

## HEAT TRANSFER CHARACTERISTICS IN A DOUBLE-PIPE HEAT EXCHANGER EQUIPPED WITH COILED CIRCULAR WIRES

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*The effects of insulating wires (acting as turbulators only), with circular cross section of 2mm diameter, forming a coil of different pitches on the heat transfer rates are experimentally investigated. The investigation is performed for turbulent water stream in a double-pipe heat exchanger for both parallel and counter flows (for comparison) with cold water in the shell side. The experiments are performed for flows with Reynolds numbers ranging from 4,000 to 14,000. Three different spring coiled wire pitch values are used. The experimental results reveal that the use of coiled circular wire turbulators leads to a considerable increase in heat transfer over those of a smooth wall tube. The mean Nusselt number increases with the rise of Reynolds number and the increasing of pitch for both parallel and counter flows. The convective heat transfer coefficient for the turbulent flow was found to increase with turbulators for all coiled wire pitch values with the highest enhancement of about 450% for the counter flow while it was 400% for the parallel one. Correlations for mean relative Nusselt numbers and coiled wire pitch are provided.*

**KEYWORDS:** Heat transfer enhancement; Coiled wire; Turbulators; turbulent pipe flow; heat exchanger

### NOMENCLATURE

<i>A</i>	heat transfer area (m <sup>2</sup> )	<i>Greek</i>	
<i>c</i>	specific heat capacity (kJ/kg K)	<i>Symbols</i>	
<i>D</i>	diameter (m)	$\Delta T_m$	logarithmic mean temperature difference (K)
<i>F</i>	friction factor	$\mu$	dynamic viscosity (N s/m <sup>2</sup> )
<i>h</i>	convective heat transfer (W/m <sup>2</sup> K)	$\nu$	kinematic viscosity (m <sup>2</sup> /s)
<i>k</i>	thermal conductivity (W/m K)	$\epsilon$	heat exchanger effectiveness
<i>L</i>	tube length (mm)	<i>Subscripts</i>	
<i>m</i>	mass flow rate (kg/s)	<i>c</i>	cold fluid
<i>Nu</i>	Nusselt number	<i>H</i>	hydraulic
<i>p</i>	pitch of coiled wire (mm)	<i>h</i>	hot fluid
<i>P</i>	dimensionless pitch ( $p/D_H$ )	<i>i</i>	inlet/inner
<i>Pr</i>	Prandtl number ( $\mu C_p/k$ )	<i>o</i>	outlet/outer/un-pulsated
<i>Q</i>	heat transfer rate (W)	<i>s</i>	steady
<i>Re</i>	Reynolds number	<i>t</i>	total
<i>T</i>	temperature (°C)		
<i>U</i>	overall heat transfer coefficient (W/m <sup>2</sup> K) average axial		
<i>V</i>	velocity (m/s)		

## 1. INTRODUCTION

Heat transfer enhancement may be achieved by numerous techniques. These techniques can be classified into three groups: passive, active and compound techniques, Bergles [1]. The passive techniques such as swirl flow devices, treated surfaces, rough surfaces, extended surfaces, coiled tubes, surface tension devices and additives to fluids do not require direct application of external power. Whereas the active techniques, such as mechanical aids, surface vibration, fluid vibration, electrostatic fields, suction or injection and jet impingement, require an external activator/power supply to bring about the enhancement. The compound techniques, are such as rough surface with a twisted tape swirl flow device or two or more of the active or passive techniques utilized simultaneously to produce an enhancement that is much higher than that when these techniques operate separately. A heat exchanger is a device facilitating the convective heat transfer of the fluid and is extensively used in many engineering applications, such as thermal power plants, chemical processing plants, air conditioning equipment, refrigerators, and radiators of automobiles. One method for heat transfer augmentation which is a passive method that does not require external power is the use of swirl flow devices (twisted tape or coiled wire, sometimes called "turbulators"). Turbulators are inserted into the flow to provide redevelopment of the boundary layer, to cause enhancement of heat transfer by increasing the turbulence or by rapid fluid mixing. For decades, many of the wire coil devices employed for augmentation of laminar or turbulent flow heat transfer have been reported and discussed. However, few researches for coiled wire inserts have been found in comparison with those for twisted tape inserts. This has been noticed from the work of Shoji et al. [2] or Garcia et al. [3]. Coiled wires are of practical interest and, therefore, their data are required to extend the use of this technique. Some investigations were conducted to determine the effect of the coiled wire on the heat transfer and friction factor for a long time [4-6]. Enhancement of heat transfer by using several coiled wire inserts, based on exergy analysis, was investigated by Prasad and Shen [7]. Correlations for friction factor and heat transfer coefficient for turbulent flow in internally heat transfer augmented tubes were proposed by Ravigururajan and Bergles [8]. Agrawal et al. [9] experimentally investigated heat transfer augmentation by using coiled wire inserts during forced convection condensation of R-22 inside a horizontal tube. Inaba and Ozaki [10] investigated that the turbulent flow induced by using a coiled wire enhances heat transfer even downstream of the coiled wire. Kim et al. [11] investigated the flow pattern, void fraction and slug rise velocity on counter current two phase flow in a vertical round tube with coiled wire inserts. They observed that the slug rise velocity and void fraction in a vertical round tube is higher for a coiled wire insert than that in a smooth tube.

A comparison of the thermal and hydraulic performances between twisted tape inserts and coiled wire inserts was introduced by Wang and Sunden [12] for both laminar and turbulent flow regions. They found that the coiled wire performs effectively in enhancing heat transfer in a turbulent flow region, whereas the twisted tape yields a poorer overall efficiency. Rahai et al. [13] experimentally investigated the influences of wire coil pitch spacing on the mixing enhancement of a turbulent jet from a Bunsen burner. Heat transfer enhancement of flowing water in a tube with flow drag reduction additives by inserting wire coils was presented by Inaba and Haruki [14].

Rahai and Wong [15] investigated turbulent jets from round tubes with coil inserts. Yakut and Sahin [16] examined the heat transfer and friction loss by placing coiled wire turbulators in a tube in addition to the shedding frequencies and amplitudes of vortices produced by coiled wire turbulators, including a coupling of flow and acoustic structures. The conjugate heat transfer and thermal stress in a tube with coiled wire inserted under uniform and constant wall heat flux was numerically investigated by Ozceyhan [17], while the heat transfer and pressure drop characteristics in a horizontal double pipe with coiled wire inserts were studied experimentally by Naphon [18]. Gunes et al. [19] investigated the heat transfer, friction factor and thermal performance characteristics in a circular tube fitted with equilateral triangular cross section wire coil. Recently, the use of twisted tape together with typical wire coil (with uniform pitch length or ratio) was first reported by Promvonge [20]. The results revealed that the combined devices gave higher heat transfer rate, friction factor and also thermal enhancement factor than the wire coil or twisted tape alone, indicating the synergy effect of both devices. The experimental results also showed that the thermal enhancement factor increased with decreasing coil spring pitch ratios while in the present study, the coil spring is made of insulated wires. Influence of combined non-uniform wire coil and twisted tape inserts on thermal performance characteristics was studied by Eiamsa-ard et al. [21]. The experiments were conducted in a turbulent flow regime with Reynolds numbers ranging from 4600 to 20,000 using air as the test fluid. Compared to each enhancement device, the heat transfer rate is further augmented by the compound devices between the twisted tape and constant/periodically varying wire coil pitch ratio. Enhancement of heat transfer using varying width twisted tape inserts was studied experimentally by Naga Saradal et al. [22]. Experiments were carried out for plain tube with/without twisted tape insert at constant wall heat flux and different mass flow rates. The twisted tapes were of three different twist ratios (3, 4 and 5) each with five different widths (26-full width, 22, 18, 14 and 10 mm) respectively. The Reynolds number varied from 6000 to 13500. Both heat transfer coefficient and pressure drop were calculated and the results were compared with those of plain tube. It was found that the enhancement of heat transfer with twisted tape inserts as compared to plain tube varied from 36 to 48% for full width (26mm) and 33 to 39% for reduced width (22 mm) inserts.

In the above literature review, most studies are mainly focused on the effects of the spring coil pitch and coiled wire thickness on the heat transfer and friction characteristics of air or water flows in tubes. The investigation on different cross sectional shapes of wires forming a coil has rarely been reported, apart from circular cross section wires. Thus, the main aim of the present work is to investigate the effects of insulating spring-coiled wires, with circular cross section of 2mm diameter, forming a coil of different pitch values used as turbulators (not as fins), on the heat transfer rates. The investigation is performed for turbulent water stream in a double-pipe heat exchanger for both parallel and counter flows with cold water in the shell side. The experiments are performed for flows with Reynolds numbers ranging from 4,000 to 14,000. Three different spring coiled wire pitch values are used.

## 2. TEST RIG AND INSTRUMENTATION

An experimental facility was designed and constructed to investigate the heat transfer characteristics of the turbulent water flow through a concentric tube heat exchanger. The test rig, as shown in Fig. 1, consists mainly of an insulated coiled wire to act as turbulators not as fins, temperatures measuring devices and a horizontal water-to-water concentric tubes heat exchanger with parallel or counter water flows. To minimize the heat losses in the system, the hot water is fed through the inner pipe, with cooling water in the outer annulus. The outer surface of the heat exchanger was well insulated to minimize convective heat loss to the surroundings, and necessary precautions were taken to prevent leakages from the system. A heat exchanger has 1000 mm in length, outer tube of 40 mm outer diameter (1.5 mm wall thickness), inner tube of 12 mm outer diameter (1 mm wall thickness), and 0.037 m<sup>2</sup> heat transfer area. The outer and inner tubes of heat exchanger were made of brass ( $k = 109$  W/mK). Six type-K thermocouples are installed through holes, with the aid of epoxy adhesive for preventing the leakage, in both the inside and outside tubes, to measure the fluid temperatures accurately at the middle and end caps (for taking the average temperatures in calculating the water properties) of the heat exchanger. Both the inlet and outlet temperatures of the cold and hot water were measured by type K thermocouples, calibrated within  $\pm 0.2^\circ\text{C}$  deviation by thermostat before being used. The coiled wire was made of a steel wire, encased in insulation of very thin ductile tubes, with circular cross sections. The cross section of wire turbulators of 2 mm wire diameter was imposed. The coiled wires of different spring pitch values ( $p = 6, 12$  and  $20$  mm) were coiled adjacent to the outer surface of the inner tube. Heat transfer coefficient ( $h_o$ ) is determined from the overall heat transfer coefficient and then Nusselt number can be calculated, Pethkool *et al.* [23] and Meter [24]. The cold water entering the system, through outer annulus, was drawn by a 0.56 KW pump from cold water supply tank, and passed to the drain out at downstream. Control valves are incorporated in each of the two streams to regulate the flow. The flow rates are measured using independent Rota-meters installed in each line. A hot storage tank (50 liter) equipped with an immersion type heater and adjustable temperature controller to maintain a temperature within  $\pm 0.5^\circ\text{C}$ , was used. Circulation of the hot water to the heat exchanger is provided by a 0.56 KW pump and water returns to the storage tank via a baffle arrangement to ensure adequate mixing. The temperature of the hot water was kept at about  $65^\circ\text{C} \pm 0.5^\circ\text{C}$ . The heat losses are the losses of heat through the insulation to the atmosphere and the axial conduction heat losses due to tube thickness. The major heat losses are assumed to be through insulation only with neglecting other losses as concluded by Baughn *et al.* [25] and Incropera and Dewitt [26].

## 3. EXPERIMENTAL PROCEDURE

An experimental program was designed to study the effect of roughness by using an insulated coiled wire with different pitches on the heat transfer through a concentric tube heat exchanger for the turbulent water flow. The studied values of Reynolds numbers of cold water are 4390, 6585, 8780, 11000, and 13160. These values correspond to mass flow rates of 0.0663, 0.0994, 0.1326, 0.1657 and 0.1988 kg/s, where the mass flow rate of the hot water was kept at 0.1144 kg/s. The inlet hot water

temperature was kept at about 65 °C. The outer surface of the test section was insulated to minimize heat losses, and necessary precautions were taken to prevent leakages in the system. The coiled wire was wrapped around the inner tube of the heat exchanger tube. Three different coil pitch values were used, (pitch value = 6, 12, and 20 mm). Reynolds number is defined as  $(4m_c/\pi D_H \mu_{ci})$  based on inlet cold-water flow conditions. Forty experiments were carried out at steady state conditions for parallel and counter flows. The value of the heat transfer coefficient of flow over the coiled wire, through the heat exchanger, was normalized with the corresponding un-coiled one. Since the temperatures of the hot and cold water vary over the length of the tubes, the temperature difference,  $\Delta T = T_h - T_c$ , is not constant over the length.

To account for the temperatures variations, a log mean temperature difference

$$\Delta T_{lm} = (\Delta T_{in} - \Delta T_{out}) / (\ln (\Delta T_{in}/\Delta T_{out})), \text{ is used, [26-30].}$$

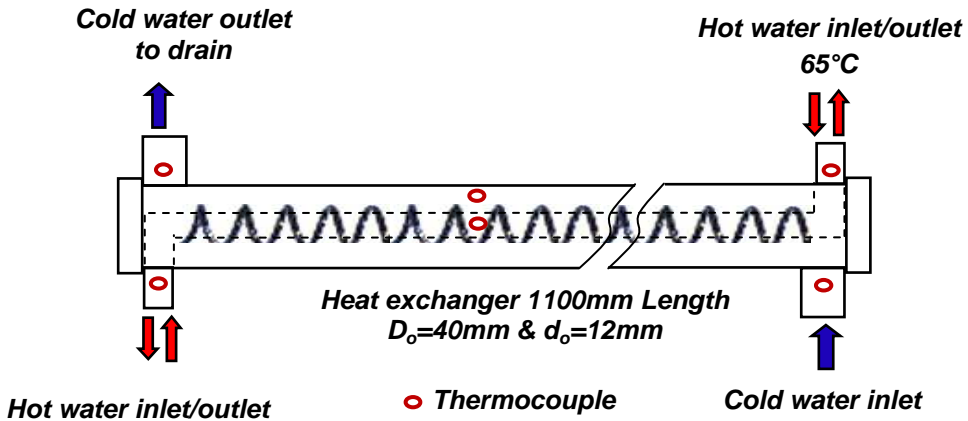


Fig. 1, Simplified schematic diagram of the apparatus

### 3.1 Theoretical analysis

The heat given by the hot fluid (i.e. water) at any Reynolds number is,:

$$Q = m_h C_{ph}(T_{hi} - T_{ho}) = U_i A_i \Delta T_{lm} \tag{1}$$

While the heat transferred to the cold fluid, (i.e. water) is:

$$Q = m_c C_{pc}(T_{ci} - T_{co}) = U_o A_o \Delta T_{lm} \tag{2}$$

As usual, this heat may be expressed in terms of a heat transfer coefficient and tube logarithmic mean temperature difference  $\Delta T_{lm}$ :

$$Q_t = UA \Delta T_{lm} \tag{3}$$

In the experiments, the tube-wall temperature was not measured, with negligible losses to surrounding air from the cold water, by equalizing the energy loss of hot fluid and the energy received by the cold fluid, convective and overall heat transfer coefficients were deduced and Nusselt numbers were acquired as follows, [26-30]:

$$\frac{1}{UA} = \frac{1}{h_o A_o} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_i A_i} \tag{4}$$

Where,  $h_i$  and  $h_o$  are heat transfer coefficients for hot and cold water respectively. The areas,  $A_i$  and  $A_o$  are the inner and the outer surface areas of the inner

tube. The diameters,  $D_i$  and  $D_o$  are the inner and the outer tube diameters of the inner tube.  $U$  is the overall heat transfer coefficient,  $k$  is the thermal conductivity of the tube material and  $L$  is the total tube length. For fully developed, turbulent flow in tubes where the Reynolds number is between 2300 and  $5 \times 10^6$  and the Prandtl number is between 0.5 and 2000, an empirical correlation to determine  $h_i$  proposed by Gnielinski, V. [28], is widely used and hence one can get  $h_o$  &  $Nu_{D_o}$ .

$$Nu_{D_i} = \frac{h_i D_i}{k} = \frac{(F/8)(Re_{D_i} - 1000)Pr}{1 + 12.7(F/8)^{1/2}(Pr^{2/3} - 1)} \tag{5}$$

The tube-side heat transfer coefficient could be evaluated from Gnielinski correlation,

$$h_i = \frac{(F/8)(Re_{D_i} - 1000)Pr}{1 + 12.7(F/8)^{1/2}(Pr^{2/3} - 1)} \times \frac{k}{D_i} \tag{6}$$

Where, for smooth tubes, the friction factor is given by:

$$F = [0.79 \ln(Re_{D_i}) - 1.64]^{-2} \tag{7}$$

For the hot and cold fluids, the Reynolds numbers are:

$$Re_D = VD_H/\nu \tag{8}$$

Heat exchanger effectiveness,  $\epsilon$ , is defined as,

$$\epsilon = m_h C_h (T_{hi} - T_{ho}) / m C_{min} (T_{hi} - T_{ci}) = m_c C_c (T_{co} - T_{ci}) / (m C_{min}) (T_{hi} - T_{ci}) \tag{9}$$

Where  $(mC)_{min}$  = minimum of either  $m_h C_h$  or  $m_c C_c$ .

Thus, the convective heat transfer coefficient  $h_i$  is determined from equation (6 and 7) and used in equation (4) to find the convective coefficient,  $h_o$ . The enhancement in heat transfer was then calculated as the ratio of average Nusselt number in case of coiled wire ( $Nu_{mp}$ ), coiled around the internal tube of the heat exchanger, relative to the smooth tube with no wire ( $Nu_{mo}$ ). All fluid properties were taken at the average fluid temperature. A more precise method of estimating uncertainty in experimental results has been presented by Kline and McClintock, which is described in Holman [29]. The method is based on careful specification of the uncertainties in the various primary experimental measurements. Assume that the result dependant variable ( $R$ ) is a given function of the independent variables  $x_1, x_2, x_3, \dots, x_n$ . Thus:  $R = R(x_1, x_2, x_3, \dots, x_n)$ . Let  $u_R$  is the uncertainty in the result and  $u_1, u_2, u_3, \dots, u_n$  are the uncertainties in the independent variables. The uncertainty in the result is given as:

$$u_R = \left[ \left( \frac{\partial R}{\partial x_1} u_1 \right)^2 + \left( \frac{\partial R}{\partial x_2} u_2 \right)^2 + \dots + \left( \frac{\partial R}{\partial x_n} u_n \right)^2 \right]^{1/2} \tag{10}$$

The following table (3.1) summarizes the calculated values of the uncertainty of the measured quantities.

**Table 3.1, values of the uncertainty of the measured quantities**

Parameter	Absolute Uncertainty	Relative Uncertainty
Mass flow rate		± 1.2 %
Reynolds number		± 1.2 %
Mean flow temperature	± 0.5 °C	
Mean Nusselt number		± 4.5 %

### 4. RESULTS AND DISCUSSIONS

The main target of the present work is to investigate experimentally the influence of inserted coiled wire with different pitch values on heat transfer in a water-to-water concentric heat exchanger with counter and parallel flows. With the values obtained from the experimental data (40 experiments), the changes in average Nusselt number values against Reynolds number values were drawn for three different insulated wire pitches for both parallel and counter flows, as shown in the Figs. 2 and 3. All curves of the average Nusselt number variation against Reynolds number, for both parallel and counter flows, show nearly the same trend. Nusselt number increases with the increase of the value of wire pitch and Reynolds number. The coiled wire turbulators yield a considerable heat transfer enhancement with a similar trend in comparison with the smooth tube (no wire), and the Nusselt number for the coils increases by increasing Reynolds number. In the parallel flow (Fig. 2), for the largest pitch value (20 mm), the increase in heat transfer rate was up to 400% compared with no wire results depending on Reynolds number value. In the counter flow (Fig. 3), largest pitch value (20 mm), the increase in heat transfer rate was up to 450% compared with no wire results depending on Reynolds number value. A comparison between the present work and the results of Promvonge [20], for the average Nusselt number versus Reynolds number at different coiled wire pitch values are presented to show the influence of coiled wire used as a turbulators on the heat transfer rate, Figs. 4 and 5. The experimental results of Promvonge [20] reveal that the use of coiled square wire or coiled round wire turbulators leads to a considerable increase in heat transfer up to 200%.

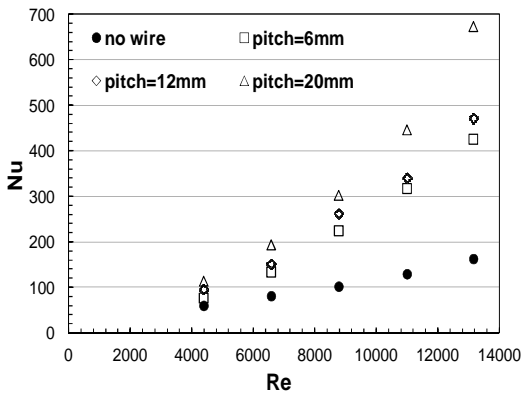


Fig. 2 Average Nusselt number in the **parallel flow** versus Reynolds number for different pitch values

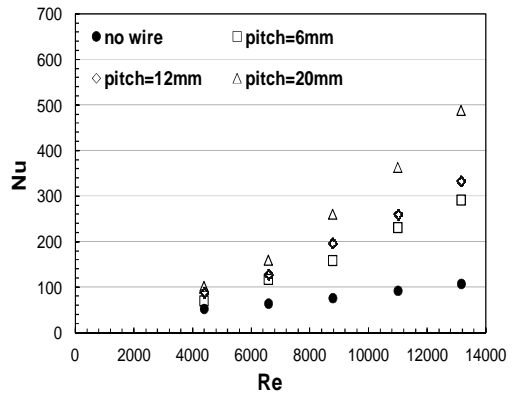


Fig. 3 Average Nusselt number in the **counter flow** versus Reynolds number for different pitch values

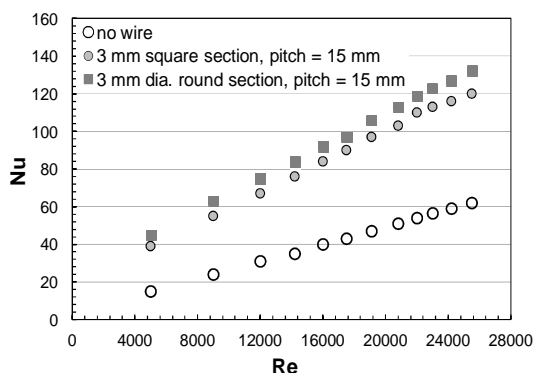


Fig. 4 Variation of Nusselt number With Reynolds number for circular and square Wire coils, data of Promvong P., [20]

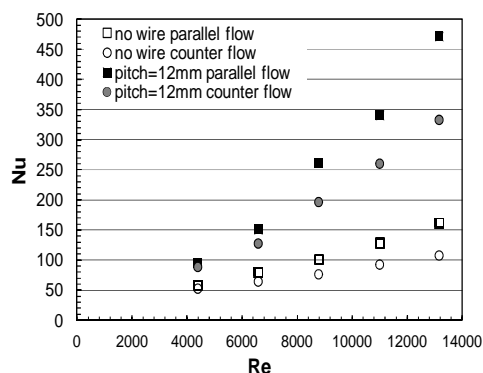


Fig. 5 Variation of Nusselt number With Reynolds number for circular wire coils of present work

The influence of both wire pitch and Reynolds number on the Nusselt number, for both parallel and counter flows, is shown in Figs. (6-9). Figures (6 & 7) show the variation of the average relative Nusselt number ( $Nu_{nr} = Nu_{mp}/Nu_{mo}$ ), versus Reynolds number for various coiled wire pitch values. Where  $Nu_{nr}$  is the ratio of average Nusselt number in case of coiled wire ( $Nu_{mp}$ ), inserted around the internal tube of the heat exchanger, relative to that of the smooth tube with no wire ( $Nu_{mo}$ ). Figures (8 & 9) show the average relative Nusselt number ( $Nu_{mp}/Nu_{mo}$ ), versus coiled wire pitch for various Reynolds numbers. The results of heat transfer show that, more enhancements in relative Nusselt number are obtained for the counter flow than that obtained for the parallel flow, although the values of average Nusselt number of parallel flow have higher values than that obtained for the counter flow, Figs. 2 and 3. More enhancements in heat transfer rates are obtained at high Reynolds number and high wire pitch values. This is because the coiled wire turbulators interrupt the development of the boundary layer of the fluid flow and increase the degree of flow turbulence. Although the turbulence intensity and the flow path obtained from the lower pitch are greater and longer than these from the higher one, the area exposed to heat transfer becomes lower because of the insulating coiled wire. So, the heat transfer rates become lower with lower wire pitch values. The convective heat transfer coefficient for the turbulent flow was found to increase with turbulators for all coiled wire pitch values with the highest enhancement of about 450% for the counter flow while it was 400% for the parallel one. In the counter flow, the heat transfer rates are somewhat greater than that in the parallel flow [30 and 31]. This may be explained by the fact that the average difference in temperature of the fluids in the counter-flow is somewhat greater than that in the parallel flow. Because the temperature effectiveness of parallel flow is limited compared to counter-flow, the thermal capacity of the counter heat exchanger can be higher than that of the parallel flow heat exchanger. In addition, the difference between parallel and counter flow heat transfer results is due probably to later development of a thermal boundary layer for parallel flow. The influence of the turbulators for all coiled wire pitch values on heat transfer is more effective in the counter flow than the parallel flow heat exchangers, [30 and 31].



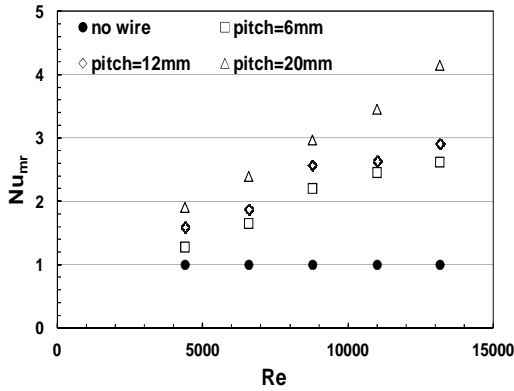


Fig. 6 Average Nusselt number ratio in the **parallel flow** versus Reynolds number for different pitch values

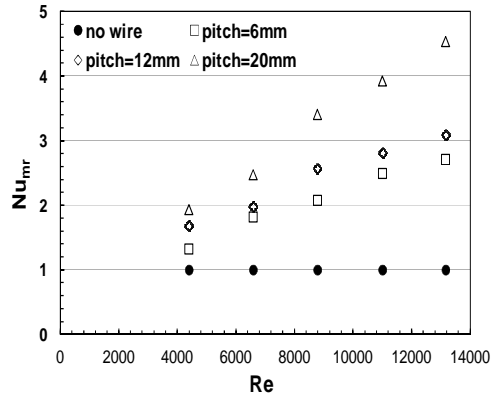


Fig. 7 Average Nusselt number ratio in the **counter flow** versus Reynolds number for different pitch values

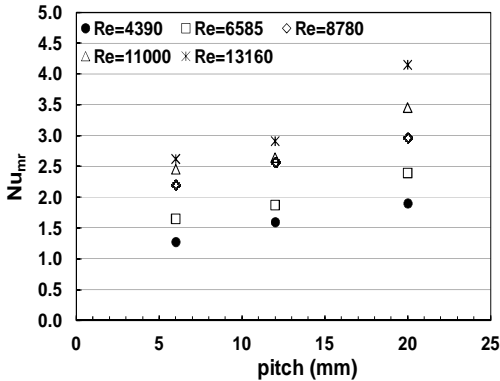


Fig. 8 Average Nusselt number ratio in the **parallel flow** versus coil pitch for different Reynolds number

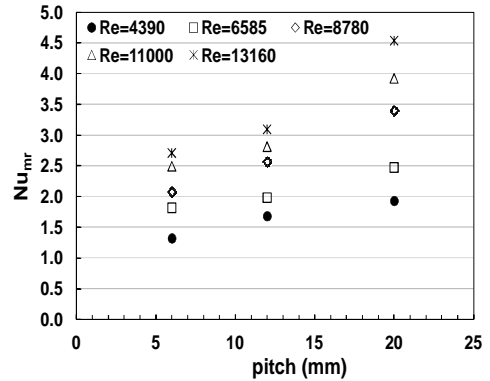


Fig. 9 Average Nusselt number ratio in the **counter flow** versus coil pitch for different Reynolds number

Figures 10 and 11 show a variation of relative effectiveness of the heat exchanger versus Reynolds number at different coiled wire pitch values, for both parallel and counter flows respectively. Similar trends were obtained for the relative mean Nusselt number. The figures show that more enhancements (up to 200%) in the effectiveness were obtained for the counter flow. At very large pitch values, the relative effectiveness will decrease and the coiled wire will have no effect on heat transfer.

### 4.1 Numerical Correlations for the Results

Correlations for the turbulent flow ( $Re=4000-14,000$ ) with different coiled wire pitch values, for predicting the relative Nusselt number and relative effectiveness through a double-pipe heat exchanger, were derived and shown in tables 4.1. The correlations are valid with a certain error for  $p = 6-20$  mm and  $Re = 4,000$  to 14,000. The maximum

standard error of  $Nu_{nr}$  for parallel and counter flows is about 9.6% and 7.4%, respectively. The maximum standard error of the effectiveness for parallel and counter flows is about 5.6% and 6.5% respectively. Figures 12 and 13 show comparison between the experimental results and the correlations of average Nusselt number ratio, and a reasonable agreement was found as shown in these figures. Figures 14 and 15 show comparison between the experimental results and the correlations of relative effectiveness and a reasonable agreement was found as shown in these figures.

**Table 4.1, correlation equations of both parallel and counter flows**  
( $Nu_{nr}$  &  $\epsilon_{mr}$ )

<b>Parallel flow</b>	$Nu_{nr} = 0.00224 \times Re^{0.677} \times P^{0.353}$	Standard error = 9.6%
<b>Counter flow</b>	$Nu_{nr} = 1.008E^{-03} \times Re^{0.7592} \times P^{0.3904}$	Standard error = 7.4 %
<b>Parallel flow</b>	$\epsilon_{mr} = 0.20487 \times Re^{0.1982} \times P^{0.0912}$	Standard error = 5.6%
<b>Counter flow</b>	$\epsilon_{mr} = 0.11964 \times Re^{0.26885} \times P^{0.081247}$	Standard error = 6.5%

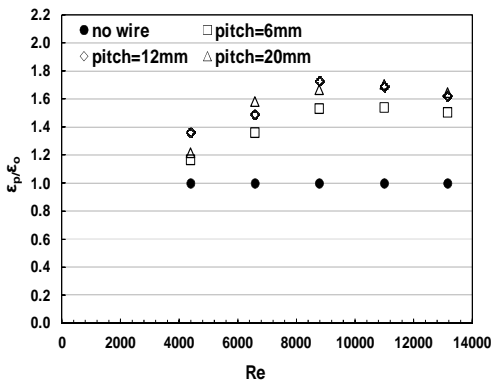


Fig. 10 Effectiveness ratio in the parallel Flow versus Reynolds number for different coil pitches

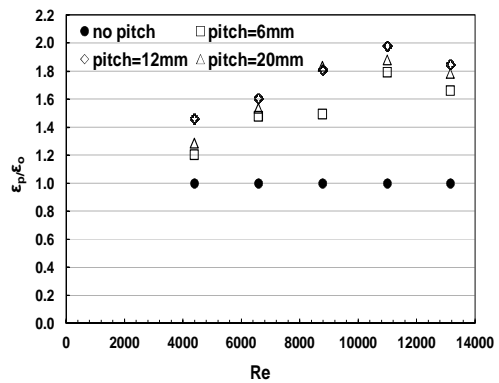


Fig. 11 Effectiveness ratio in the counter Flow versus Reynolds number for different coil pitches

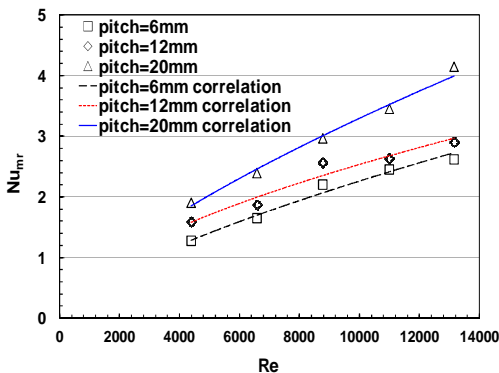


Fig. (12) Correlation of results of  $Nu_{nr}$  for parallel flow

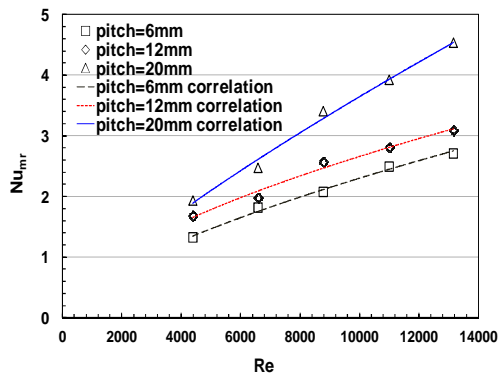


Fig. (13) Correlation of results of  $Nu_{nr}$  for Counter flow

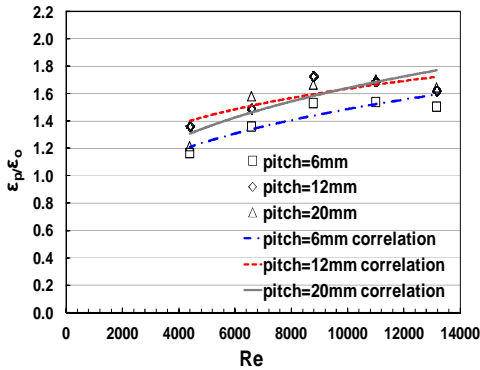


Fig. (14) Correlation of results of  $\epsilon_{pr}/\epsilon_o$  for parallel flow

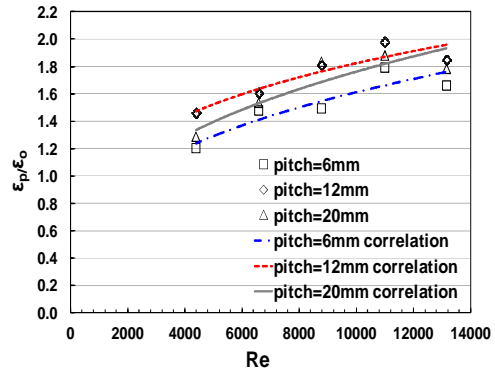


Fig. (15) Correlation of results of  $\epsilon_{pr}/\epsilon_o$  for Counter flow

## CONCLUSIONS

An experimental study has been performed to investigate heat transfer characteristics of water flow in an annulus space fitted with coiled circular wire turbulators for the turbulent regime,  $Re = 4000-14,000$  and  $Pr = 7$ . Observations of behavior of the heat transfer coefficient under the influence of surface roughness, by inserting a coiled wire, which was twisted around the inner pipe of the heat exchanger, revealed that the heat transfer coefficient was strongly affected with coil pitch and Reynolds number. In counter flow, the ratio of heat transfer rates to that of a smooth tube (relative average Nusselt numbers) is somewhat greater than that in the parallel flow compared with a smooth tube. The heat transfer coefficient was found to increase with Reynolds number and coil pitch, with the highest enhancement observed at Reynolds number of 13160 and 20 mm pitch for the counter flow. The results showed that the values of relative average Nusselt number of counter flow up to 450% was obtained for higher coiled pitch values and higher mass flow rates. While, an enhancement in relative average Nusselt number of parallel flow up to 400% was obtained for higher coil pitch values and higher mass flow rates. The improvement in the heat transferred to a flowing fluid in a heat exchanger by increasing the turbulence level in the flow using a coiled wire, has been previously concluded by many investigators.

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## خصائص انتقال الحرارة في مبادل حراري انبوب داخل انبوب يحتوي على سلك

### ملفوف ذو مقطع دائري

تأثير الاسلاك المعزولة المستخدمة كمول للإضطراب فقط ذات المقطع الدائري بقطر 2 مم والمشكلة على هيئة ملف بخطوات مختلفة مستخدمة كمزودات للإضطراب وذلك على معدلات انتقال الحرارة عمليا قد تم دراسته. التجارب العملية تمت على السريان المائي المضطرب داخل المبادلات الحرارية من نوع انبوب داخل انبوب لكل من السريان المتوازي والمتعاكس (وذلك للمقارنة) للماء البارد المار خلال الفراغ الحلقي للمبادل. التجارب تمت لمعدل سريان متعدد بأرقام رينولدز تتراوح ما بين 4000 وحتى 14000. تم استخدام ثلاث خطوات مختلفة للملف المعزول. نتائج التجارب أوضحت أن استخدام الملف ذات الخطوات المختلفة كمزودات للإضطراب في المائع تؤدي الى زيادة ملحوظة في انتقال الحرارة مقارنة بالأنابيب الملساء وذلك في حالي السريان المتوازي والمتعاكس. قيمة رقم نوسلت المتوسطة تزداد بإزدياد رقم رينولدز وزياد خطوة الملف وذلك في حالي السريان المتوازي والمتعاكس. معامل انتقال الحرارة بالحمل للسريان المضطرب يزداد بإزدياد الإضطراب في السريان بوجود الملف ذو الخطوات المختلفة وكانت أقصى زيادة هي 450% مع السريان المتعاكس بينما كانت 400% في حالة السريان المتوازي. تم استنباط معادلات لرقم نوسلت النسبي المتوسط و معدل الخطوة.