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Pressure drop and heat transfer augmentation due to coiled wire inserts during laminar flow of oil inside a horizontal tube

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ABSTRACT

An experimental investigation has been carried out to study the enhancement in heat transfer coefficient by coiled wire inserts during heating of engine oil inside a horizontal tube. The test-section was a double-pipe counter-flow heat exchanger. The engine oil flowed inside the internal copper tube, while saturated steam, used for heating the oil, flowed in the annulus. First of all, the data were acquired for the heating of engine oil while flowing in the plain tube. Later, seven coiled wires having pitches of 12–69 mm and wire diameters of 2.0 mm and 3.5 mm were put one by one in the oil-side of test-section. The effects of Reynolds number and coiled wire geometry on the heat transfer augmentation and fanning friction factor were studied. Finally, two empirical correlations have been developed for predicting the heat transfer enhancement of these coiled wire inserts. These correlations predict the experimental Nusselt number in an error band of ±20 percent.

1. Introduction

The enhancement of heat transfer has always been a consistent endeavor of designers to reduce the heat exchanger dimensions and consequently the equipment material cost. The use of enhancement techniques to augment inside tubes convective heat transfer coefficient has been investigated for many years [1,2]. These techniques are classified into two main categories viz. active techniques and passive techniques. Active techniques are such methods which need external force field to augment heat transfer, for example, tube or fluid vibrations; on the other hand, passive techniques are such ones which need no external energy to enhance heat transfer, such as, tube inserts like twisted tapes, coiled wires, in-tube meshes and brushes and so on [3–6]. The heat transfer can substantially be increased by providing microfin tube or helical wire inserts [7]. It is also envisaged that during heating of liquid the rotational centrifugal convection has favorable effect [6,8].

Heat transfer during laminar flow is often envisaged in the engineering applications, especially at viscous liquid heating and cooling processes. The insert technology is useful for laminar region and coil wire insert has less pressure drop penalty in comparison to twisted tape insert [9]. However, the fanning friction factor of the tube with the coiled wire inserts increases [10]. In an engine oil heating system the oil-side heat transfer coefficient is very low; hence, the enhancement of oil-side heat transfer coefficient is imperative in order to increase the overall heat transfer coefficient of the system.

Chiou [11] investigated the effect of coiled wire inserts during cooling of oil inside the tubes. The enhancement in heat transfer coefficient was observed due to disruption of laminar sub-layer of liquid film and increasing the degree of turbulence. The coiled wires usually do not generate swirling flow. Chiou [11] also affirmed that in the heating mode operation, the swirling flow has favorable centrifugal convection effect, which can substantially increase the heat transfer coefficient between the flow and tube wall. In the cooling mode operation; however, the swirling flow may have an adverse centrifugal convection effect which may even reduce the convection effect.

Garcia et al. [12] investigated the heat transfer enhancement due to coiled wire inserts in laminar-transient-turbulent regimes for $80 < Re < 90{,}000$ and $2.8 < Pr < 150$ experimentally. They used water and water–propylene glycol mixtures at different temperatures to cover a wide range of fluid properties; however, they considered the properties of the fluid to be constant in each test run.

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Nomenclature

C	constant of Eq. (5)
$C(\alpha)$	constant of Eq. (6)
C_P	specific heat, J kg^{-1} K^{-1}
d	tube diameter, m
d_c	coil diameter, m
e	wire thickness, m
f	fanning friction factor
G	mass velocity, kg m^{-2} s^{-1}
h	heat transfer coefficient, W m^{-2} K^{-1}
k	thermal conductivity of oil, W m^{-1} K^{-1}
L	tube length, m
m	constant of Eq. (8)
$\dot{m}$	mass flow rate, kg s^{-1}
Nu	Nusselt number, h d$_i$ k^{-1}
p	coil pitch, m
Pr	Prandtl number
$R3$	performance evaluation factor, Eq. (12)
Re	Reynolds number, G d$_i$ μ^{-1}
T	temperature, K
u	velocity, m s^{-1}

Greek Letters

α	helix angle, degree
ΔP	pressure drop, Pa
ΔT_{lm}	logarithmic mean temperature difference, K
Φ	correction factor of Eq. (8)
μ	dynamic viscosity, Pa s
ρ	density, kg m^{-3}

Subscripts

c	coiled wire inserted tube
f	fanning friction factor
i	inside tube
in	inlet
h	hydraulic
m	mean
Nu	Nusselt number
o	outside tube
out	outlet
p	plain tube
w	tube wall

Uttarwar and Raja Rao [13] studied 7 coiled wires by using servotherm oil as test fluid. They covered a Reynolds number range from 30 to 700 and a Prandtl number range from 300 to 675.

A review of the existing literature revealed that numerous investigations have been carried out to study the effect of different passive techniques on augmentation of heat transfer coefficient in the turbulent flows [3], but limited research work has been envisaged on the use of these techniques during laminar flows and especially for the temperature-dependent-property fluids.

Therefore, present investigation has been undertaken to study the effect of coiled wire inserts on heat transfer and pressure drop during heating the engine oil in a laminar flow heat exchanger.

2. Experimental apparatus

The experimental set-up was a well instrumented heating system which was designed for the heat transfer studies of the engine oil. The schematic diagram of experimental set-up is shown in Fig. 1. This system mainly included a test-section (7), an oil storage tank (1), an oil cooler (8) and a flow meter with an accuracy of 6 lpm. Two types of oil pumps i.e. gear pump (P_1) and centrifugal pump (P_2) were used to circulate oil in the test-section (7). The pumps were connected in parallel arrangement to cover the entire range of oil flow rate. The test-section was a double-pipe counter-flow heat exchanger and the oil was heated inside the inner tube by absorbing heat from the saturated steam flowing in the annulus providing constant boundary condition. The inner tube with inner diameter of 26.04 mm, outer diameter of 28.57 mm and length of 1700 mm was made of copper. The outer tube with inner diameter of 152.4 mm, having same length as inner tube, was made of carbon steel. The inlet and outlet oil temperatures of test-section was measured using thermometers inserted in the small holes made in the inlet and outlet of the test-section and sealed to prevent any leakage. For the measurement of outside wall temperatures of the inner tubes, J-type thermocouples were installed at five axial locations; at each location, two thermocouples were used, one at the top and the other at the bottom of the tube. The leads of the thermocouples were soldered on the outer surface of the inner copper tube and were well insulated to avoid direct contact to the coolant water. For each temperature measuring station, one hole has been drilled on the outer pipe to facilitate the passing of thermocouples leads to the temperature indicator. The holes were completely waterproofed to prevent water leakage. All the inlet, outlet, and wall temperatures were measured manually 3 times with accuracy of 0.1 °C in the time steps of 10 min, and the average values were used for further analysis. Manometers at inlet and outlet of the test-section were provided to measure the oil pressure drop. All instruments were calibrated prior to installation and the test-section was well insulated with glass wool to prevent heat loss.

For the augmentation of heat transfer in the oil heating system, coiled wire inserts were put in the oil-side of the test-section. Fig. 2 shows the dimensional parameters of coiled wire inserts. These coils of different pitches were fabricated from carbon steel wires of different diameters and were inserted throughout the full length of the test-section tube. The characteristic parameters of coiled wire inserts are given in Table 1.

In the beginning of experiments, the desired oil flow rate was adjusted by needle valves which were located on both sides of the pumps (P_1 and P_2). The oil flow rate was measured by the flow meter provided in the set-up. The oil was heated by steam while passing through the test-section. The oil flowing from test-section was cooled in the oil cooler (8) before sent back to the oil storage tank (1). After reaching the steady state, oil flow rate, oil temperatures and pressures at inlet and outlet, and temperatures at outer surface of inner tube were measured. A total of 120 experimental runs were carried out. The range of different parameters is as follows:

Reynolds number	$10 < \mathrm{Re} < 1500$
Prandtl number	$120 < \mathrm{Pr} < 300$
Helix angle	$49^\circ < \alpha < 82^\circ$
Twist ratio of coil	$0.46 < p/d_i < 2.65$
Wire thickness ratio	$0.077 < e/d_i < 0.134$

The thermo-physical properties of the oil are given in Table 2.

Further, an uncertainty analysis of all the experimental results was carried out using the method proposed in [14], and it was

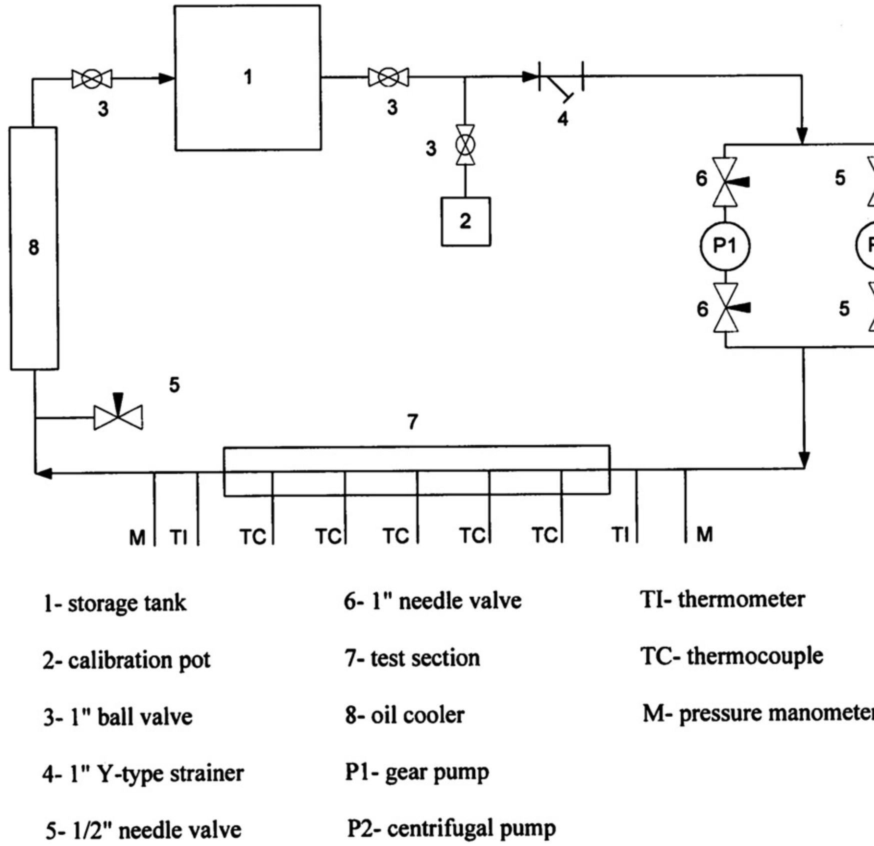

Fig. 1. Schematic view of test set-up.

found that the expected experimental error was within ±7.5 percent for all the runs.

3. Results and discussion

First of all, the data have been acquired for the heating of oil inside a plain tube to establish the integrity of the experimental set-up. The oil-side experimental fanning friction factor, f, has been determined using the following Equation.

$$f = \frac{\Delta P d_i}{2\rho u_m^2 L} \tag{1}$$

The oil-side fanning friction factor, f, is a function of oil Reynolds number, Re. The variation of experimental fanning friction factor, f, with the oil Reynolds number, Re, has been shown in Fig. 3. It is observed that with the rise of oil Reynolds number the fanning friction factor reduces. The experimental fanning friction factor has changed from 1.11 to 0.02 as the Reynolds number has increased from 22 to 1320. Therefore, the theoretical fanning friction factor has also been calculated using the Equation (2) for laminar flow.

$$f = 16/\mathrm{Re} \quad 100 < \mathrm{Re} < 1500 \tag{2}$$

The variation of theoretical fanning friction factor with Reynolds number has been shown by a solid straight line in Fig. 3. At Reynolds numbers of 22 and 1320 the deviation in experimental fanning friction factor, f, is 58 percent and 29 percent, respectively. However, for the other oil flow rates the experimental fanning friction factors, f, are in an error band of +16 percent to −7 percent with an average deviation of 5 percent with those predicted by Equation (2). This fairly good agreement between the experimental and theoretical values of fanning friction factor, f, establishes the integrity of the experimental set-up. These data from plain tube shall also be used to compare the performance of coiled wire inserts.

Fig. 4 has been drawn taking Reynolds number as abscissa and fanning friction factor as ordinate for the coil inserts having wire diameter of 2.0 mm, while the coil pitch has varied from 65 mm to

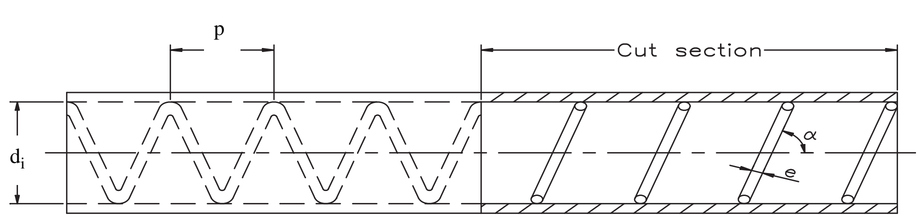

Fig. 2. Dimensional parameters of the coils.

Table 1
Characteristic parameters of the coiled wire inserts.

Tube no.	e (mm)	p (mm)	α (degree)
0	0	∞	–
1	2	65	49.2
2	2	47	58.1
3	2	28	69.7
4	2	12	81.0
5	3.5	69	45.7
6	3.5	49	55.3
7	3.5	26	69.8

12 mm. For the sake of comparison the experimental data for the plain tube are also shown in Fig. 4. The solid line depicts the variation of theoretical fanning friction factor with the oil Reynolds number for the plain tube. Also for the tubes having coiled wire inserts, the experimental fanning friction factor, *f*, reduces with the rise in Reynolds number for all the coil pitches. In fact, the inserted coil pitch has a little bearing on the fanning friction factor up to the oil Reynolds number of 500. The rise in fanning friction factor due to coiled wire insert is 22 percent and 35 percent, for the Reynolds number of 40 and 500 respectively, irrespective of the coil pitch. Equation (3) represents the best fit for all the coiled wire inserts with wire diameter of 2.0 mm in the range of $20 < \mathrm{Re} < 500$.

$$f = 16.8/\mathrm{Re}^{0.96} \tag{3}$$

This equation is illustrated with the dotted line in Fig. 4. Equation (3) predicts the experimental data in an error band of ± 8 percent. As the oil Reynolds number exceeds the value of 500, there is visible a change in the trend of the variation of fanning friction factor with the oil Reynolds number. The fanning friction factor for the coil having the pitch of 65 reduces from 0.03 to 0.02 for the rise in Reynolds number from 650 to 1150. For the other coil pitches viz. 47 mm, 28 mm and 12 mm there is not any noteworthy variation in the fanning friction factor with the rise in Reynolds number when it exceeds 500. In fact, for $\mathrm{Re} > 500$ the fanning friction factor, *f*, stays close to a constant value of 0.04. Further, for the oil Reynolds number higher than 500 the effect of coil pitch is negligible on the fanning friction factor except for the coil with the pitch of 65 mm. However, the Reynolds number of oil for plain tube keeps on reducing. Therefore, the fanning friction factor for the tubes with coil inserts, as the Reynolds number higher than 500, is much higher in comparison with that of plain tube. In fact, with the rise in Reynolds number the coiled wire induces a secondary flow, which in turn, promotes turbulence that leads in fanning friction factor growth [13].

In order to further investigate the effect of coil pitch another coiled wire insert with wire diameter of 3.5 mm has been investigated. Fig. 5 has been drawn taking oil Reynolds number, Re, as abscissa and fanning friction factor, *f*, as ordinate. It is observed that for the wire diameter of 3.5 mm the effect of coil pitch on fanning friction factor is significant and the fanning friction factor reduces with rise in oil Reynolds number up to 350. The trend of change in fanning friction factor is in coherence with that for the plain tube.

Table 2
Thermo-physical properties of engine oil.

Property	Temperature, °C			
	30	50	70	90
Viscosity, μ, Pa s	0.0379	0.0192	0.011	0.007
Specific heat, *Cp*, J kg^{-1} K^{-1}	1925.65	1999.8	2074.0	2148.2
Thermal conductivity, k, W m^{-1} K^{-1}	0.1621	0.1600	0.1585	0.1568
Density, ρ, kg m^{-3}	855.6	841.1	826.6	812.1

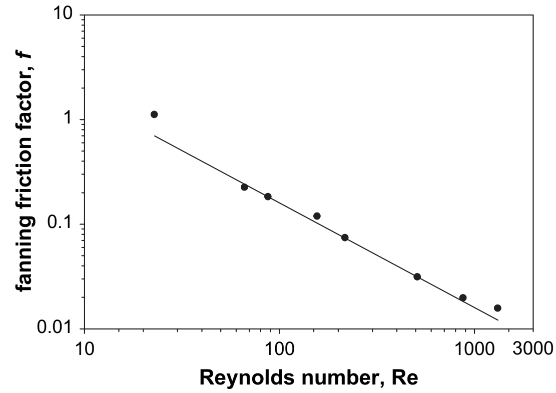

Fig. 3. Variation of fanning friction factor with Reynolds number for the flow of oil inside a plain horizontal tube.

At a given flow rate up to the Reynolds number of 350 the fanning friction factor rises with the decrease of coil pitch. The fanning friction factor, *f*, for the coil wire insert with different coil pitches is also given in Fig. 5 for the Reynolds numbers of 20 and 350. For the coil pitch of 26 mm the fanning friction factor is nearly 3.25 times than that of plain tube at the Reynolds number 350. However, with the rise in coil pitch, at Reynolds number 350, the fall in fanning friction factor is envisaged. The fall in fanning friction factor with the rise in Reynolds number is sharper for the tubes having coiled wire inserts in comparison with that of plain tube. For example, at the coil pitch of 69 mm the fanning friction factor at the Reynolds number 20 is 2.1 times that of the plain tube; on the other hand, at Reynolds number 350 the fanning friction factor, *f*, for the same coil pitch is only 1.33 times of the plain tube.

For all the coiled wire inserts, the change in fanning friction factor with oil Reynolds number has become insignificant as the Reynolds number is higher than 350. In fact, a trivial increase in fanning friction factor is also observed beyond Reynolds number of 350 in the case of coil pitch of 26 mm. In fact, as Reynolds number is increased beyond 350, the fanning friction factor has become more or less constant at 0.07, 0.12 and 0.17 for the coil pitches of 69 mm, 49 mm, and 26 mm, respectively. Hence, the effect of oil flow rate on fanning friction factor is not significant when Reynolds number is higher than 350.

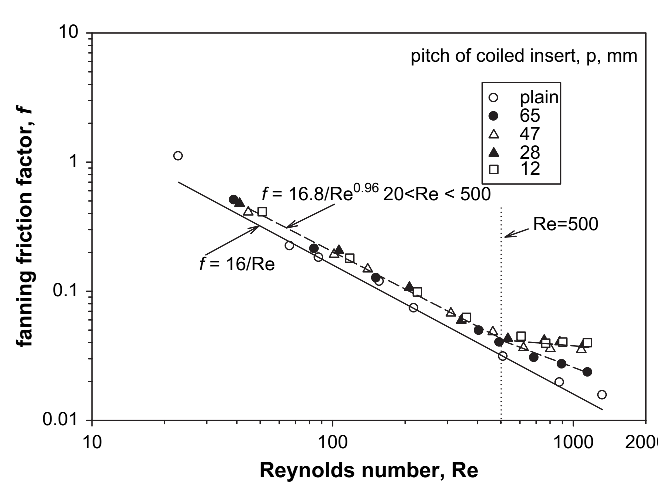

Fig. 4. Variation of fanning friction factor with the Reynolds number for different pitches of coiled insert with 2.0 mm wire thickness.

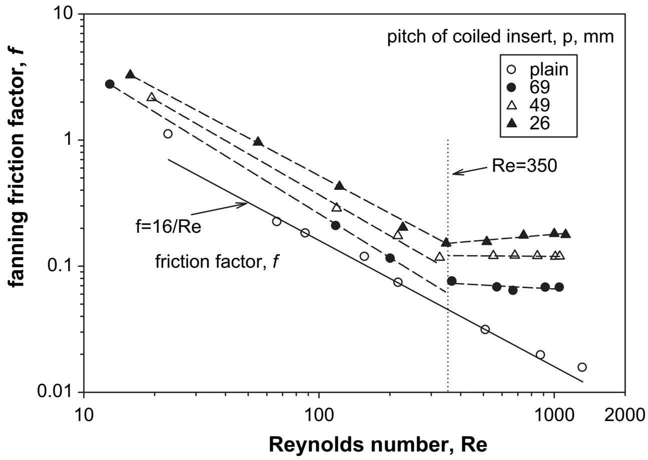

Fig. 5. Variation of fanning friction factor with the Reynolds number for different pitches of coiled insert with 3.5 mm wire thickness.

The oil-side heat transfer coefficient, h, has also been calculated to estimate the heat transfer improvement. Following Equation has been used to determine the oil-side heat transfer coefficient, h.

$$h = \frac{\dot{m}C_P(T_{\rm in} - T_{\rm out})}{\pi d_{\rm i} L \Delta T_{\rm lm}} \tag{4}$$

The bulk properties of the flowing fluid have been evaluated at the average of inlet and outlet temperatures of the oil. The variation of oil-side Nusselt number with the oil Reynolds number for a plain tube has been shown in Fig. 6. The solid line shows the variation of oil Nusselt number predicted by Equation (5) for the heat transfer of plain tube.

$$\mathrm{Nu} = C\left(\frac{\mathrm{Re\,Pr}}{L/d_{\rm i}}\right)^{1/3}\left(\frac{\mu}{\mu_{\rm w}}\right)^{0.14}; \; C = 1.86 \tag{5}$$

The experimental Nusselt numbers using the heat transfer coefficient from Equation (4) are approximately 24 to 41 percent higher (with an average value of 19 percent) than those predicted by Equation (5), in the Reynolds number range of 60–1400. If the constant, C, of Equation (5) is modified to 2.32, the above equation predicts the experimental data in an error band of $\pm$10 percent with an average deviation of 5 percent. The dotted line in Fig. 6 predicts the Nusselt number using Equation (5) with modified value of constant, 'C' as 2.32. The effect of coiled wire insert pitch and wire thickness on oil Nusselt number has also been investigated following the studies on plain tube. The results of plain tube were used as reference for comparing. Fig. 7 depicts the variation of Nusselt number with Reynolds number for the different coil pitches with wire diameter of 2.0 mm. The results of plain tube are also shown in Fig. 7 as comparison. It is observed that before Reynolds number reaching 750, the trend of change in Nusselt number is just same as that in the case of plain tube. Further variation in coil pitch has little bearing on the enhancement in Nusselt number. In fact, data from all coiled wire inserts lie in a range of $\pm$20 percent of the best fit line. Initially, at the oil Reynolds number 100, the coil wire inserts enhance the Nusselt number 2.2 times and the enhancement in fanning friction factor is only 1.26 times. At oil Reynolds number, 700 the enhancement in Nusselt number is 1.8 times and the fanning friction factor is the lowest for the 65 mm pitch coiled wire insert. However, there is a significant rise in Nusselt number when Re > 750. At oil Reynolds number 775 the Nusselt number was increased to 1.95 times due to coiled wire insert and to 2.93 times at Reynolds number of 1375. Therefore, it can be concluded that the coiled wire pitch of 65 mm for the wire thickness of 2.0 mm should be selected to enhance the heat transfer as it offers the lowest fanning friction factor in comparison with other inserts for the same order of enhancement in heat transfer coefficient.

The variation of Nusselt number with oil Reynolds number for different coil pitches with wire diameter of 3.5 mm has been depicted in Fig. 8. Similar to the coiled wire thickness of 2.0 mm, a uniform increase in oil Nusselt number has been observed up to the oil Reynolds number of 750 for the coiled wire thickness of 3.5 mm as well. At the Reynolds number of 100 the enhancement in Nusselt number is up to 2.2 for the coiled wire having the pitch of 26 mm. For the same coil, the enhancement in Nusselt number has increased to 3.2 for the oil Reynolds number of 1000. The Nusselt number for the coil pitch of 49 mm is only 13 percent more than that of the coil having 69 mm pitch, at the Reynolds numbers less than 750. The performance of 49 mm coil pitch insert is improved significantly when the Reynolds number exceeds 750 and at high Reynolds number it becomes very close to that of the insert having pitch of 26 mm.

In order to investigate the effect of Prandtl number, Pr, on the Nusselt number, oil has been introduced to the heat exchanger with three different temperatures (39, 55, and 65 °C) for tube numbers 1,

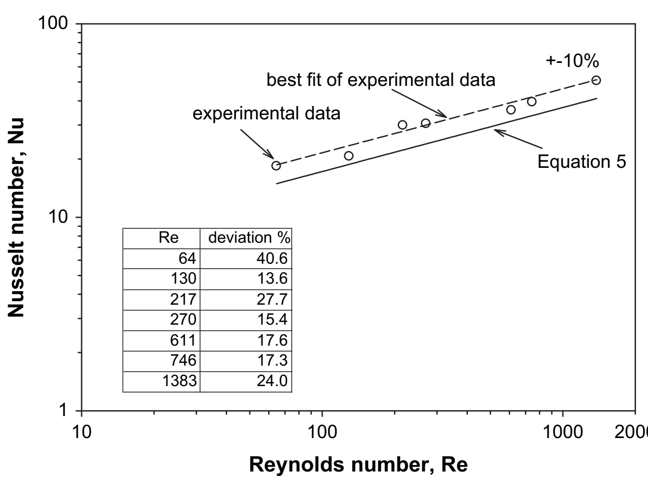

Fig. 6. Variation of Nusselt number for the flow inside a plain tube.

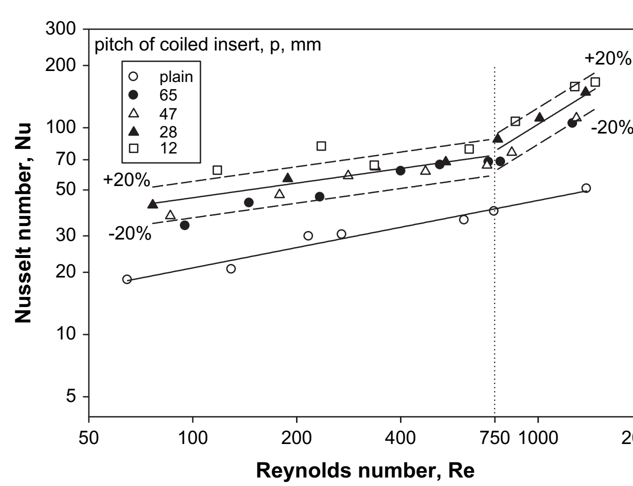

Fig. 7. Variation of Nusselt number with Reynolds number for different pitches of coiled insert having wire diameter of 2.0 mm.

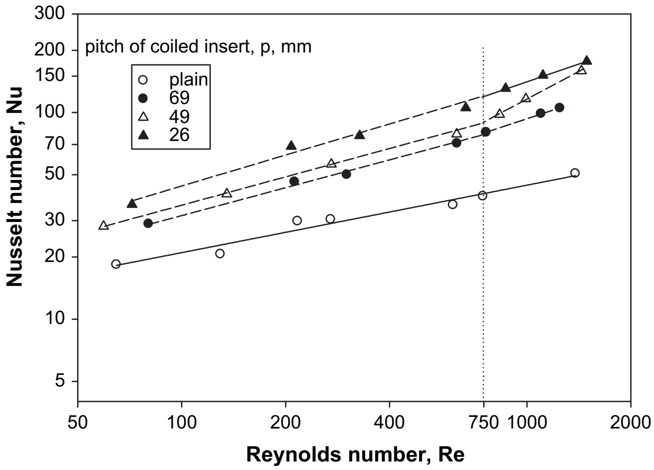

Fig. 8. Variation of Nusselt number with Reynolds number for different pitch of coiled insert having wire diameter of 3.5 mm.

2, 5, and 6 (Table 1), while tube wall temperature was maintained constant. The collected data and evaluation of the Nusselt number showed that Nusselt number varies with $\mathrm{Pr}^{0.33}$.

It is noted that Nusselt number is a function of Reynolds number, Helix angle, and Prandtl number as follow:

$$\mathrm{Nu} = C(\alpha)\mathrm{Re}^{m}\mathrm{Pr}^{0.33}\Phi \tag{6}$$

where 'Φ' is a correction factor to take into account the change in viscosity of fluid due to temperature variation across the thermal boundary layer, and is calculated as $\Phi = (\mu/\mu_w)^{0.14}$. The '$m$' can be evaluated from Fig. 7 and Fig. 8 as the slope of the graphs which can be expressed as

$$m = 0.26(\tan\alpha)^{-0.37} \tag{7}$$

Investigating the variation of $C(\alpha)$ with $\tan\alpha$, $C(\alpha)$ can be stated as follow:

$$C(\alpha) = 1.63\tan(\alpha) \tag{8}$$

Substituting Equations (7) and (8) in Equation (6), the following correlation for Nusselt number is developed:

$$\mathrm{Nu} = 1.63\tan\alpha(\mathrm{Re})^{\left[0.26(\tan\alpha)^{-0.37}\right]}\mathrm{Pr}^{0.33}(\mu/\mu_w)^{0.14} \tag{9}$$

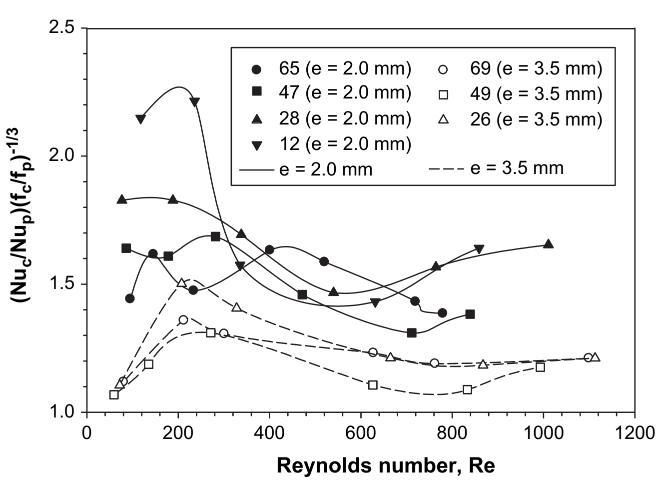

Fig. 9. Variation of the performance of different coiled wire inserts with Reynolds number.

The above correlation is valid in the range of $49° < \alpha < 61°$, $80 < \mathrm{Re} < 900$, and $1.8 < p/d_i < 2.65$.Similarly, for tube numbers 3 and 7, the following correlation is developed.

$$\mathrm{Nu} = 0.91\tan\alpha(\mathrm{Re})^{\left[0.29(\tan\alpha)^{-0.21}\right]}\mathrm{Pr}^{0.33}(\mu/\mu_w)^{0.14} \tag{10}$$

which is valid in the range of $61° < \alpha < 73°$, $90 < \mathrm{Re} < 950$, and $1.0 < p/d_i < 1.8$; but no appropriate correlation could be found for tube number 4 ($\alpha = 82°$).

Equation (10) is similar to the proposed correlation by Uttarwar and Raja Rao [13] for the range of $32° < \alpha < 61°$. However, for $\alpha > 61°$, not covered by their correlation this equation is unique. The mean deviations of the heat transfer coefficients calculated by Equations (9) and (10) are 7 and 8 percent respectively; while the standard deviation values are 13.84 and 15.42 percent, respectively.

In general, the developed correlations predict the experimental Nusselt number in an error band of ±20%.

The enhancement in heat transfer by inserts is accompanied by the rise in pressure drop. Therefore, as a common practice, it is useful to estimate the heat transfer enhancement and pressure drop increase simultaneously. For this purpose, the performance criterion outlined by Bergles et al. [15] has been used as $R3 = \mathrm{Nu_c}/\mathrm{Nu_p}$, where $\mathrm{Nu_c}$ is the Nusselt number of coiled tube and $\mathrm{Nu_p}$ is the Nusselt number of the plain tube for equal pumping power and heat exchanging surface area. To meet this limit, the following relationship shall be maintained [12]:

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

Therefore, the performance evaluation factor, $R3$, can be written as [16]:

$$R3 = \left(\frac{\mathrm{Nu_c}}{\mathrm{Nu_p}}\right)\left(\frac{f_c}{f_p}\right)^{-1/3} \tag{12}$$

In Fig. 9 the variation of performance evaluation factor, $R3$, with Reynolds number, Re, (range 50–1000) is shown. For all the inserts the performance evaluation factor, remains more than unity. However, fore all the flow rates the performance of wire thickness 2.0 mm is much better than the performance of insert with wire thickness of 3.5 mm. It is interesting to note that the performance of all inserts increases up to 200 Re and then reduces. The variation in performance evaluation factor, $R3$, at 12 mm pitch is the highest and beyond Reynolds number of 200 the performance of this coiled wire insert sharply detoriates. When the Reynolds number further increases beyond 800 the performance of the insert with 2.0 mm wire thickness tends to improve and the performance evaluation curves for the 3.5 mm wire diameter tend to converge.

4. Conclusions

Based on this study, following conclusions have been drawn.

1. The experimental fanning friction factor, f, for plain tube is in a range of −7 to +16 percent of that predicted by the theoretical model, Equation (2). However, experimental Nusselt number for plain tube is 24–41 percent higher than that predicted by the Equation (5). Further, the modified Equation (5) predicts the experimental Nusselt number in an error band of ±10 percent.
2. Irrespective of coil pitch, the rise in fanning friction factor, f, due to the coiled wire insert of 2.0 mm wire thickness is in a range of 22–35 percent for the Reynolds numbers less than 500. For Reynolds numbers higher than 500, the reduction in coil pitch causes increase in fanning friction factor. The coiled wire insert

of 3.5 mm wire thickness enhances the fanning friction factor as the coil pitch reduces, up to the Reynolds number 350, and further increase in oil Reynolds number does not reduce the fanning friction factor.

3. Coiled wire insert of 2.0 mm wire thickness enhances the oil-side Nusselt number, while the coiled wire pitch has a little effect on the Nusselt number. The insert enhances the Nusselt number by 2.2 times than that of the plain tube. However, for the coiled insert of 3.5 mm wire thickness, the enhancement in Nusselt number can be attained up to 3.2 times.
4. According to the performance evaluation factor, it was seen that wire coil inserts with lower wire diameters have better performance, especially at low Reynolds numbers. Also, the increase in the coil pitch made a moderate decrease in performance parameter.
5. Two empirical correlations have been developed which predict the enhanced heat transfer coefficients within an error band of $\pm20\%$.

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