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Experimental and Numerical Investigation to Enhance the Performance of Helical Coiled Tube Heat Exchanger by Using Turbulators

Prof. Dr. Qasim Saleh Mahdi

Mechanical Engineering Dept., College of Engineering , Al-Mustansiriyah University

qasim602006@yahoo.com

Lecturer. Dr. Sahar Abdul Fattah

Mechanical Engineering Dept., College of Engineering , Al-Mustansiriyah University

dr.sahardahash@yahoo.com

Mr. Osama Mahdi Jasim

Mechanical Engineering Dept., College of Engineering , Al-Mustansiriyah University

osama_mahdi58@yahoo.com

Abstract:

In the present work, numerical and experimental investigation is carried out to evaluate the performance of helical coiled tube heat exchanger with wire coil insert. Experimental work included design of wire coiled and inserted into a vertically positioned plain copper helical tube heat exchanger. The effects of wire coil insert with different parameters on heat transfer and friction loss in the helical tube were examined with Dean number ranging from 700 to 2000.The experiments of coiled wire were performed, firstly with three different pitches (P= 15, 20 and 30)mm with constant circular cross section(a=1mm), secondly with different cross section (circular and square) at constant pitch 15mm, and thirdly with two different square wire thickness (a =1, 2) mm at constant pitch 15mm. The numerical analysis was carried out to simulate the flow and heat transfer in a helical coil tube with wire coil insert at pitch (15mm) with different flow rates by using ANSYS FLUENT package 14.0. The experimental results showed that the use of tubular leads to a considerable increase in heat transfer and friction loss over those of a smooth coiled tube. The Nusselt number increases with the increasing of Dean number and the reduction in pitch for wire coils. The coiled square wire provides higher heat transfer than the circular one under the same conditions. A new empirical correlations have been developed to predict Nusselt number of the helical coil heat exchanger. Also the comparisons between numerical and experimental results show good agreement.

Key words: Heat exchanger, Coiled tube, Coil wire insert.

دراسة عملية وعددية لتحسين أداء مبادل الحراري ذو أنبوب ملتف حلزوني باستخدام العوائق

ا.د.قاسم صالح مهدي
قسم الهندسة الميكانيكية
الجامعة المستنصرية

م.د.سحر عبد الفتاح عبود
قسم الهندسة الميكانيكية
الجامعة المستنصرية

أسامة مهدي جاسم
قسم الهندسة الميكانيكية
الجامعة المستنصرية

الخلاصة :

في البحث الحالي دراسة عددية وعملية لتقييم اداء مبادل حراري ذو انبوب حلزوني بداخله سلك حلزوني شبيهه بالنابض كأداة محسنة لانتقال الحرارة ومقارنة أداؤها في الأنابيب مع الأنابيب الملساء. شمل العمل التجريبي تصميم سلك ملفوف وضع داخل أنبوب حلزوني موضوع بشكل عمودي تم دراسة تأثير أبعاد مولد الدوامات على انتقال الحرارة ومعامل الاحتكاك ضمن رقم دين (700 -2000) تم اجراء الاختبارات عند متغيرات عديدة للسلك الحلزوني(مولد الدوامات)،اولا عند استخدام سلك ذو مقطع دائري بثلاثة ابعاد مختلفة من الخطوة (15و20و30 =p)ملم، ثانيا عند استخدام سلك بمقطعين (دائري ومربع)لخطوة ثابتة(p=15mm)وثالثا عند استخدام سلك ذو مقطع مربع بطول ضلع (a=1,2mm)لخطوة ثابتة قدرها(p=15mm).تضمن الجانب العددي نمذجة لسلوك الجريان وانتقال الحرارة للمبادل الحراري باستخدام برنامج حاسوبي(ANSYS FLUENT 14)لتحديد تأثير السلك الحلزوني بخطوة ثابتة (P=15mm) أظهرت النتائج التجريبية أن استخدام مولدات الدوامة يؤدي إلى زيادة كبيرة في نقل الحرارة وفقدان الاحتكاك اكثر من تلك أنبوب الحلزوني الاملس . يزداد عدد نسلت مع زيادة عدد دين ونقصان الخطوة للسلك الحلزوني. لقد بينت المقارنة بين النتائج العملية والنظرية حصول تطابق جيد بينهما. طورت معادلة تجريبية لتوقع عدد ناسلت لأنبوب حلزوني مع إضافة السلك الحلزوني بالداخل.

Nomenclature

Symbol	*Description*	*Units*
A_c	Tube cross-sectional area	m^2
A_s	Tube surface area	m^2
p	pitch	mm
d_t	Tube diameter	mm
Cp	Specific heat at constant pressure	J/kg. K
D_c	Coil diameter	mm
f	Fanning friction factor	
LMTD	log-mean temperature difference	
h	heat transfer coefficient	$W/m^2.K$
k_f	Fluid thermal conductivity	W/m.K
$\dot{m}$	Mass flow rate	kg/s
Nu	Nusselt number	
P	Pressure	Pa
Re	Reynolds number	
$De=Re(d_t/D_c)^{0.5}$	Dean number	
T	Temperature	^{o}C, K
u	Velocity vector	m/s

Introduction:

In many engineering applications the high-performance thermal systems are needed and thus, various methods to enhance heat transfer in the system have been developed extensively. The conventional heat exchangers can be generally improved by means of various augmentation techniques with emphasis on several types of surface enhancements. the heat transfer enhancement enables the size of the heat exchanger to be considerably decreased. In general, the enhancement techniques can be divided into two main groups: active and passive techniques. The active techniques require external forces, e.g. electric field, acoustic, and surface vibration. The passive techniques require special surface geometries such as roughness surface, treated surface and extended surface or fluid additives. Coil wire insert consider one of passive techniques used to enhance heat transfer in helical coil heat exchanger. For decades, many of the wire coil devices employed for augmentation of laminar or turbulent flow heat transfer have been reported and discussed. The use of both active and passive techniques to enhance the heat transfer rate was reported by **Cengiz et. al. 1995** [2]. They studied the effect of rotation of helical pipes on the heat transfer rates and pressure drop for various air-flow rates. The coils were made from copper tubes with a diameter of 10 mm and a length of 3200 mm, respectively. The results showed that although the rotation caused an increase in pressure drop, the heat transfer rates were augmented. A correlation of the heat transfer coefficient for the case of rotating coils was proposed to represent the data within 10% error. In their second paper **1997**[3], the heat transfer and pressure drop in a heat exchanger constructed by placing spring shaped wire with varying pitch were studied. The results indicated that the Nusselt number increased with decreasing pitch/wire diameter ratio. On the basis of the experimental data for both empty helical pipes and helical pipes with springs installed inside. A comparison of the thermal and hydraulic performances between twisted tape inserts and coiled wire inserts was introduced by **Wang and Sunden2002** [1] 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. The effect of coil pitch and other corresponding parameters on heat transfer enhancement and pressure drop of the horizontal concentric tubes with coiled wire inserts were presented by **Naphon2007** [5] and it was found that the coiled wire inserts are especially effective on laminar flow region in terms of heat transfer enhancement. **Promvonge2008** [6] introduced a comparison of the thermal performance of a tube with square and circular cross-sectioned coiled wire inserted tube and the experimental results revealed that the square cross-sectioned coiled wire insert provides better overall enhancement than the circular one does under the same conditions. Analysis of coiled-tube heat exchangers to improve heat transfer rate with spirally corrugated wall investigated by **Zachar 2010** [4]. Different geometrical parameters of helical corrugation on the outer surface of helically coiled-tube heat exchangers are examined to improve the inside heat transfer rate. An empirical formula has been suggested to indicate the dependency of the Nusselt Number on the Dean and Prandtl Numbers. The suggested formula indicating the effect of the basic corrugation parameters for the heat transfer rate. The presented results show that the ratio of the helical pitch and tube diameter (p/d) and the ratio of the corrugation depth and the tube diameter (h/d) nearly increase or decrease in the same way the heat transfer rate of the studied heat exchangers. The results also show that a spirally corrugated helical tube with corrugation parameters (p/d = 1, h/d = 0.1) can increase the heat transfer rate nearly 100% larger than a smooth helical pipe in the Dean number range 30 < De < 1400. The literature survey shows that there are a lack information of heat transfer enhancements studies on coiled wire insert in helical coil tube. Therefore, this paper is focused experimentally and numerically on the enhancement of performance and the optimum values of design parameters for a helical coil heat exchanger with coiled wire inserts placed separately from the tube wall.

Theoretical Formulation:

In the present work, the water used as the test fluid flowed through an inserted helical coil as shown in **Figure(1)** . The steady state heat transfer rate is assumed to be equal to the heat loss in the test section **salimpour2009** [11], which can be expressed as :

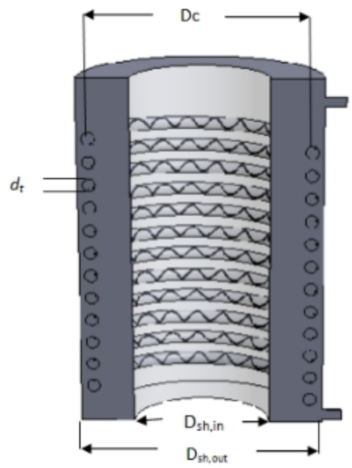


Fig .(1) Helical Coil with Wire Insert

Heat transfer rate through helical coil (hot side)

$$Q_h = \dot{m}_h C_p\,(T_{hi} - T_{ho}) \quad \text{……………..…….. 1}$$

Heat transfer rate through shell side (cold side)

$$Q_{sh} = \dot{m}_{sh} C_p\,(T_{sh_o} - T_{sh_i}) \quad \text{……………..…….. 2}$$

The average heat transfer rate can be define as:

$$Qav = (Q_h + Q_{sh})/2 \quad \text{……………..…….. 3}$$

Experimentally heat transfer coefficient can be calculated such as:

$$\mathbf{h_{in}} = \frac{\mathbf{Q_{av}}}{\mathbf{A_s LMTD}} \quad \text{.....................}\ \mathbf{4}$$

Where $\boldsymbol{A_s = \pi d_{coil} L}$ **..................... 5**

And Nusselt number can calculate as:

$$Nu_{in} = \frac{\mathrm{h_{in}} * d_c}{K} \quad \text{.....................}\ 6$$

The pressure difference can be computed from the following equation:

$$\boldsymbol{\Delta P = f \frac{L}{D} \frac{\rho U^2}{2}} \quad \text{.....................}\ \mathbf{7}$$

Critical Reynolds Number:

According to the research of **Ghorbani 2010**[7]**,** the critical Reynolds number for the helical tube flow, which determines the flow is laminar or turbulent, is related to the curvature ratio as follows:

$$\mathbf{Re_{crit} = 2100\,[\,1+12(d_{in}/D_c)^{0.5}]} \quad \text{.....................}\ \mathbf{8}$$

Dean Number:

The Dean number, De, is a dimensionless group in fluid mechanics, which occurs in the study of flow in curved tube, and is defined as :

$$\boldsymbol{De = Re\,(\tfrac{d}{D})^{0.5}} \quad \text{.....................}\ \mathbf{9}$$

Experimental Test Rig:

Figure (2)and (3) show the photograph and schematic diagram of test rig. The main parts of the test rig are; test section, water heater, water tank, pumps, valves, instruments for measurement. Helical coiled heat exchanger (test section) is designed and manufactured. It is consists of an external and internal shell , smooth helical coil tube, and wire coil insert as a turbulator as shown in **Figure.(4)** and twisted tape are listed in table (1).A water heater is used for heating the water. The hot water is circulated through the helical coiled heat exchanger .The cold water is circulated through the shell of the test section. Twelve thermocouples type (K) are used to measure the hot and cold temperatures at coil tube inlet and outlet ,shell inlet and outlet ,coil tube surface.

Fig .(2) Photo of Test Rig

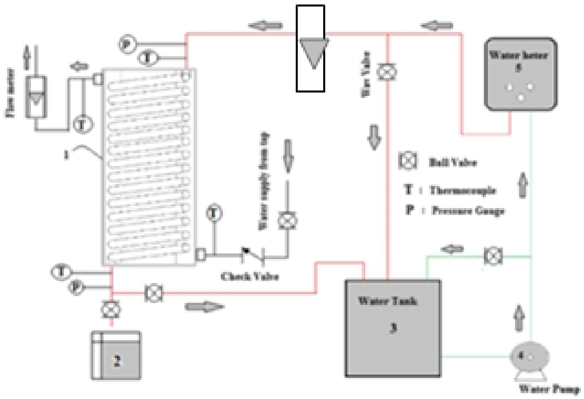


Fig .(3) Schematic of Test Rig

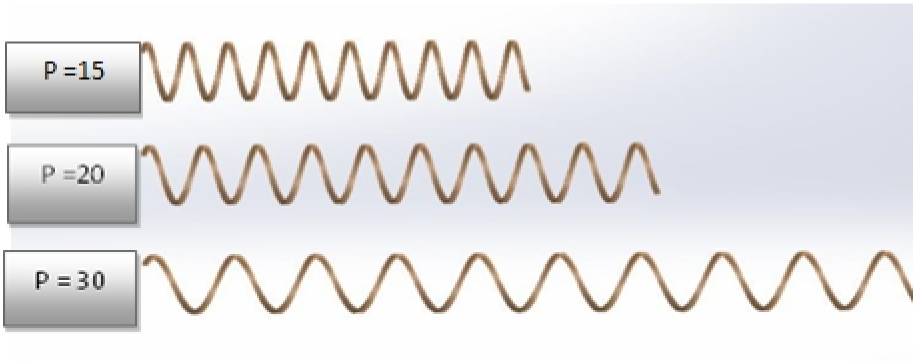


Fig .(4) Wire Coil Insert Pitches

Table (1) Characteristic Dimension of the Turbulators

Tube set	t mm	P mm	d_{spring} Or w	Metal
CW1	1	15	14	Copper
CW2	1	20	14	Copper
CW3	1	30	14	Copper
CW4	2	15	14	Copper
CW5	2dia.	15	14	Copper

Numerical simulation:

The numerical simulation of laminar forced convection heat transfer ,temperature distribution and flow field in helical coil tube with and without insert wire has been done by using ANSYS FLUENT 14 package . The geometry system in present work consists of helical coil tube having inlet and outlet portion with coil wire inside, cylindrical shell also have inlet and outlet portion as shown in **Figure (5).**This system has been drawing by using software program called SOLID WORK PREMIUM 2012 **Ashish Kulkarni 2010** [8].

Fig .(5) Helical Coil With Wire Insert

Assumptions:

After defining the computational mesh and boundary conditions, the user needs to define the assumptions need to be made as follows:

- Steady state
- Newtonian fluid
- Incompressible
- Three dimensional
- Laminar flow

Governing equation:

The conservation equation for continuity, momentum, and energy equations can be written as follows:

1. Continuity Equation

$$\rho\left(\frac{1}{r}\frac{\partial r v_r}{\partial r}+\frac{1}{r}\frac{\partial v_\theta}{\partial \theta}+\frac{\partial v_z}{\partial z}\right)=0 \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 10$$

2. Momentum Equation in the r-component

$$\rho\frac{\partial v_r}{\partial t}+\rho\left(v_r\frac{\partial v_r}{\partial_r}+\frac{v_\theta}{r}\frac{\partial v_r}{\partial \theta}-\frac{v^2{}_\theta}{r}+v_z\frac{\partial v_r}{\partial z}\right)=\mu\left(\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial r v_r}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 v_r}{\partial \theta^2}-\frac{2}{r^2}\frac{\partial v_\theta}{\partial \theta}+\frac{\partial^2 v_r}{\partial z^2}\right)-\frac{\partial P}{\partial r}+\rho g_r \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 11$$

Momentum Equation in the θ-component

$$\rho\frac{\partial v_\theta}{\partial t}+\rho\left(v_r\frac{\partial v_\theta}{\partial r}+\frac{v_\theta}{r}\frac{\partial v_\theta}{\partial \theta}+\frac{v_r v_\theta}{r}+v_z\frac{\partial v_\theta}{\partial z}\right)=\mu\left(\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial r v_\theta}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 v_\theta}{\partial \theta^2}+\frac{2}{r^2}\frac{\partial v_r}{\partial \theta}+\frac{\partial^2 v_\theta}{\partial z^2}\right)-\frac{\partial P}{\partial \theta}+\rho g_\theta \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 12$$

Momentum Equation in the z-component

$$\rho\frac{\partial v_z}{\partial t}+\rho\left(v_r\frac{\partial v_z}{\partial r}+\frac{v_\theta}{r}\frac{\partial v_z}{\partial \theta}+v_z\frac{\partial v_z}{\partial z}\right)=\mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial v_z}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 v_z}{\partial \theta^2}+\frac{\partial^2 v_z}{\partial z^2}\right)-\frac{\partial P}{\partial z}+\rho g_z \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots 13$$

3. Energy equation

$$\rho Cp\frac{\partial T}{\partial t}+\rho Cp\left(v_r\frac{\partial T}{\partial r}+v_\theta\frac{1}{r}\frac{\partial T}{\partial \theta}+v_z\frac{\partial T}{\partial z}\right)=k\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}\right)+\Phi_H \quad \ldots\ldots\ldots 14$$

Boundary condition:

Velocity and Temperature Inlet Boundary Condition:

The velocity inlet to the helical coil is specified over a range of (0.06 to 0.28) m/sec. constant temperature inlet to the coil and shell side are (61,20)℃ respectively.

Pressure Outlet Boundary Condition:

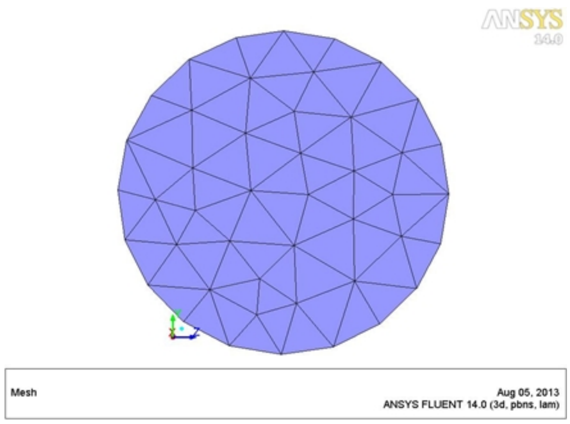
A pressure outlet is specified at the outlet domain where the pressure is assumed to be atmospheric pressure.

Wall Boundary Condition:

No slip boundary condition is specified for the wall of coil. These conditions are used to bound fluid and solid regions. The above governing equations together with the boundary condition have been solved numerically using SIMPLE algorithm and Segregated solver, this is more desirable and quick for the case studied, because the case considered is a large space with a complex geometry and to reduce the memory and time required for reaching the converged solution.

Mesh Topology (Mesh Type):

FLUENT can use meshes comprising triangular or quadrilateral cells (or a combination of the two) in 2D, and tetrahedral, hexahedral, pyramid, or wedge cells (or a combination of these) in 3D. The choice of which mesh type to use will depend on the application. For simple geometries, quadrilateral/ hexahedral meshes can provide higher – quality solution with fewer cells than comparable triangular/ tetrahedral meshes. For complex geometries, quadrilateral/ hexahedral meshes show no numerical advantage and one can save the meshing effort by using triangular/ tetrahedral meshes. Also, FLUENT can use a hybrid mesh. This mesh is with different cell types (hexahedral, tetrahedral and etc.) **Versteeg 1995** **[9]**.In this study tetrahedral mesh type was used because it is superior in the complex geometry. As shown in **Figure (6).** The accuracy of an iterative solution depends on “convergence criterion”. This criterion may be based on an acceptable value for the residual in the discretization equation or an acceptable difference between two successful iterations. In this study a residual is 1×10-6 is achieved for conservation energy as shown in **Figure (7).**



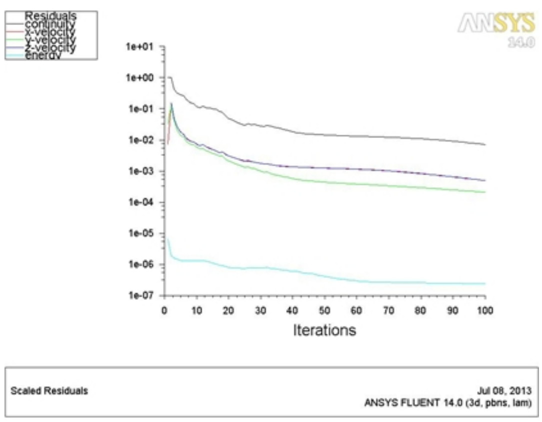


Fig .(6): Mesh Cells of Helical Coil

Fig .(7) Residuals for Running

Results and discussion:

Verification of Helical tube without Turbulator :

Figure (8) and **(9)** show qualification of the heat transfer (Nu) and friction factor (f) of helical tube with out insert. evaluated by comparing the experimental data with the previous correlations (Kalb and **Seader1974** **[10]**, **Cengiz et al.1997**[3] ,*Xin and* ***Ebadian***1997**[11]**, **Salimpour2009** **[12,13]**) The results shown in Figures reveal that the data for helical tube without coil wire insert are in good agreement with the previous reports for both the Nusselt number (Nu) and the friction factor (*f*) correlations. As found, the mean absolute percentage deviations of the present experimental Nusselt number data are ±18% from the values predicted.

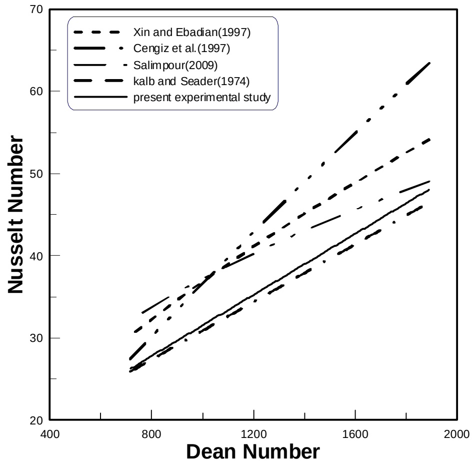


Fig .(8):The Comparison of Tube Side Nusselt Numbers and References

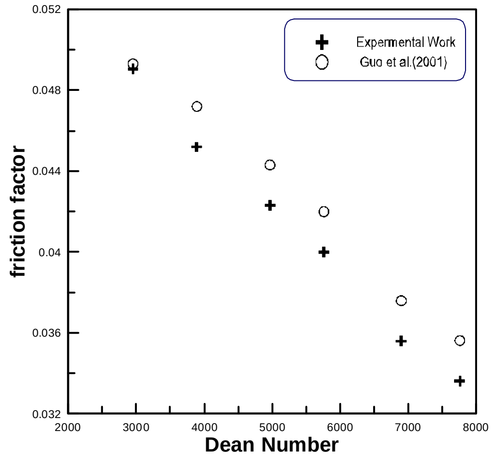


Fig .(9):Verification Test of Plain Tube for Friction Factor

Experimental Results:

Figures (10) shows the variation of the Nusselt number with Dean Number for three spring pitches (P = 15, 20 and 30 mm) of the 1 mm circular wire coil. In the figure, the heat transfer increases considerably with the increase of Dean Number. A close examination reveals that the heat transfer rate at the minimum spring (15mm) pitch is greater than that at the higher one over the Dean number range .This is because the turbulence intensity and the flow path obtained from the minimum pitch are increased. For pitch (P = 15 mm), the increase in heat transfer (Nusselt number) is in the range of (20–78.26%) over that of the helical tube without insert for the Dean number ranging from 700 to 2000. A similar trend is found for the other pitches, the improvement using the spring pitch of P = 15 mm is seen to be about 5–20% better than that of the one of P= 20 mm. The variation of friction factor with Dean Number for the three spring pitches of the 1 mm square wire coil is shown in **Figure (11).**

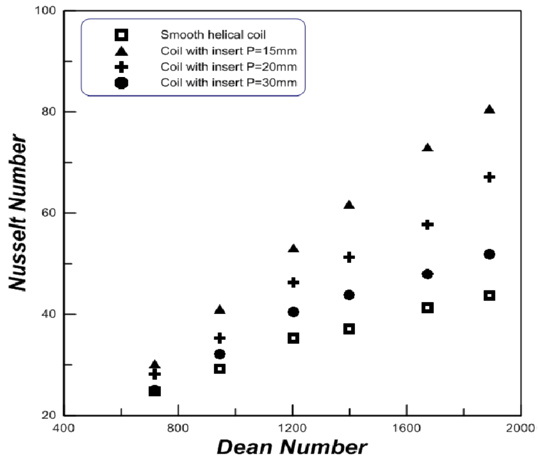


Fig .(10)Variation of Nusselt Number with Dean Number for Different Pitches

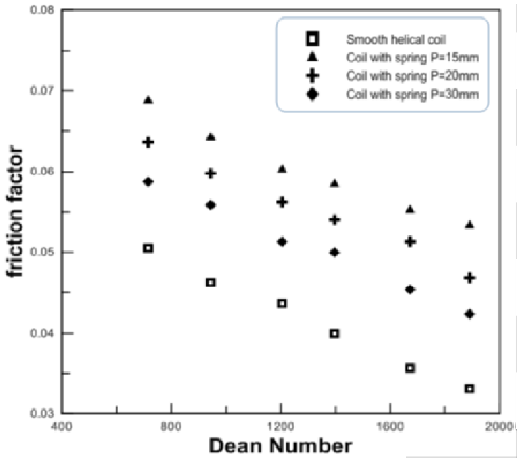


Fig .(11) Variation of Friction Factor with Dean Number for Square Wire

In the figure, the friction factor tends to decrease with the increase of Dean number and spring pitch values. The increase in friction factor with the swirl laminar flow is much higher than that with the smooth helical tube without insert flow. This can be attributed to dissipation of the dynamic pressure of the fluid due to the action caused by the reverse flow. As expected, the friction factor obtained from the smaller pitch is higher than that from the higher pitch The mean increases in friction factor of using the coiled square wire for spring pitches, P = 15,20 and 30 mm are about 57.5% and 38.23% and 26.47 % times that of the helical tube without insert, respectively.

Wire coil inserts of different wire cross sections: circular and square, are presented in the form of Nusselt number and friction factor. The results obtained for two wire cross sections with spring pitch, P = 15 mm as shown in **Figure (12).** In the figure, both coiled wire turbulators yield a considerable heat transfer enhancement with a similar trend in comparison with the smooth tube, and the Nusselt number from both coils increases for increasing Dean number. 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. It is worth noting that the coiled square wire provides higher heat transfer than the circular one for all Dean number values. The average increases in Nusselt number for using square and circular wires are, respectively, found to be about 81.96%and 78.26% times that of the smooth coiled tube. **Figure (13)** presents the variation of friction factor with Dean number for both wire cross sections. In the figure, it is apparent that the use of both coils results in a substantial increase in friction factor above that of the smooth tube. The friction factor of the square wire coil is found to be higher than that of the circular one about 3–4.47%..

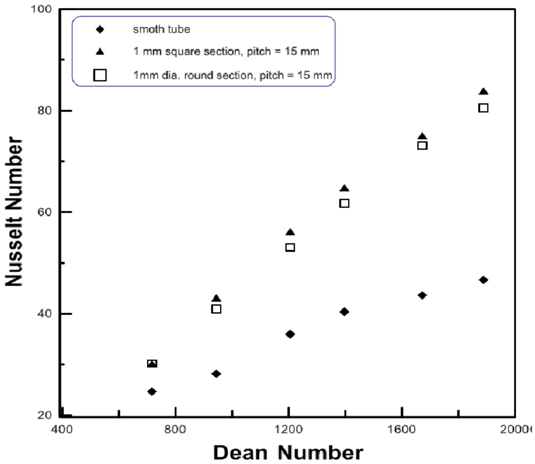


Fig .(12) Variation of Nusselt Number with Dean Number for Circular and Square Wire Coils

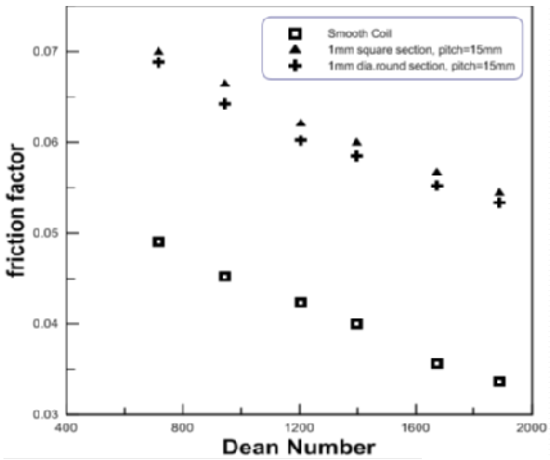


Fig .(13) Variation of Friction Factor with Dean Number for Circular and Square Wire Coils

The influence of the square wire thickness (1 & 2) mm used to form a coil spring with 15 mm pitch on the heat transfer and friction factor in the helical coil tube is presented in **Figure (14)** and **(15),** respectively. In these figures, it is visible that the 2 mm square wire provides both heat transfer and friction factor increases than the 1 mm one for different Dean number values. The Nusselt number increases with increasing Dean number, while the friction factor tends to decrease. This can be attributed to the stronger turbulence intensity, and higher mixing induced by the larger square wire over those of the small one. The increase in Nusselt number for the 2 mm wire is found to be about 2–5.1% over that of the 1 mm one, depending on Dean number values. The friction factor increase for both coil wires. The mean increases in friction losses for the 2 mm square wires are about 7.69% than those of the 1 mm wire.

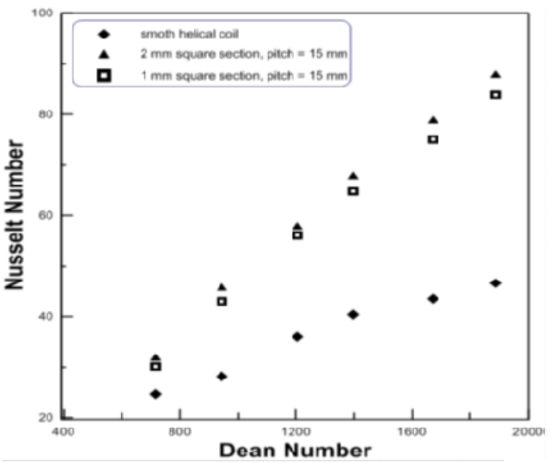


Fig .(14):Variation of Nusselt Number with Dean Number for Different Square Wires Thickness

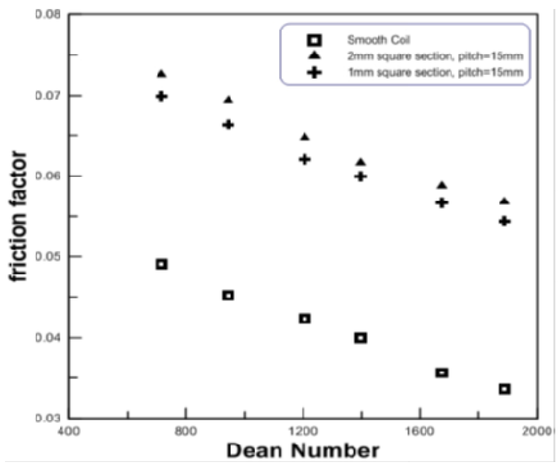


Fig .(15):Variation of Friction Factor with Dean Number for Different Square Wires Thickness

Numerical Result

Validation of the numerical result with results published in literature:

Figure (16) shows the numerical results of Nusselt number of the wire coil inserted helical tube are compared with the numerical results which were carried out by Zachar 2010[4]. The figure show that the simulation data are in good agreement within ±12% deviation for the Nusselt number.

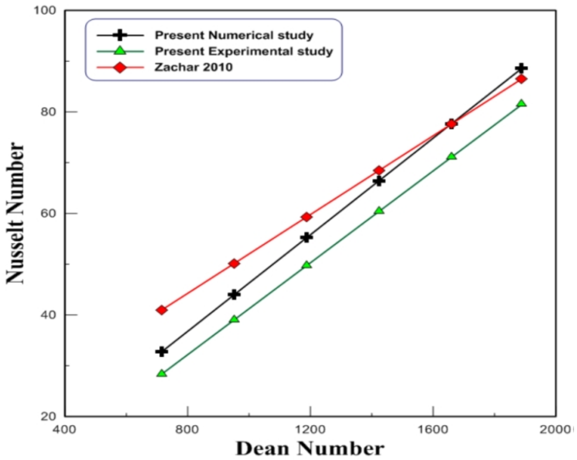


Fig .(16) Comparison Between Predict Nusselt Number with Other Researches.

Temperature Contours and velocity contours :

The temperature distribution across the length has been estimated as shown in **Figures (17)** to **(20)** temperature contours of helical coil with wire coil insert in three dimensions at different velocity inlet.

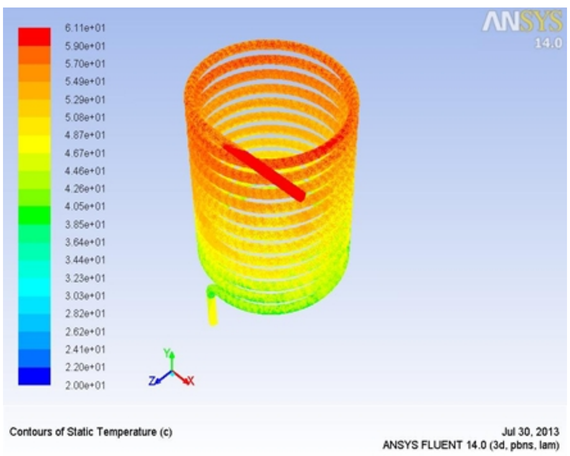


Fig .(17) Temperature Contour of Smooth Helical Coil Tube at 200 l/hr.

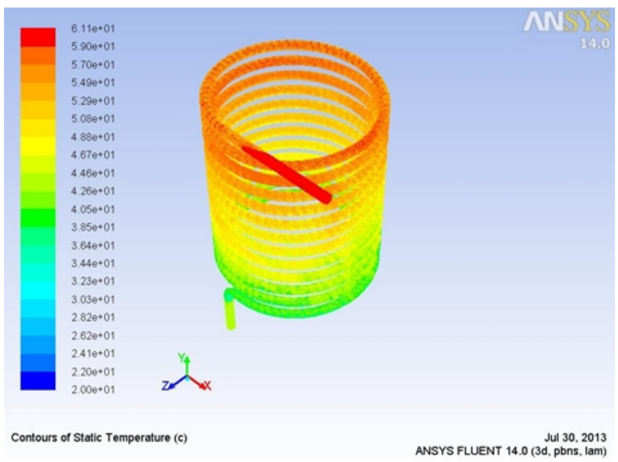


Fig .(18) Temperature Contour of Helical Coil with Wire Coil Insert at 200l/hr.

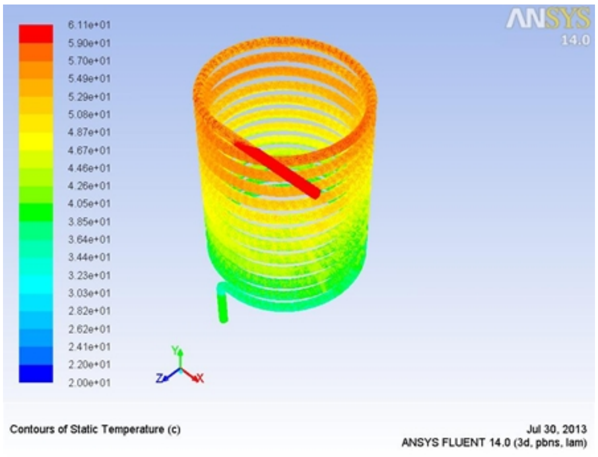


Fig .(19)Temperature contour of Smooth Helical Coil Tube at 150l/hr.

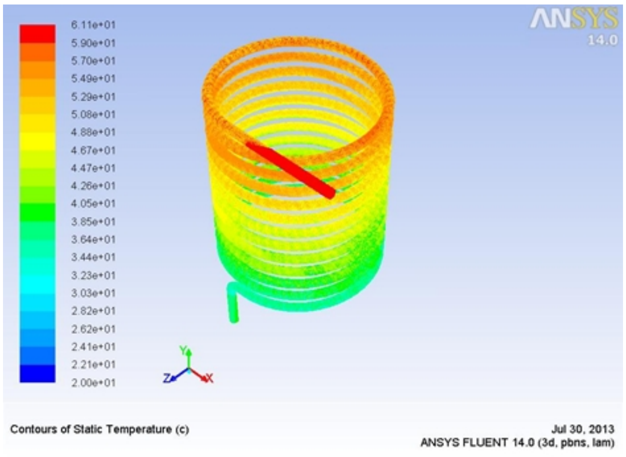


Fig .(20)Temperature Contour of Helical Coil with Wire Coil Insert at 150l/hr.

Figure (21) shows temperature contours at (XZ) plane. From this figures it can be seen gradient of temperature distribution along the sections of turns and maximum temperature occurs at the center of tube. It can be concluded that the temperature field of the inserts case is far more homogeneous than the helical tube without insert case. This is primarily due to the fact that the additional swirling motion of fluid particles in the secondary flow field induces significantly larger heat transfer rate and mixing in the flow case of the tube with inserts. **Figures (22) to (24)** show velocity vector of helical coil containing spring inside. It is clearly seen that the secondary flow field contains much larger velocities than the secondary flow of the smooth tube. The induced additional motion in the cross section of the helical pipe with inserts is the main source of the heat transfer augmentation.

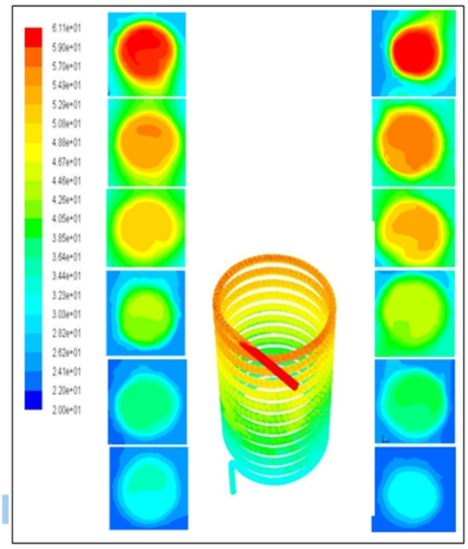


Fig .(21) Contour of Temperature at 100l/hr. of Helical Coil Tube with Wire Coil Insert

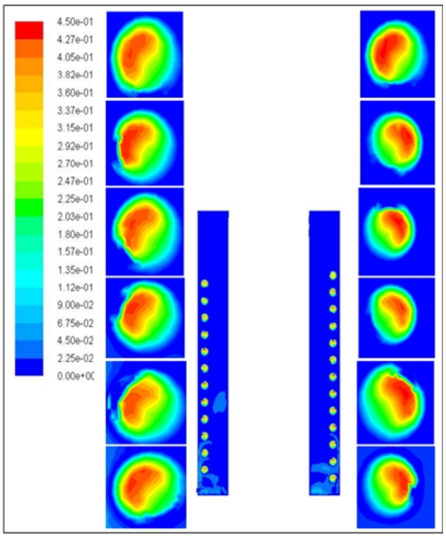


Fig .(22) Velocity Contour of Helical Coil Tube with Wire Coil Insert at 100l/hr.

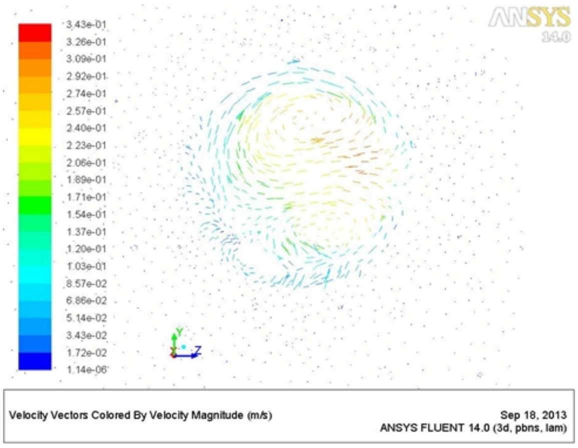


Fig .(23) The Tangential (Secondary) Flow Field of Helical Tube with Wire Coil Insert at De=1673.

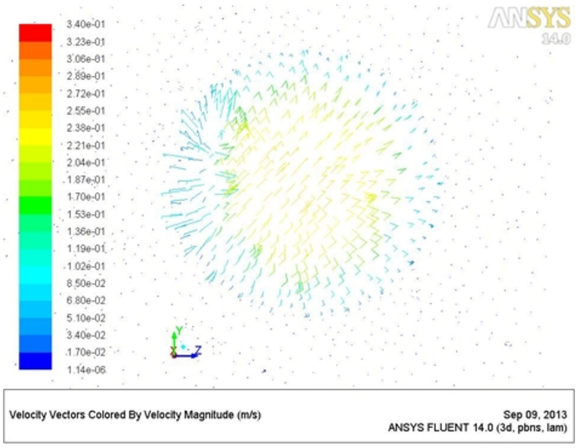


Fig .(24) The Tangential (Secondary) Flow Field of Smooth Helical Coil Tube at De=1673

Development of Empirical Equation

Using the present experimental data and based on the correlations of **Garcia et al.2005[14]** for wire coils inserted tubes, the following correlation is developed to predict the Nusselt number of water flow inside wire coil inserted tubes with an error ± 15%.:

$$Nu = 0.0324 De^{0.91} Pr^{0.3} \qquad 760 \le De \le 2000 \qquad 3 \le P \le 4$$

$$Nu = 0.0424 De^{0.98} Pr^{0.3} \left(\frac{P}{d}\right)^{-0.05} \left(\frac{t}{d}\right)^{0.09} \qquad \text{For} \quad 1 \le P \le 3 \text{ and } t = 1\&2 \text{ mm}$$

Conclusions

An experimental and numerical study has been performed to investigate the water flow friction and heat transfer characteristics in a helical coil tube fitted with coiled wire turbulators for the laminar regime, De = 700-2000. The use of coiled wire causes increase in pressure drop, which depends mainly on spring pitches and wire thickness, if wire coils are compared with a smooth coiled tube at constant pumping power, an increase in heat transfer is obtained, especially at low Dean number. The coiled square wire should be applied instead of the round one to obtain increase in heat transfer and performance, leading to more compact heat exchanger. The numerical approach is generally found to be agreeing reasonably with the experimental results obtained in this work. This agreement demonstrates that the computation scheme is adequately accurate.

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