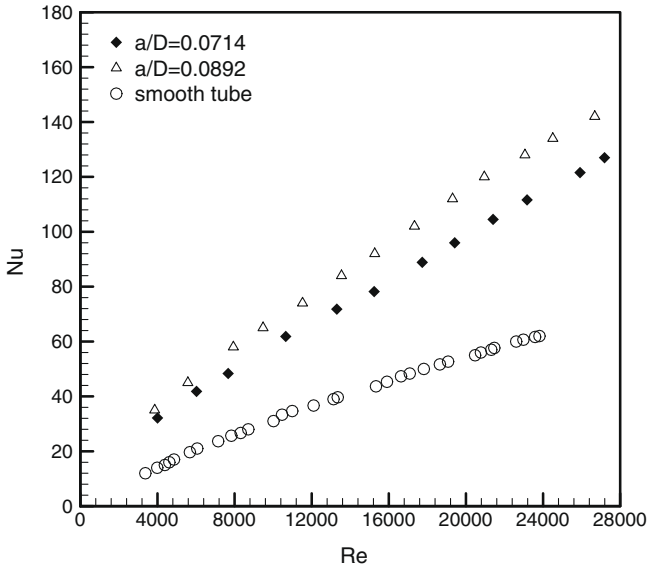


Fig. 6. Variation of Nusselt number with Reynolds number for different a/D ratios ($P/D = 1$).

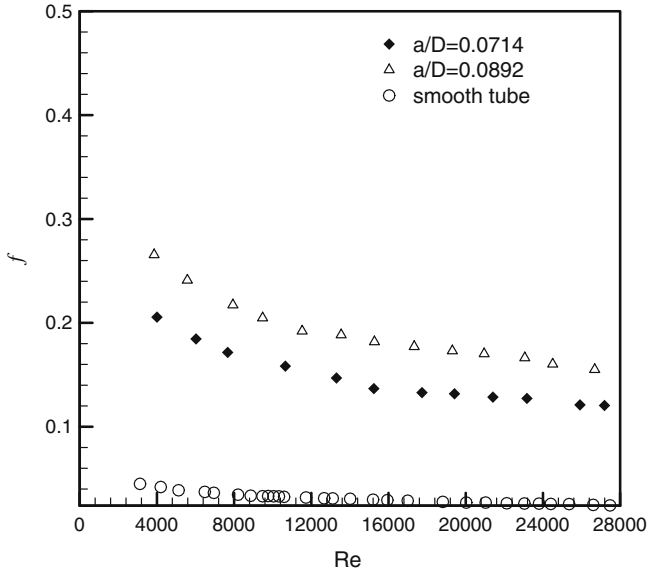


Fig. 7. Variation of friction factor with Reynolds number for different a/D ratios ($P/D = 1$).

heat flux boundary condition. The experiments were performed with coiled wire with two different ratio of equilateral triangle length side to tube diameter ($a/D = 0.0714$ and 0.0892). The effect of a/D on heat transfer and friction factor under turbulent flow conditions for a constant pitch ratio of $P/D = 1$ are presented in Figs. 6 and 7, respectively. It is clear from these figures that the tube with coiled wire introduces both higher heat transfer rate and friction factor than the smooth tube.

The Nusselt number increases with the increase of Reynolds number, as expected while the wire with $a/D = 0.0892$ provides higher heat transfer than the one with $a/D = 0.0714$ for overall Reynolds number range, as seen in Fig. 6. This is related with the thicker wire provides better fluid mixing in the flow field, causes an increase in turbulence intensity and as a result yields higher heat transfer rate. The increment in Nusselt number for the wire with $a/D = 0.0892$ is found to be approximately 11–17% over that of the one with $a/D = 0.0714$, depending on Reynolds number.

On the other hand, as expected, the coiled wire inserts cause a remarkable pressure drop as well, in comparison with the smooth tube because of the spent of the dynamic pressure of the fluid due to higher surface area and flow blockage of the coiled wire through the flow field. The friction factor decreases with increasing Reynolds number, as presented in Fig. 7. It is observed that the smaller wire ($a/D = 0.0714$) leads to a reduction in the pressure drop. The friction factor for the wires with $a/D = 0.0892$ and 0.0714 are averagely 6.3 and 4.8 times those of the smooth tube, respectively.

4.3. Effect of P/D

The variation of Nusselt number with Reynolds number for three pitch ratios ($P/D = 1$, 2 and 3) of the wire with $a/D = 0.0892$ is shown in Fig. 8. It is evident from this figure heat transfer increases significantly with the reduction of pitch ratio. The highest heat transfer is achieved for the lowest pith ratio of $P/D = 1$ due to the turbulence intensity and flow path of this pitch ratio is greater and longer than that from the others. The increase in Nusselt number is in the range 111–157%, 88–128% and 59–102% over that of the smooth tube depending on Reynolds number values for $P/D = 1$, 2 and 3, respectively.

The variation of friction factor with Reynolds number for three pitch ratios ($P/D = 1$, 2 and 3) of the wire with $a/D = 0.0892$ is also shown in Fig. 9. The friction factor decreases with increasing Reynolds number and pitch ratio. The low pitch ratio means increasing the number of coils, so the highest friction factor observed for the case of $P/D = 1$ due to the fact that increasing the number of coils disturb the entire flow field and cause more friction. For all pitch ratios, a considerable increase in friction factor is observed in comparison with the smooth tube. This is because the higher surface area and the reverse flow cause the dissipation of dynamic pressure of the fluid. The mean increases in friction factor of using the coiled wire inserts with pitch ratios, $P/D = 1$, 2 and 3 are about 6.3, 4.85 and 3.95 times that of the smooth tube, respectively.

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

$$\mathrm{Nu} = 0.598417\ \mathrm{Re}^{0.745064}\ (P/D)^{-0.268374}\ (a/D)^{0.813205}\ \mathrm{Pr}^{0.39} \tag{14}$$

$$f = 83.70924\ \mathrm{Re}^{-0.305268}\ (P/D)^{-0.388}\ (a/D)^{1.319018} \tag{15}$$

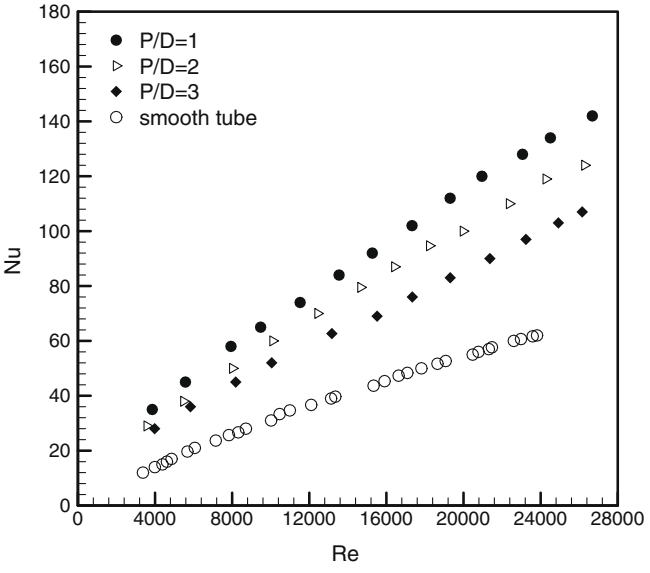


Fig. 8. Variation of Nusselt number with Reynolds number for different pitch ratios ($a/D = 0.0892$).

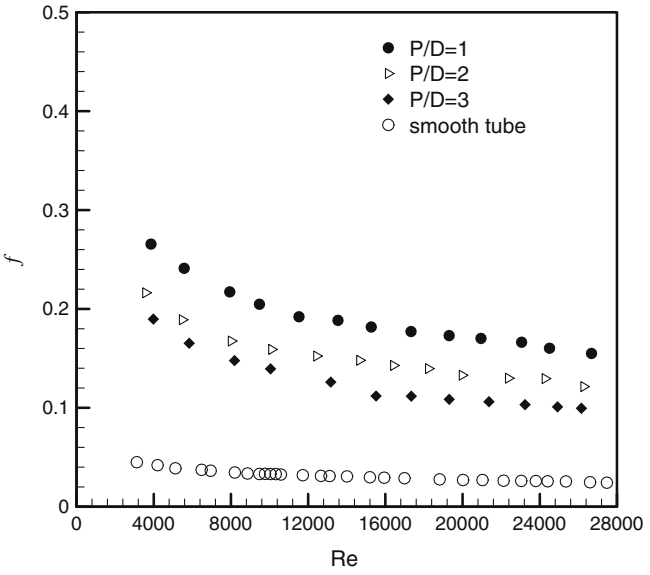


Fig. 9. Variation of friction factor with Reynolds number for different pitch ratios ($a/D = 0.0892$).

4.4. Performance evaluation

Heat transfer ratio (Nu_c/Nu_s), friction factor penalty (f_c/f_s) and overall enhancement efficiency $[(N_c/Nu_s)\cdot(f_s/f_c)^{1/3})]$ are used to evaluate effect of coiled wire inserts.

The heat transfer ratio (Nu_c/Nu_s) versus Reynolds number is illustrated in Fig. 10. It must be pointed out here that for a net energy gain, that is, for an effective heat transfer, the heat transfer ratio must be greater than unity. It is obvious from Fig. 10 that the heat transfer increase introduced by coiled wire inserts is remarkable. The Nusselt number ratio value tends to decrease with the increase of Reynolds number from 3500 to 15,000 for all cases, however the heat transfer ratio does not change significantly after the value of Reynolds number is 15,000. The heat transfer ratio rises with the reduction of pitch ratio and increment of wire thickness. However, the wire with $a/D = 0.0714$ and $P/D = 1$ provides higher heat transfer than wire with $a/D = 0.0892$ and $P/D = 3$ This is because the reverse flow provided by the wire with $a/D = 0.0714$ and $P/D = 1$ improves the convection of the tube wall by rising the mean velocity and temperature gradient and reducing the cross section of flow field. As a result, wire with $a/D = 0.0714$ and $P/D = 1$ produce higher heat fluxes and momentum transfers not only depending on the large effective driving potential force but also the higher pressure drop in comparison with the other wire inserts with $a/D = 0.0714$ and wire with $a/D = 0.0892$ and $P/D = 3$. The highest heat transfer ratio of 2.5 is achieved by using wire with $a/D = 0.0892$ and $P/D = 1$ which interrupts the development of the boundary layer of the fluid flow and rises the turbulence intensity in the flow field better than the others.

Fig. 11 presents the variation of friction factor ratio (f_c/f_s) with Reynolds number. The friction factor ratio rises with the reduction of pitch ratio and increase of wire thickness similar with heat transfer rate. The wire with $a/D = 0.0714$ and $P/D = 1$ causes higher pressure drop than wire with $a/D = 0.0892$ and $P/D = 3$ due to the fact that the 4 mm wire exhibits more flow resistance with $P/D = 1$. The highest friction factor ratio of 6.35 is observed for the wire with $a/D = 0.0892$ and $P/D = 1$, 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. 12. Heat transfer enhancement is provided with the expense of rising friction losses caused by coiled wire inserts. So, a performance analysis is essential 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 η is greater than unity. So, it is evident from this figure that using coiled wire inserts is advantageous over the smooth tube. The overall enhancement efficiency tends to decrease up to Reynolds number is about nearly 15,000 for all inserts and it is greater than 1 for the Reynolds number ranging from 3500 to 8000 for all cases. The overall enhancement ratios for wire with $a/D = 0.0714$ and $P/D = 1$ and wire with $a/D = 0.0892$ and $P/D = 3$ are quite close to each other The use of wire with $a/D = 0.0714$ and $P/D = 2$ and 3 after Reynolds number is 12,000 and 8000 respectively, and is not thermodynamically advantageous since the increase in friction factor is more effective than the increase

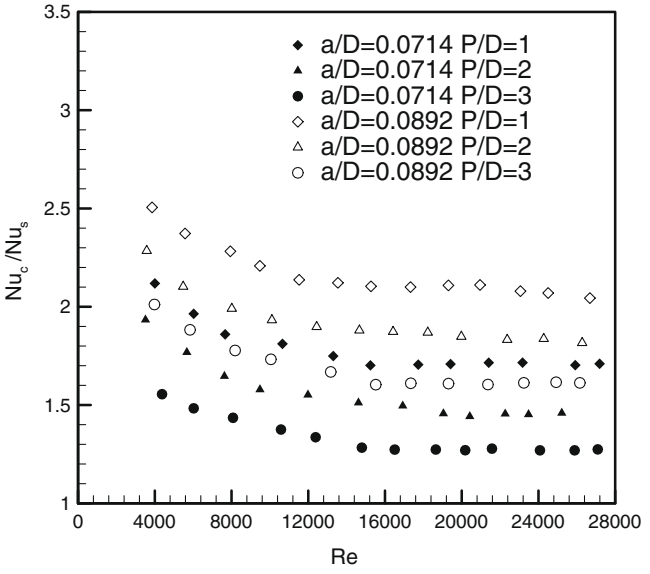


Fig. 10. Variation of heat transfer ratio (Nu_c/Nu_s) with Reynolds number.

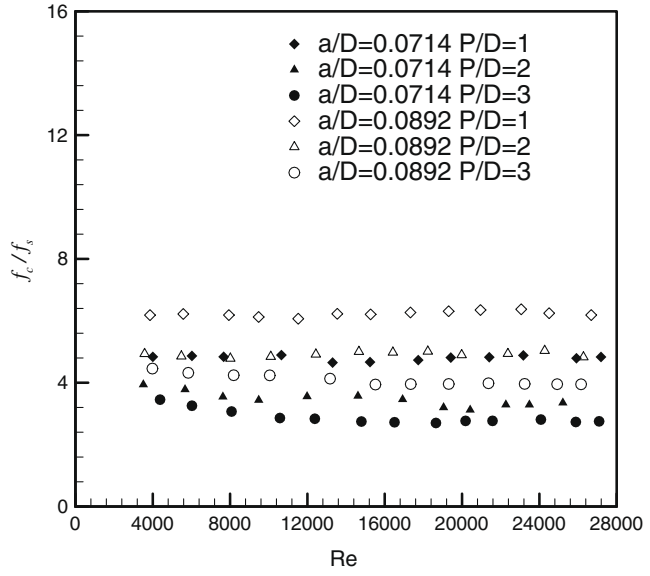


Fig. 11. Variation of friction factor ratio (f_c/f_s) with Reynolds number.

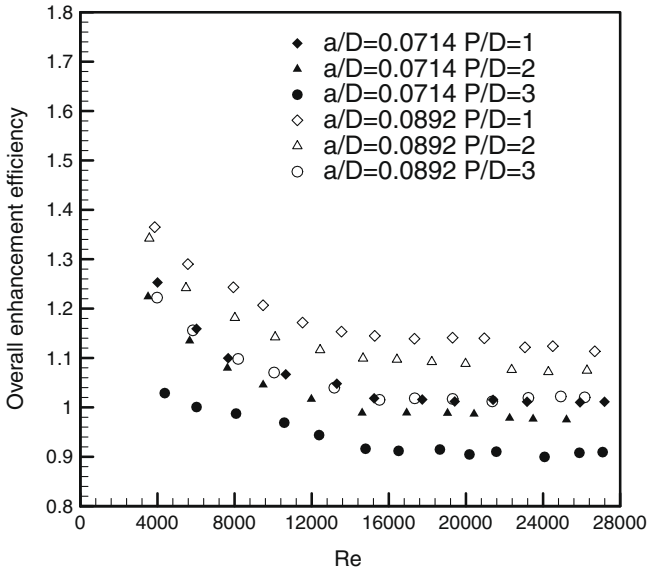


Fig. 12. Variation of overall enhancement efficiency (η) with Reynolds number.

in heat transfer while using the wire with $a/D = 0.0892$ at different pitch ratios is thermodynamically advantageous at all Reynolds numbers.

Promvonge [15] experimentally compared the thermal performance of square and circular cross sectioned coiled wire inserts which were attached onto the tube wall ($s = 0$ mm) in the range of Reynolds number 5000–25,000. The average enhancement efficiencies of the 3 mm square and circular ones with $P/D = 0.316$ are about 1.18 and 1.1, respectively. Consequently, in the present study, the average overall enhancement factors of the wire with $a/D = 0.0714$ at $P/D = 1$, 2 and 3 are about 1.06, 1.03 and 0.94 respectively, and the ones with $a/D = 0.0892$ at $P/D = 1$, 2 and 3 are about 1.187, 1.135 and 1.06, respectively for $s = 1$ mm.

5. Conclusions

The present experimental study is focused on the investigation of heat transfer and pressure drop in a coiled wire inserted tube. The equilateral triangle cross sectioned coiled wire inserted separately from the tube wall and the coiled wire inserts cause a remarkable increase in both pressure drop and heat transfer in comparison with the smooth tube depending on coil pitches and wire thickness. A considerable heat transfer enhancement, $Nu_c/Nu_s = 1.27–2.5$ is provided by using coiled wire inserts. In order to determine the net final gain, the coiled wire inserts are compared with the smooth tube at a constant pumping power, an increase in overall enhancement efficiency is obtained especially at low Reynolds number. The results reveal that using thin wire with large coil pitch is not thermodynamically advantageous especially at high Reynolds numbers as in the case of a wire with $a/D = 0.0714$ and $P/D = 3$. The highest overall enhancement efficiency of 36.5% is achieved for the wire with $a/D = 0.0892$ and $P/D = 1$ at Reynolds number of 3858. Consequently, it is possible to report that according to the operating conditions, the use of equilateral triangle cross sectioned coiled wire inserts in heat exchanger industry is convenient in order to obtain higher heat transfer and thermal performance.

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