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Thermal Characterization of Turbulent Flow in a Tube With Discrete Coiled Wire Insert

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Abstract:

The present study aims to investigate the effect of adding discrete coiled wire to enhance the heat transfer in heat exchanger. The coiled wire used in experiments were made from steel circular cross section wire with (b = 4 mm) diameter. The outer coil wire's diameter was (B=36.5 mm) and length after been winding is (l = 100 mm), the discrete distribution of coil wires, by different spacing (S) between them with (S/l = 0.22 ,0.57 &1.2) ,which require to distribute 10, 8 & 6 coiled wires along test section respectively, were considered in the experimental study . The experiments have been carried at constant heat flux rate and different flow rates of air that started from (Re=6000) and increased gradually to reach the following Re. (7500, 9000, 10500, 12000 and 13500). The results show that the use of the S/l=0.57 results in the augmentation of pressure loss around 33.2% higher than that of the S/l=0. In addition, it can be found that for using coiled wire turbulator with spacing ratios(0, 0.22, 0.57,& 1.2), the improvement of average heat transfer rate is, respectively, about 112%, 119.3%, 141.3%&132% higher than those the plain tube while the friction factor is 300%,392% ,432%, and 322% times of the plain tube.

Key Words: Heat exchanger enhancement; Discrete coiled wire; turbulent flow; Pressure drop.

التوصيف الحراري لجريان مضطرب بأنبوب مزود بسلك ملفوف متقطع كأداة حشر

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الخلاصة:

يهدف البحث لدراسة تأثير إضافة ملفات متقطعه لتحسين انتقال الحرارة في المبادل الحراري. الملف المستخدم في التجارب مصنوع من مادة الستيل دائري المقطع قطره (b=4 mm) . القطر الخارجي للملف كان (B=36.5 mm) وطوله بعد اللف هو (l=100 mm)، توزيع التقطيع للملف بفراغات (S) بـ (S/l = 0.22 ,0.57 &1.2)، والتي تتطلب توزيع 10، 8 و6ملفات على طول جزء الاختبار تباعاً ، كما هو مفروض في الدراسة العملية.التجارب أُجريت عند معدل ثابت للفيض الحراري وبمعدلات مختلفة لجريان الهواء ابتداء من (Re=6000) وتتزايد بشكل تصاعدي لتصل Re كالتالي (7500، 9000، 10500، 12000 و 13500). النتائج أظهرت أن استخدام s/l= 0.57 النتائج تزيد من خسائر

الضغط بنسبة 33.2 % أعلى من S/l= 0. بالإضافة ، يمكن أن يوجد أن استخدام ملف الحشر بنسب فراغية (0، 0.22، 0.57 و1.2) معدل تحسين انتقال الحرارة يكون بالتتابع 112 %، 119.3 %، 141.3 و132 % أو هو أعلى من قيمته في الأنبوب الأملس بينما معامل الاحتكاك يزداد بمقدار 300 %، 392 %، 432 % و 322 % مما هو علية في الأنبوب الأملس.

كلمات مرشدة : تحسين المبادل الحراري ; ملف حشر متقطع : جريان مضطرب ; خسائر الضغط.

Nomenclature

A	heat transfer area, m^2
b	wire diameter, m
B	coiled wire diameter, m
Cp	heat capacity of air, J/kg .K
d	small end diameter of conical ring ,m
D	inner diameter of test tube, m
f	friction factor
h	heat transfer coefficient, $W/m^2.K$
I	current, A
k	thermal conductivity of air, W/m. K
L	length of test tube, m
l	coiled wire length, m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number (=h D/k)
ΔP	pressure drop, Pa
Pr	Prandtl number ($=\frac{\nu}{\alpha}$)
Q	heat transfer rate, W
Re	Reynolds number ($=\frac{uD}{\nu}$)
S	space between coiled wire, m
s	ring pitch, m
T	temperature, °C
t	thickness of test tube, m
S/l	relative spacing, ratio between the coiled wire spacing and its length
U	mean axial velocity, m/s
V	voltage, v.
Y	relative spacing, (S/l)
α	thermal diffusivity, ($=k/\rho c_p$)
η	thermal enhancement efficiency
ν	kinematics viscosity, m^2/s
ρ	density, kg/m^3
μ	dynamic viscosity, Ns/m^2

Subscripts

a	air
ave	average
b	bulk
i	inlet
m	mean
o	out
p	plain tube
pp	pumping power
t	Turbulator
w	wall

1. Introduction

Enhancing the heat transfer process in the heat exchanger is of the topics that occupied the attention of researchers for decades. Several enhancement techniques have been developed in order to improve heat transfer and reduce a heat exchanger size. Most techniques deal with flow destabilization to improve fluid mixing within the exchanger [1-5]. One important group of heat transfer enhancement devices are turbulators in forms of coiled-wire, conical and circular rings. Mechanistically, turbulator increases the composite velocity, enhances the radial turbulent fluctuation which causes an efficient eruption of the thermal boundary layer, Thianpong et, al. [6]. Turbulators in several shapes have been proposed. Thianpong et, al. [6], studied heat transfer, friction factor and thermal performance characteristics in a tube equipped with twisted-rings which were experimentally investigated. The investigation also concerns with the effects of width ratio and pitch ratio of the rings. Paisarn [5], performed to study the heat transfer characteristics and the pressure drop of the horizontal concentric tube with twisted wires brush inserts. Effect of twisted wires density, inlet fluid temperature, and relevant parameters on heat transfer characteristics and pressure drop are considered. They found that the twisted wire brushes insert have a large effect on the enhancement of heat transfer, however, the pressure drops also increase. Zimparov [7], predicted the friction factors and heat transfer coefficients for turbulent flow in corrugated tubes with twisted tape inserts. Bharadwaj et al [8] experimentally determined the heat transfer and pressure drop in a spirally grooved tube with twisted tape insert. In another study, Smith and Pongjet [9] carried out an experiment to investigate the heat transfer and friction factor of the fully developed turbulent airflow through a uniform heat flux tube fitted with diamond-shaped turbulators in tandem arrangements. Strong turbulence and recirculation flow were generated by using tandem diamond-shaped turbulators (D-shape turbulator) connected to each other by a small rod and placed inside the test tube. Others have studied the impact of the form section of the wire to improve the amount of heat transmitted. Promvonge [10], 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. Wang and Sunden [11] 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. Sibel et al [12], where they used the coiled wire which has equilateral triangular cross section and was inserted separately from the tube wall. In the present study, the main concern of this study, is to investigate experimentally the heat transfer characteristics and pressure drop of a tube with built-in discrete coil-wire. The effects of various relevant parameters and coil spacing are also investigated. The result obtained from the tube with different number of coil-wire built-in at constant tube length of heat exchanger, is compared with those without coil-wire. Correlations for the heat transfer coefficient and friction factor are proposed respectively.

2. Apparatus Description:

The open loop experimental facility used in the experiments is schematically shown in **Figure (1)**

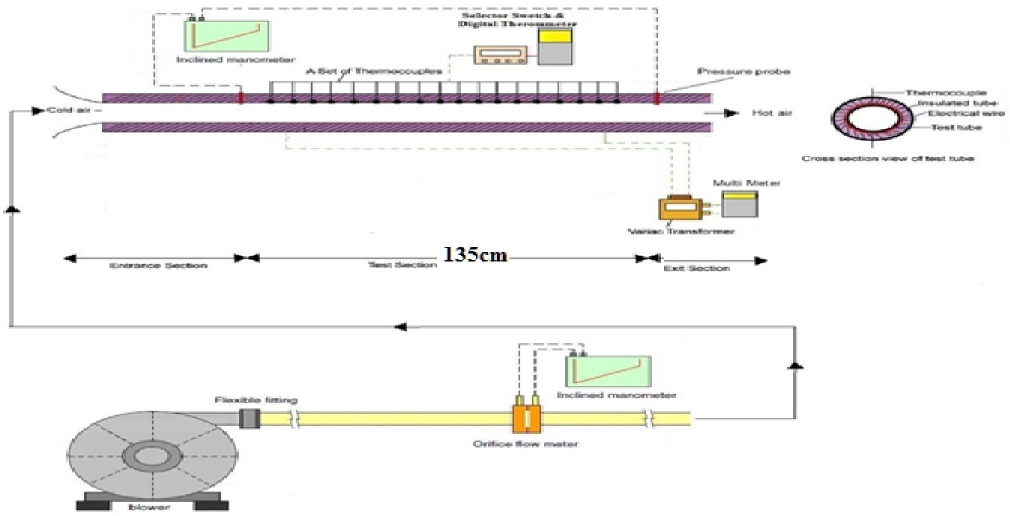

Fig .(1) Schematic diagram of the experiment set-up

The facility includes a blower of 6.5 kW, orifice meter to measure the flow rate, the heat transfer test tube with a coiled wire insert. The aluminum test tube has (L=1350 mm) length, (45 mm) inner diameter (D_i), (50 mm) outer diameter (D_o), and (2.5 mm) thickness (t). The tube was heated continually by winding flexible electrical wire providing a uniform heat flux. The electrical output power was controlled by a variac to obtain a required voltage. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings. A selector switch and digital thermometer were used to get multi-channel temperature measurements. Eighteen thermocouples were fixed on the local wall of the tube to achieve the temperature distribution along the Aluminum pipe surface. One thermocouple was placed at the inlet of the test section and two thermocouples were placed at outlet of test section to measure the temperature of the bulk air at inlet and outlet of test section, respectively. The mean local wall temperature was determined by means of calculations based on the reading of (chrome alumel) thermocouples. Four thermocouples were attached on the multilayer thermal insulation surface to determine the heat losses by conduction to the surrounding. To reduce the axial heat losses from the test section, a Teflon bell mouth which was made from Teflon was fixed at the inlet and another Teflon piece has been attached at the exit of test section. An inclined manometer is used to measure the pressure drop along the test section.

Figure (2) shows the details of coiled wire turbulator. Coil-wire insert is fabricated by bending a (b= 4 mm) diameter steel wire into a coil with a coil diameter of(B=36.5 mm)and coil pitches of(9 mm), which have a length of (l = 100 mm) after been winding .

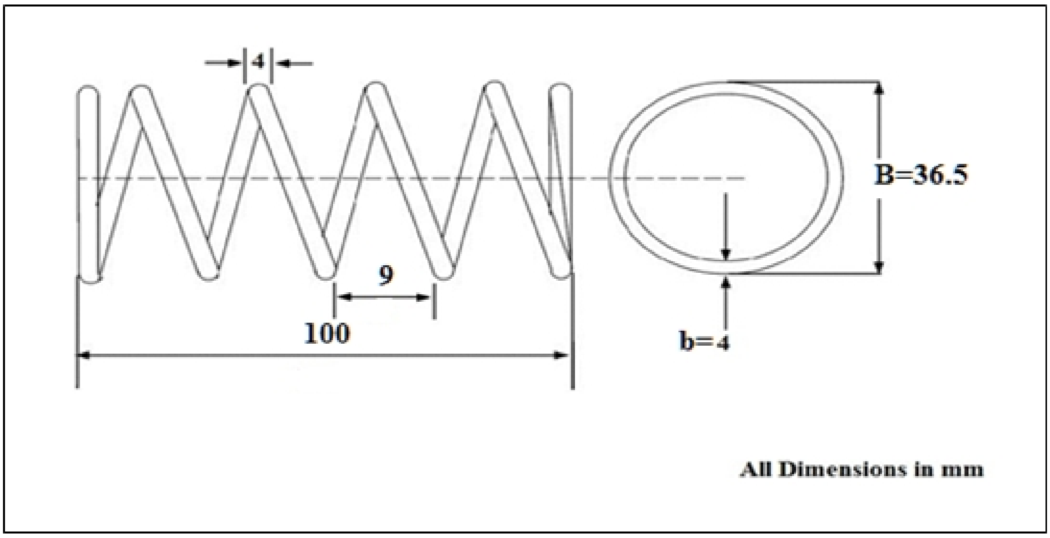

Fig .(2) Details of coiled wire turbulator.

Figure (3) shows the details of test tube with the arrangement of coiled wire insert. **Figure(3a)** shows a Continuous distribution of twelve coil wire along the test section, while **Figures (3b, 3c, 3d)** shows the discrete distribution of coil wires, a different number of coiled wires (10, 8 & 6) have been distributed along test section to achieve different relative spacing (S/l = 0.22 ,0.57 & 1.2) respectively. Teflon rings were manufactured according to the wire diameter in order to fix the coiled wire separately from the tube wall. These Teflon rings were densely attached onto the inserts, thus the contact of inserts with tube inner wall was prevented, as shown in Figure(3).

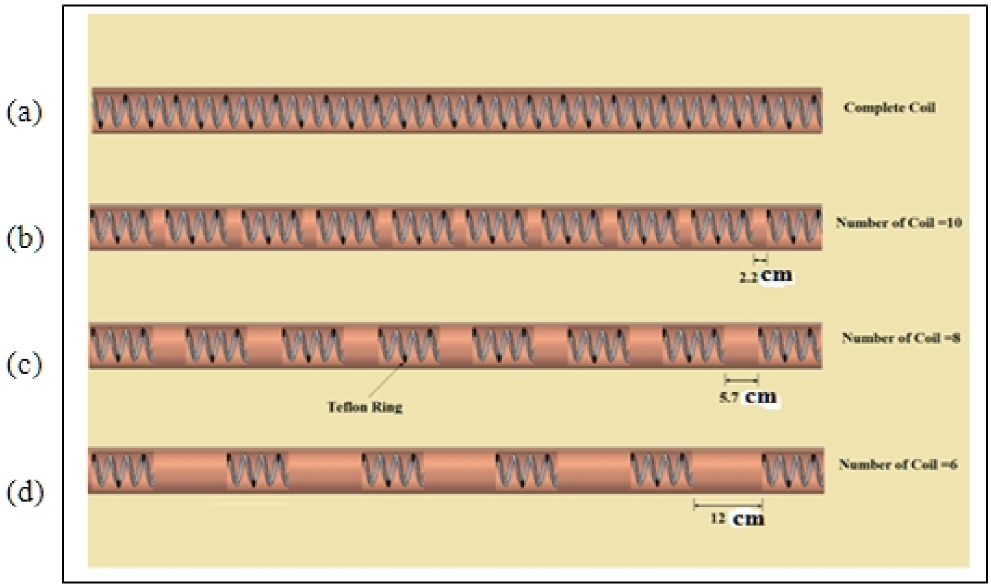

Fig .(3) Details of test tube with coiled wire turbulator

3. Processing Data

The experiments have been carried at constant heat flux rate and different flow rates of air that enters the test section. In the experiments, air flow rate have been started from (Re=6000) and increased gradually to reach the following Re. (7500, 9000, 10500, 12000 and 13500) while the heat flux is constant. Four cases of coiled wire in different distribution forms, which is used as Turbolater, are performed at same heat flax with different Reynolds numbers as mentioned above, in each case of coiled wire distribution.

Before any data were recorded, the system was allowed to approach the steady state. A steady state condition was usually obtained within 3 hours in which all of thermocouples were recorded and noted. The various characteristics of the flow, the Nusselt number and the Reynolds number, were based on the average of tube wall temperature and average bulk air temperatures, Promvonge [13]. So, the local wall temperature, inlet and outlet air temperature, the pressure drop across the test section and the air flow velocity were measured for heat transfer of the heated tube. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature.

To analyze the process of heat transfer from wall of test tube to the working fluid, a software program (MATLAB-2009) is developed to perform all calculations of the data analysis.

In the present work, air is used as the tested fluid and flowed through a uniform heat flux and insulated tube. The net heat input (Q_1) to the fluid was determined from the electrical energy input to the system as follows:

$$Q_1 = V.I - Q_{losses} \qquad \text{…………………..… (1)}$$

Where: Q_{losses} is the heat losses by convection and radiation from the test section to the surroundings.

At steady state condition, the heat transfer rate to the cold air in the test section (Q_2), can be expressed as:

$$Q_2 = \dot{m}\, Cp\, (T_{bo} - T_{bi}) \qquad \text{…………………..… (2)}$$

The bulk temperature of the fluid at any axial position along the tube axis was computed by assuming a linear temperature variation along the length since the experiment results show that the temperature distribution along the tube is linear. The heat supplied by the electrical winding in the test tube is found to be 0.8-2.7% higher than the heat absorbed by the fluid for the thermal equilibrium test due to convection and radiation heat losses from the test section to the surroundings. The set of data taken in a run were accepted only if the difference between the net heat input and the enthalpy rise was less than 5%, Mohammed [14]. So that the actual heat input to the test section was taken as the average of Q_1 and Q_2.

$$Q_{average} = \frac{Q_1 + Q_2}{2} \qquad \text{…………………..… (3)}$$

The inner tube-side heat transfer coefficient, $\bar{h}$, can be calculated from the average heat transfer rate obtained from

$$Q_{average} = \bar{h} A_s (\bar{T}_w - T_b) \quad \text{.....................(4)}$$

The heat transfer coefficient by convection from the wall of test section can be written as:

$$\bar{h} = \frac{Q_{average}}{A_s(\bar{T}_w - T_b)} \quad \text{.....................(5)}$$

A_s = surface area of the test tube (m^2).
The overall bulk air temperature (T_b) [15], which is calculated as:

$$T_b = \frac{T_{bo} + T_{bi}}{2} \quad \text{.....................(6)}$$

Where: T_{bi} :the air temperature at the entrance of the test tube (K).
T_{bo} :the air temperature at the exit of the test tube (K).
$\bar{T}_w$: is the average test tube surface temperature that is recorded from eighteen points as follows:

$$\bar{T}_w = \frac{\sum_{i=1}^{18} T_{wi}}{18} \quad \text{.....................(7)}$$

Where: T_{wi} : the local test tube surface temperature.
The average Nusselt number can be calculated from the following equation, Incropera'[15]:

$$\overline{Nu} = \frac{\bar{h}.D}{k} \quad \text{.....................(8)}$$

The friction factor (f) is found from:

$$f = \frac{\Delta P_t}{\left(\frac{L}{D}\right)\left(\frac{\rho U^2}{2}\right)} \quad \text{.....................(9)}$$

Where: ΔP_t = pressure drop through the test tube & U is the mean air velocity in the tube.

The overall enhancement efficiency (η) is expressed as the ratio of the, h_t of an enhanced tube with coiled wire insert to that of a smooth tube, h_p at a constant pumping power is introduced by Promvonge & Eiamsa-ard [13]:

$$\eta = \frac{h_t}{h_p}\bigg|_{pp} = \frac{\frac{Nu_t}{Nu_p}}{[\frac{f_t}{f_p}]^{1/3}} \quad \text{......................... (10)}$$

4. Results and discussion

4.1. Verification of the experimental results

First of all, the results obtained from experiments on heat transfer and friction factor characteristics in the plain tube are verified in terms of Nusselt number and friction factor. (**Figures 4 and 5)** show the data verification of the heat transfer rate and friction factor of the plain tube. The results of the present plain tube are also compared with those from the proposed correlations by Promvonge [13] for The Nusselt number and proposed correlations by Blasius [cited in 15] and Promvonge [13] for the friction factor. In the figures, the present results reasonably agree well with the correlations within 5-12% deviation for Nusselt number (Nu) and 10-15% for friction factor (f). The Blasius correlations for Nusselt and friction factor [cited in 13] are:

$$\mathbf{Nu = 0{:}023\ Re^{0.8}\ Pr^{0.4}} \quad \text{......................... (11)}$$

$$\mathbf{f = 0.448\ Re^{-0.275}} \quad \text{......................... (12)}$$

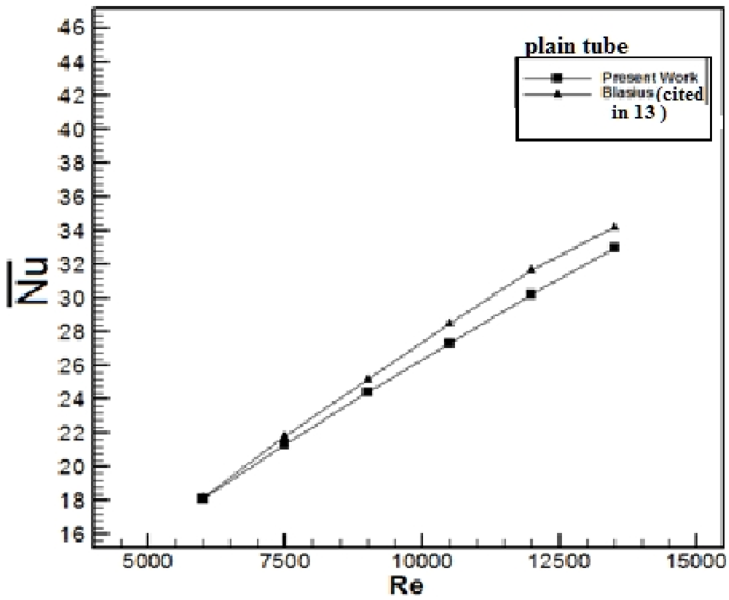

Fig .(4) Verification of Nusselt number of plain tube.

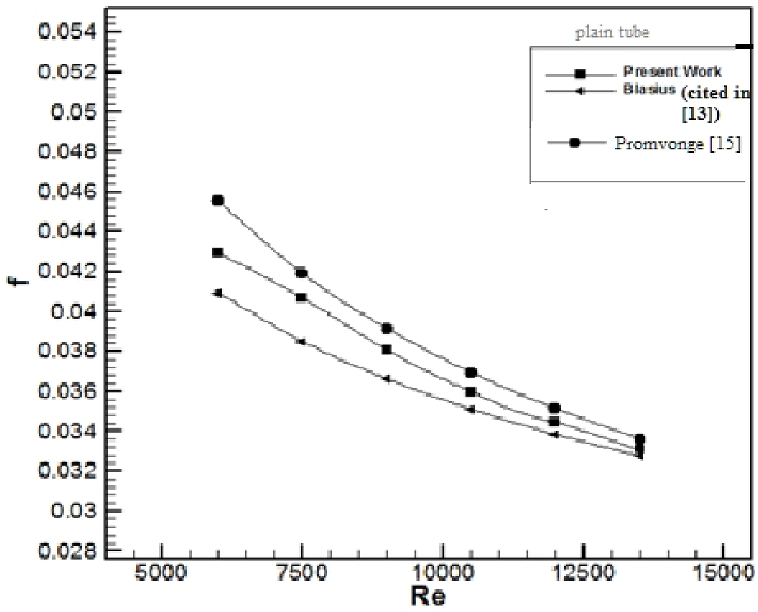

Fig .(5) Verification of friction factor of plain tube.

With the help of experimental data of the tube, the following empirical correlations of Nusselt number and friction factor are suggested for the plain tube and presented in Eqs. (13) and (14) respectively.

They are found to represent the experimental data within ±1 to 7.3% error limits. See **Figures (6&7)**

$$\mathbf{Nu = 0.0939\, Re^{0.6353} Pr^{0.4}} \qquad \text{…………………..… (13)}$$

$$\mathbf{f = 55.8615 Re^{-0.788}} \qquad \text{…………………..… (14)}$$

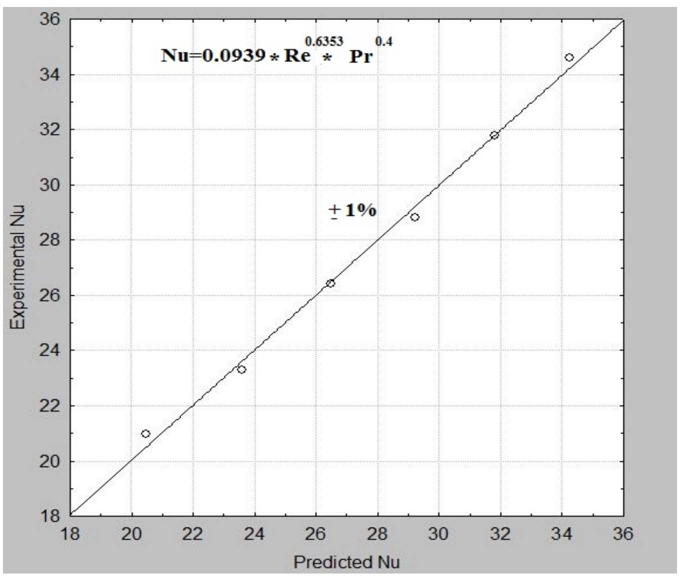

Fig .(6) Correlation predictions of Nusselt number with experimental results.

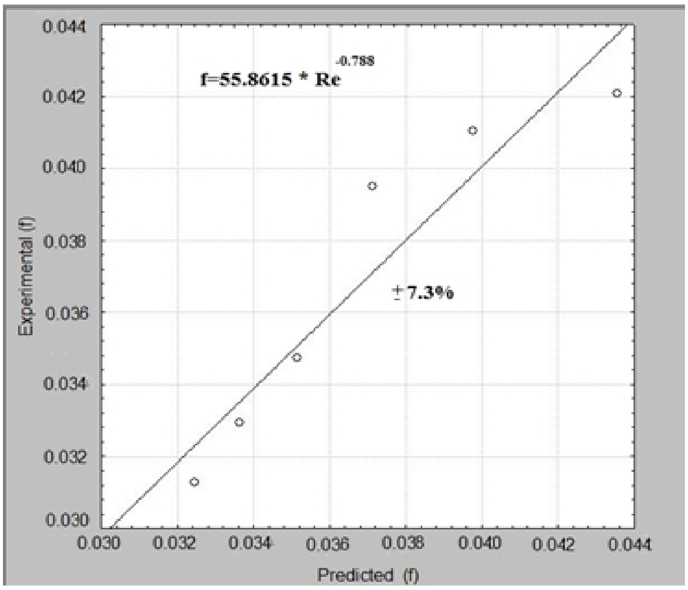

Fig .(7) Correlation predictions of friction factor with experimental results.

4.2. Axial temperature distribution

The temperature distribution along the tube axes for plain tube and other cases, different number of coiled wire distribution, at Re=6000 is shown in (**Figure 8).** The figure show that the values of temperature increase along the tube length. The temperature decrease every time when a set of coiled wire have been inserted into the test tube and the best results was when number of coiled wire is 8 pieces. Air turbulent start to increase when coiled wire inserted, which affect temperature distribution along test tube. Best temperature decrease have been achieved with 8 pieces (S/l =0.57) of coiled wire been inserted which is better than 6 piece (S/l =1.2) of coiled wire, that leads to conclude spacing and specific number of coiled wire can induced high level of turbulent

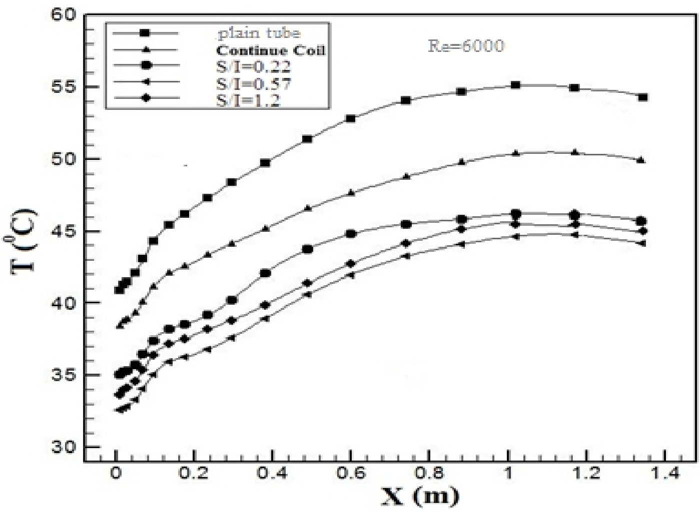

Fig .(8) Temperature distribution along the tube axes for plain wall tube &various number of coiled wire distribution.

4.3. Effect of insert coiled wire

Figure (9), present the heat transfer variation in terms of average Nusselt number for different coiled wire inserted in the tube with Reynolds numbers.

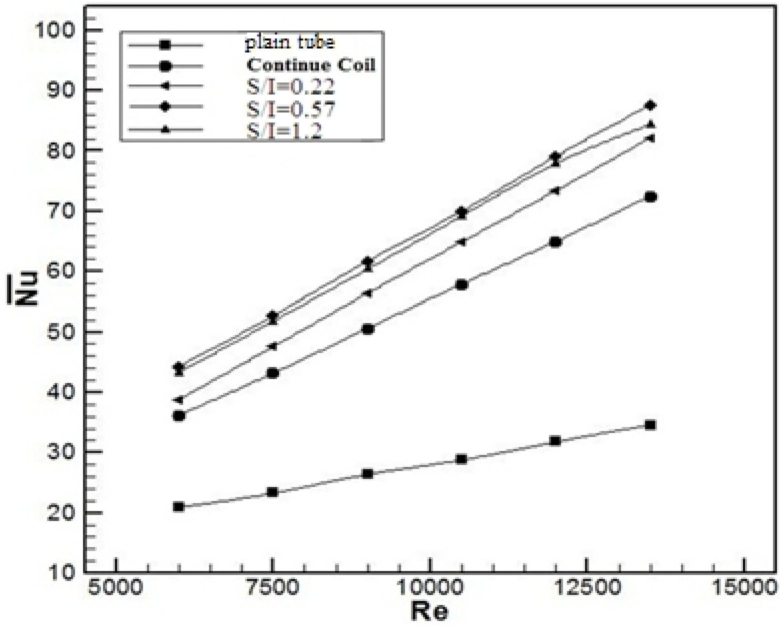

Fig .(9) Nusselt number versus Reynolds Number with various spacing ratios.

In **(Figure 9),** it is visible that the heat transfer increases with increasing Reynolds number for the same conditions, and also the spacing between coiled wires affect the amount of heat transfer improvement winning. It is expected that the turbulator can give rise to the reverse/turbulent flow and boundary layer eruption causing high convective heat transfer and momentum processes. In addition, the slight difference of the heat transfer rate between S/l=0.57 and S/l=1.2 can be observed. This can be attributed to a strong turbulent flow remain for the S/l=0.57. The use of this relative spacing (S/l =0.57) between inserted coiled wire leads to the increase in Nusselt number at 141.3% & 25% in comparison with plain tube and continues coiled wire distribution (S/l=0) respectively of all range Reynolds numbers studied.

Therefore, the highest heat transfer is achieved with the relative spacing (S/l =0.57) between inserted coiled wire turbulator operated in a tube in the range of S/I investigated.

The friction factor variations of using the different included coiled wire spacing (S/l= 0, 0.22, 0.57, 1.2) with Reynolds number between 6000 and 13,500, are illustrated in Fig-10. In this arrangement of coiled wire turbulators, the high turbulence intensity or recirculation can be obtained continuously in the test tube but at different levels depending on the included space ratio between coiled wire. Fig-10, shows that the mean fanning friction factor decreases with increase in Reynolds number. The continues coiled wire(S/l = 0) distribution would provide the lowest reverse/turbulent flow and boundary layer eruption for the surface areas that affect

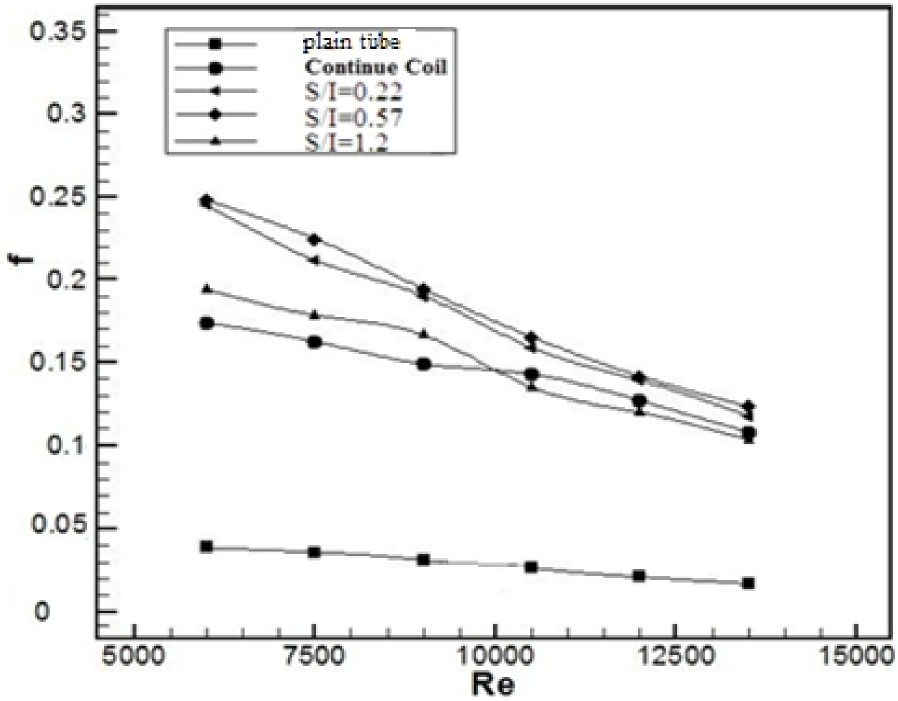

Fig .(10) Friction factor versus Reynolds Number with various spacing ratios.

considerably the pressure loss of fluid flow. Therefore, the large friction loss comes from the higher space ratios (0.57&1.2). The use of the S/l=0.57 results in the augmentation of pressure loss around 33.2% higher than that of the S/l=0. In addition, it can be found that for using coiled wire turbulator with spacing ratios(0, 0.22, 0.57,& 1.2), the improvement of average heat transfer rate is, respectively, about 112%, 119.3%, 141.3%,132% higher than those the plain tube while the friction factor is 300%,392% ,432%, and 322% times of the plain tube. From experimental results above, the Nusselt number and friction factor for plain tube and coiled wire turbulators with different spacing ratio are correlated as in equations shown in **Tables (1), Figures.(11 – 14)** show the relation between the present experimental data with the predicted data obtained from the proposed correlations, which is deviated within ±7% and ±2.3%, for the friction factor and Nusselt number, respectively, so the empirical correlations are valid for the experimental conditions of $6000 \leq Re \geq 13{,}500$, $Pr = 0.7$ and different space ratio between coiled wire.

Thus, the correlation could predict the values of Nusselt number and friction factor satisfactorily in the range of parameters studied.

Table .(1) Correlations of Nusselt number and friction factor for plain tube and coiled wire turbulators with different spacing ratio.

No.	S/l	Correlation of Nusselt number	Correlation of Friction factor
1	Plain tube	$Nu= 0.0939.Re^{0.6353}.Pr^{0.4}.Y^{0}$	$f= 55.8615.Re^{-0.788}$
2	0	$Nu= 0.02224.Re^{0.965}.Pr^{0.4}.Y^{0}$	$f= 14.728.Re^{-0.5068}$
3	0.22	$Nu=0.02322.Re^{0.8678}.Pr^{0.4}.Y^{0.0355}$	$f= 21.424.Re^{-0.884}.Y^{-0.0884}$
4	0.57	$Nu=0.02322.Re^{0.8678}.Pr^{0.4}.Y^{0.0355}$	$f= 21.424.Re^{-0.884}.Y^{-0.0884}$
5	1.2	$Nu=0.02322.Re^{0.8678}.Pr^{0.4}.Y^{0.0355}$	$f= 21.424.Re^{-0.884}.Y^{-0.0884}$

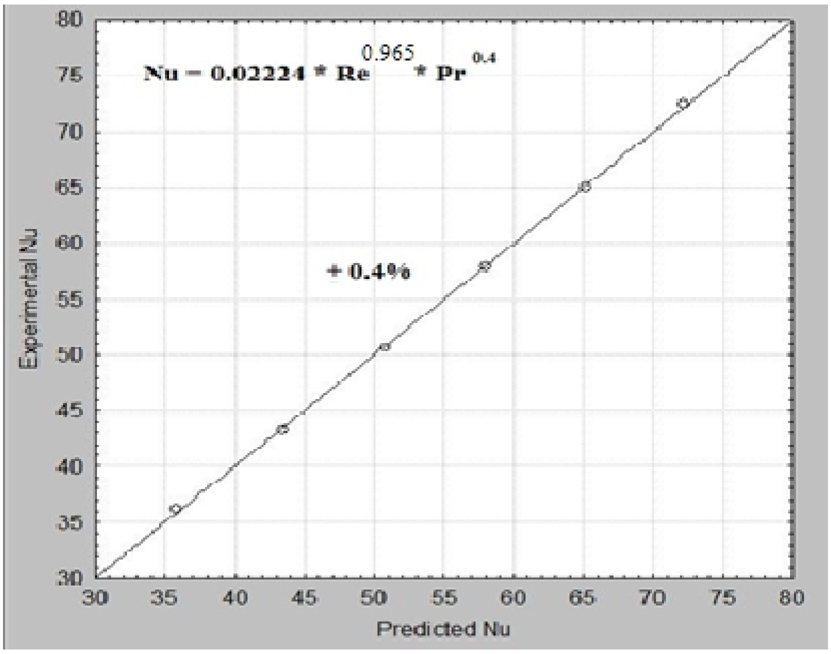

Fig .(11) Correlation predictions of Nusselt number with experimental results for S/l=0.

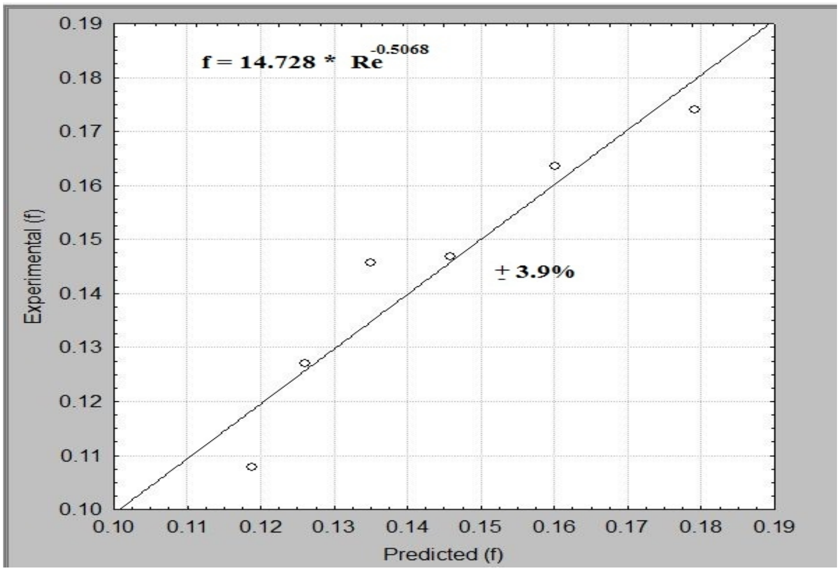

Fig .(12) Correlation predictions of friction factor with experimental results for S/l=0.

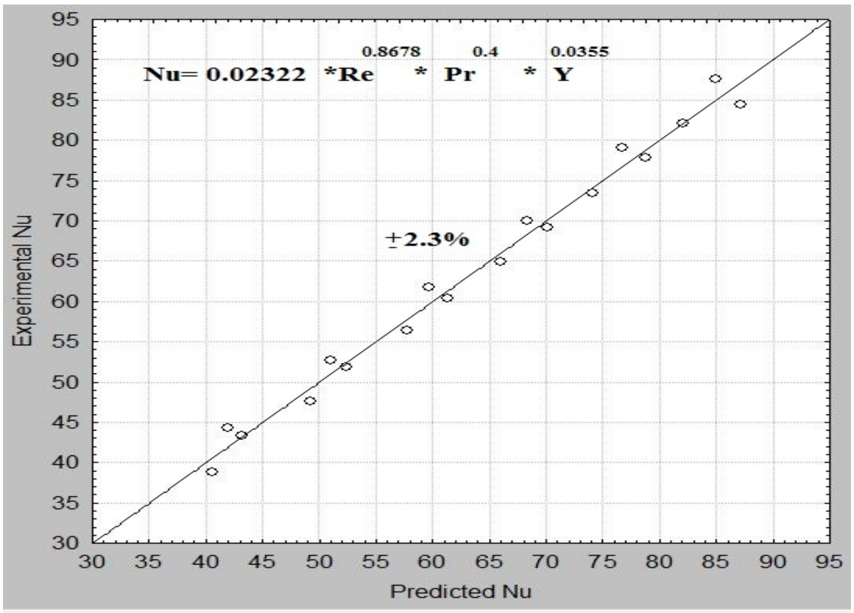

Fig .(13) Correlation predictions of Nusselt number with experimental results for S/l= (0.22, 0.57&1.2)

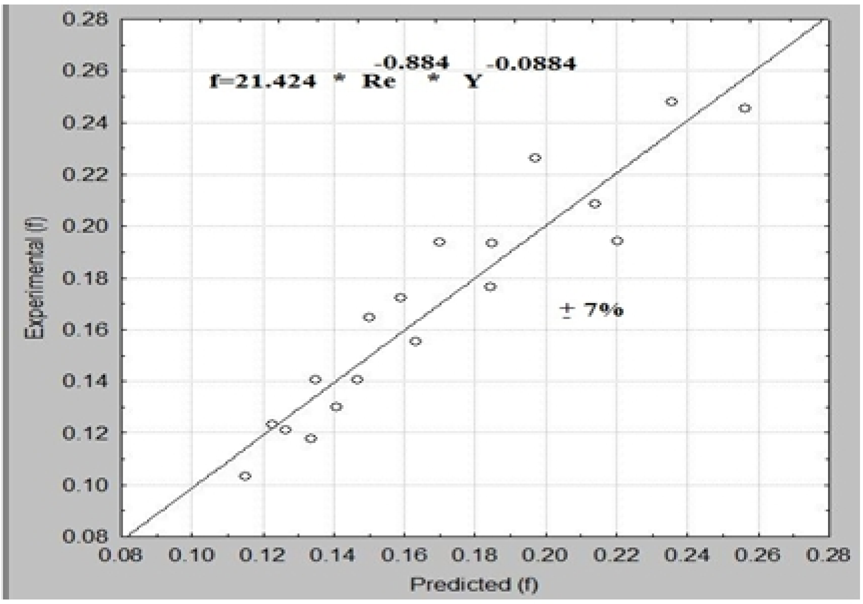

Figure-14: Correlation predictions of friction factor with experimental results for S/I= (0.22, 0.57&1.2).

4.4. Performance evaluation:

In order to asses thermal performance of using the coiled wire element, the enhancement efficiency (η) for a constant pumping power comparison from Eq. (10) is introduced and presented in **Figure (15)** for various S/I values. In the figure, it is worth noting that the enhancement efficiency tends to increase with increasing Reynolds number for all cases.

The highest S/I (6 coiled wire pieces) provide the highest enhancement efficiency especially at Reynolds number value (=10500). Enhancement efficiencies are found to be around 151%, 145%, 134% &128% more than plain tube efficiency for 6,8,10 & 12 pieces of coil respectively, as shown in **Figure (16).**

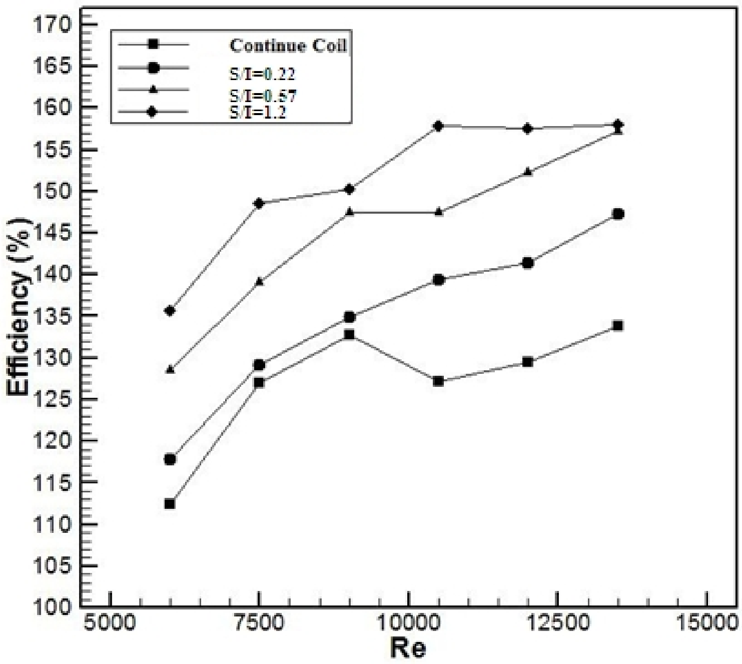

Fig .(15) Enhancement efficiency versus Reynolds number with various relative spacing.

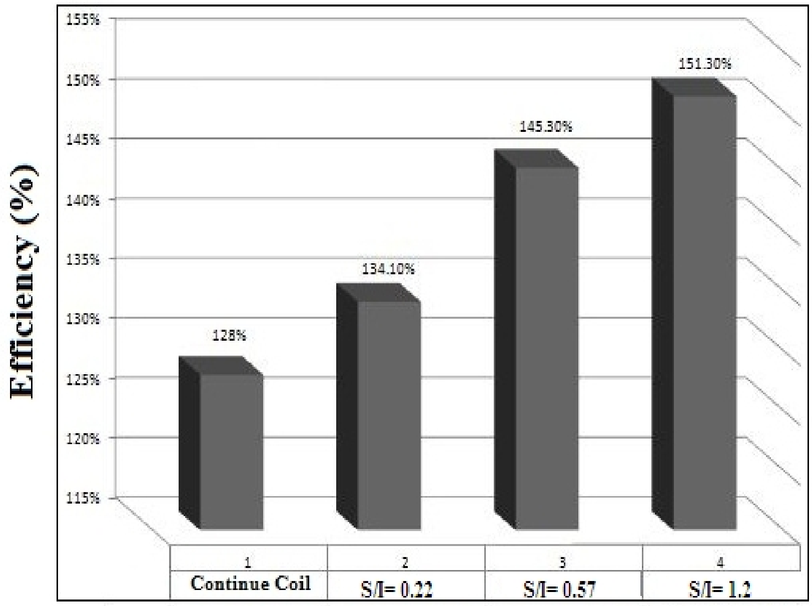

Fig .(16) Percentage of enhancement of efficiency of Turbulators

5. Conclusions:

- Heat transfer increases with increasing Reynolds number for the same conditions (about 90% between lower and upper limit of study range of Reynolds number), and also the spacing between coiled wires affect the amount of heat transfer improvement winning.
- The highest heat transfer is achieved with the relative spacing (S/l =0.57) between inserted coiled wire turbulator operated in a tube in Nusselt number at 141.3% & 15% in comparison with plane tube and continues coiled wire distribution (S/l=0) respectively of all range Reynolds numbers studied.
- Using coiled wire turbulator with spacing ratios(0, 0.22, 0.57,& 1.2), the improvement of average heat transfer rate is, respectively, about 112%, 119.3%, 141.3%,132% higher than those the plain tube while the friction factor is 300%,392% ,432%, and 322% times of the plain tube.
- Lowest reverse/turbulent flow come from continues coiled wire distribution (S/l = 0). So, the large friction loss comes from the higher space ratios (0.57&1.2). The use of the S/l=0.57 results in the augmentation of pressure loss around 33.2% higher than that of the S/l=0.

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