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Numerical investigation of heat transfer performance of various coiled square tubes for heat exchanger application

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Abstract

In heat exchanger application, working fluid inside the tubes is subjected to considerable temperature changes. In order to improve heat transfer performance, various strategies are proposed and evaluated; one of them is the application of coiled tubes. Coiled tubes have been used widely in heat exchanger application mainly due to the presence of secondary flow which enhances heat transfer considerably. This study addresses heat transfer performance of three configurations of coiled tubes with square cross-section, i.e. in-plane, helical and conical coiled tubes, subjected to large temperature difference. Their heat transfer performance is numerically evaluated and compared with that of a straight tube with identical cross-section and length. A concept of Figure of Merit (FoM) is introduced and utilized to fairly asses the heat transfer performance of the coiled tube configurations. The results indicate that FoM increase as the wall temperature increase. In addition, combination of temperature-induced buoyancy flow and curvature-induced secondary flow considerably affect the flow behavior and heat transfer performance inside the tubes.

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Keywords: coil; heat exchanger; large temperature difference; secondary flow; square cross-section

Introduction

Coiled/curved tube has been widely used in major industry especially in heat exchanger application. It offers higher transfer rate due to the presence of secondary flow induced by the coil curvature. As such, coiled tube has received considerable attention from researcher worldwide. A vast number of studies on coiled/curved tube have been carried out. Among the first studies on coiled/curved tubes was conducted by Dean who observed secondary flow development in a circular toroidal tube when the ratio of the viscous force acting on a fluid flowing in a curved pipe to the centrifugal force (defined as Dean number, De) exceeds the critical value [1,2]. Kurnia et al. conducted a series of computational evaluation of heat transfer performance in coiled tubes and compared with those of straight tubes [3-5]. The results indicate that coiled tubes offers higher heat transfer performance at cost of higher pressure drop. Altac and Altun [6] numerically investigated combined developing flow and heat transfer in spiral tube. From the study, they proposed correlations for the normalized apparent friction factor and the mean axial Nusselt number.

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Nomenclature

c_p	specific heat at constant pressure
k	thermal conductivity
P	pressure
T	temperature
u	velocity
ρ	density
μ	dynamic viscosity

In heat exchanger application, fluid flow through the tube is subjected to high wall temperature. Several studies on the performance of coiled tube in heat exchanger application and various factors affecting its performance have been conducted and reported. Fernandez-Seara et al. [7] evaluated the influence of some crucial geometrical parameters (tube's outer diameter, coil diameter, pitch and length) to the heat transfer performance and pressure drop. To optimize heat transfer performance of coiled heat exchanger, Zachar [8] introduced helical corrugation on the outer side of the helically coiled-tube heat exchangers which enhances the heat transfer by 80-100% as compared to that without corrugation while imposing 10-600 % higher pressure drop. Rennie and Raghavan [9] reported that heat transfer performance of tube-in-tube helical coil heat exchanger increase as Dean Number increase. Kumar et al [10] experimentally investigated and reported that Al_2O_3/water nanofluid enhance heat transfer with the cost of higher pressure drop.

While numerous studies on coiled heat exchanger have been reported, majority of these studies focus on circular cross-section tube. It is therefore of interest to study heat transfer performance of non-circular coiled tube in high temperature heat exchanger application. The present study address transport phenomena in a coiled non-circular tube with the objective to evaluate the flow characteristic and heat transfer performance in various coiled tube geometries subjected to large temperature difference.

Mathematical Model

A three-dimensional CFD model for an incompressible laminar Newtonian fluid flow inside the coiled tubes with square cross-section in coiled non-circular tubes (in-plane spiral, helical spiral and conical spiral) is developed. Schematics representation of the model is presented in Figure 1 and details of the geometrical parameters are tabulated in table 1. Several constant temperatures are prescribed at the wall of the tube. Since this work consider only laminar flow, a precise numerical solution is adequate to simulate reality closely.

2.1 Governing equations

The conservation equations for mass, momentum and energy for the flow inside the tubes are given by [6,7]:

$$\nabla \cdot \rho \mathbf{u} = 0, \tag{1}$$

$$\nabla \cdot (\rho \mathbf{u} \otimes \mathbf{u}) = -\nabla P + \nabla \cdot \left[\mu \left(\nabla u + (\nabla u)^T \right) \right], \tag{2}$$

$$\rho c_p \mathbf{u} \cdot \nabla T = k \nabla^2 T, \tag{3}$$

where ρ is the working fluid density, **u** is the fluid velocity, P is the pressure, μ is the fluid dynamic viscosity, c_p is the specific heat, k is thermal conductivity and T is temperature.

2.2 Constitutive relations

Due to a considerable high wall temperature which will induce phase change for water, only air is studies as a working fluid in this study. To accommodate considerable variation in the air properties due to large temperature range, a polynomial function is adopted to represent temperature dependent properties which were taken from Kays et al. [11]. These functions are applicable for temperature 298-600 K. The air properties are given by

$$\begin{gathered} \rho = 5.28 \times 10^{-6} T^2 - 6.66 \times 10^{-3} T + 2.70,\ \mu = -2.47 \times 10^{-11} T^2 + 6.15 \times 10^{-8} T + 2.29 \times 10^{-6}, \\ k = -2.41 \times 10^{-8} T^2 + 8.64 \times 10^{-5} T + 2.39 \times 10^{-3}, c_p = 3.54 \times 10^{-4} T^2 - 1.64 \times 10^{-1} T + 10.22 \times 10^2. \end{gathered} \tag{4}$$

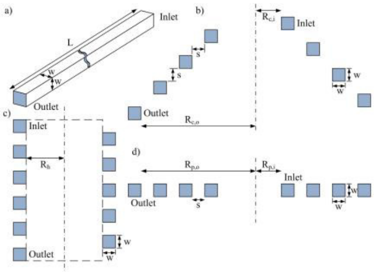

Fig. 1. Schematic representations of a) straight, b) conical spiral, c) helical spiral and d) in-plane spiral tubes.

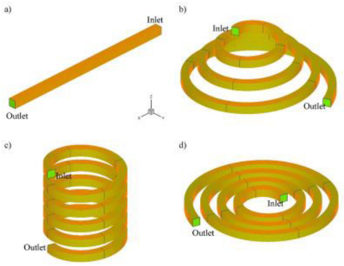

Fig. 2. Computational domains of a) straight, b) conical spiral, c) helical spiral and d) in-plane spiral tubes.

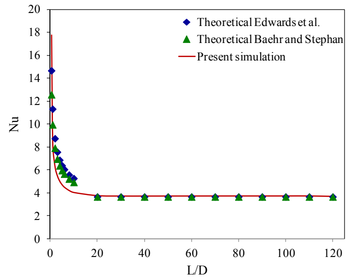

Fig. 3. Comparison of theoretical and computational Nusselt number for straight tube.

The heat transfer performance of the studied coiled tubes is discussed in terms of mixed mean temperature along the tubes, total heat transfer and FoM which are given by

$$T_{mean} = \frac{1}{VA_c}\int_{A_c} \mathbf{u} T dA_c,\ \dot{Q}_{total} = \dot{m}c_p\left(T_{mean,L} - T_{mean,0}\right),\ \text{and}\ FoM = \frac{\dot{Q}_{total}}{P_{pump}}, \tag{5}$$

respectively. Here, A_c is the cross-section area of the tube, V is the mixed mean velocity and P_{pump} is pumping power defined as

$$V = \frac{1}{A_c}\int_{A_c} \mathbf{u} dA_c,\ P_{pump} = \frac{1}{\eta_{pump}}\dot{V}\Delta p, \tag{6}$$

where η_{pump} is pump efficiency (assumed to be 70%), $\dot{m}$ and $\dot{V}$ are the mass and volumetric flow rate, $T_{mean,L}$ and $T_{mean,0}$ are the mixed mean temperature at length L and at the channel inlet, respectively.

2.3 Boundary Conditions

- *Inlet*: At the inlet, constant inlet velocity ($\mathbf{u}_{in}$) of 0.78 m/s corresponding to Reynolds numbers of 500 is prescribed. Constant temperature of 298.15 K (25°C) is defined at the inlet, i.e. $\mathbf{u} = \mathbf{u}_{in}$, $T = T_{in}$.
- *Outlet:* At the outlet, we specify zero gauge pressure and zero streamwise gradient of the temperature, i.e. $p = 0$, $\mathbf{n}\cdot\nabla T = 0$.
- *Wall*: At the wall, no slip conditions and constant wall temperature are set, i.e. $\mathbf{u} = 0$, $T = T_{wall}$

2.4 Numerics

The computational domain which is depicted in Fig. 2 was drawn by using AutoCAD 2010 and then exported to pre-processor software GAMBIT 2.3.16 for meshing and labelling. The mathematical model given by equations 1-3 together appropriate boundary conditions and constitutive relations comprising five dependent variables—*u, v, w, P,* and *T*—was solved by finite volume solver Fluent 6.3.26. To modify the thermo-physical properties of the fluid, user-defined functions (UDF) is written in C language and incorporated to the model.

Table 1: Geometrical parameters

Parameters	Value	Unit	Parameters	Value	Unit
w	1×10^{-2}	m	R_h	4×10^{-2}	m
s	1×10^{-2}	m	$R_{c,i}$	2×10^{-2}	m
$R_{p,i}$	2×10^{-2}	m	$R_{c,o}$	9×10^{-2}	m
$R_{p,o}$	9×10^{-2}	m	L	1.2	m

To ensure mesh independent result, mesh independence test was conducted by systematically increased the number of cell (doubling it) until no significant changes on result (local pressure, velocity and temperature) is observed. From the test, it revealed that the fourth finer mesh which accounted for 2×10^5 cell was sufficient to solve the model. The numerical model was solved with Semi-Implicit Pressure-Linked Equation (SIMPLE) algorithm, second-order upwind discretization and Algebraic Multi-grid (AMG) method. As an indication of the computational cost, it is noted that on average, around 100-400 iterations are needed for a convergence criteria for all relative residual of 10^{-6}; this takes approximately 5-15 minutes on a workstation with a quad-core processor (1.8 GHz) and 8 GB of RAM.

Results and Discussion

In effort to evaluate the heat transfer performance of coiled square tubes, numerical simulation was carried out for both coiled and straight tubes with identical cross-section.

3.1 Model validation

To assess the validity of the mathematical model, the numerical result is compared with the correlation of Nu number proposed by Edwards et al. [12] and Baehr and Stephan [13]. Fig. 3 presents the comparison between the theoretical and numerical value of Nu for straight tube. As can be seen, a relatively well agreement is achieved, highlighting the validity of the mathematical model. Correlation proposed by Edwards et al. assumes that the flow is fully developed when fluid entering the heating section but it can also be used to approximate the developing flow.

3.2 Effect of geometry

Fig. 4 presents the axial profile of air flow inside the tube at a high wall temperature of 573.15 K (300°C). Similar to the previous finding for low wall temperature [3], secondary flow due to the curvature is clearly observed in all coiled tube. Looking into temperature distribution inside the tube, it is confirmed that heat transfer is considerably affected by the presence of secondary flow, as depicted in Fig. 5. Higher temperature profile is observed in all coiled tube geometries as compared to that of straight tube which highlights the beneficial effect of secondary flow in heat transfer performance. An interesting finding from this investigation is that, due to the buoyancy effect, the temperature gradient in the straight tube is higher at the bottom wall as compared to that at the top wall. This buoyancy effect is hindered in the coiled tubes due to the stronger effect of secondary flow which create higher temperature gradient at the outer wall and lower temperature gradient at the inner wall.

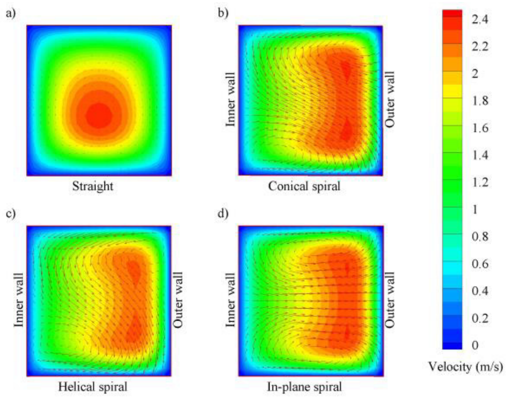

Fig. 4. Axial velocity profile of airflow at T_{wall} 523.15 K, Re 500 and L = 20 cm

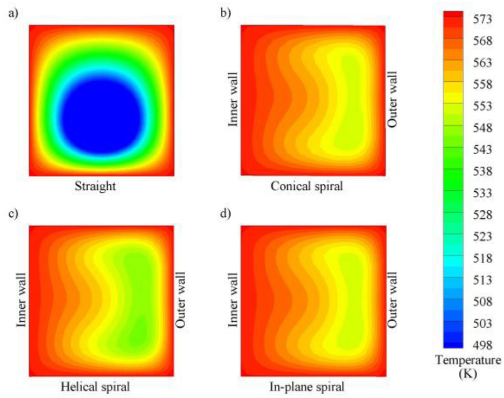

Fig. 5. Temperature distribution at T_{wall} 573.15 K, Re 500 and L = 20 cm

3.3 Effect of wall temperature

A further point of interest in this study is to evaluate the effect of wall temperature on the heat transfer performance. Four wall temperature (T_{wall}) are investigated: 323.15 K (50°C), 373.15 K (100°C), 473.15 K (200°C) and 573.15 K (300°C). Fig. 6 presents mixed mean temperature along the tube at inlet Reynolds number 500 for various wall temperatures. It can be clearly seen that, for all temperature, heat transfer performance of coiled tube overcome straight tube, indicated by higher bulk fluid temperature along the tubes. In addition, coiled tubes require considerably shorter length to achieve the wall temperature as compared straight tube. For example at wall temperature 323.15 K (50°C) coiled tube requires approximately one-third of that needed by the straight tube to reach the wall temperature. This implies that aside providing higher heat transfer, coiled tubes also offers saving from material cost. On closer inspection it is revealed that at low temperature, in-plane spiral achieves the highest bulk temperature closely followed by conical spiral and then helical spiral while straight tube possesses smallest bulk temperature. As temperature increase, conical spiral has the highest bulk temperature closely flowed by in-plane spiral. This behavior is most likely the result of the temperature-induced buoyancy flow which excels heat transfer in conical spiral. At low wall temperature the influence of this flow is marginal but at high wall temperature the effect is more pronounce.

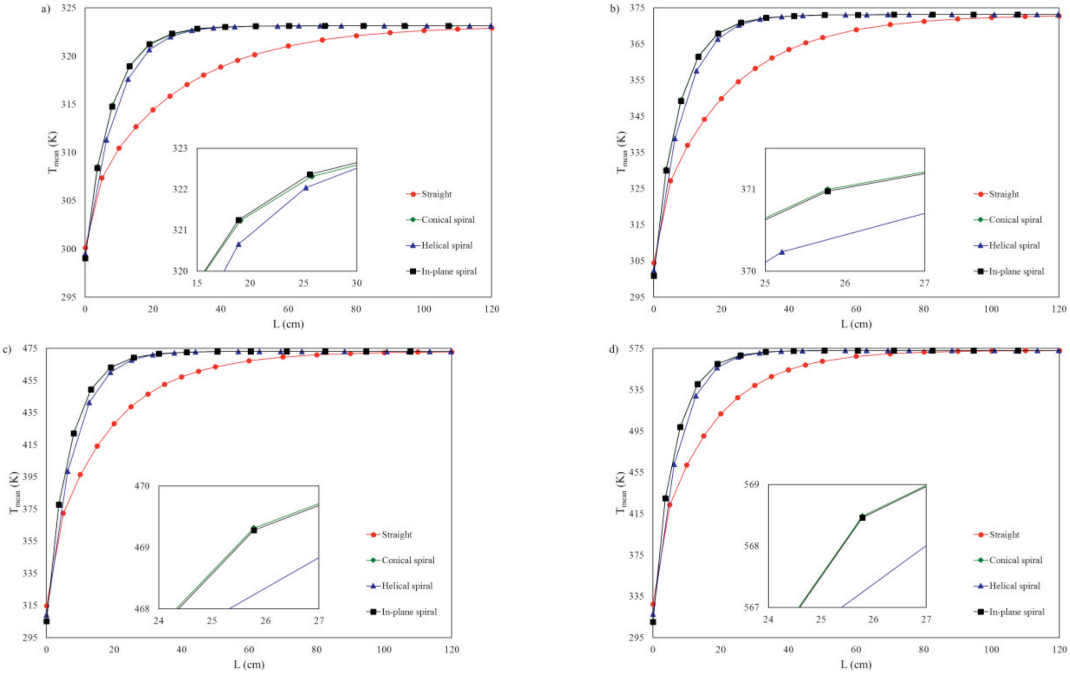

Fig. 6. Mixed mean temperature along the tube for various geometries at Re 500 and wall temperature of a) 323.15 K, b) 373.15 K, c) 473.15 K, and d) 573.15 K.

3.3 Overall heat transfer performance

Table 2 summarize the overall heat transfer performance for all parameters investigated in the present study. Among the considered geometries, in-plane spiral has the highest heat transfer, closely followed by conical spiral and then helical spiral while straight tube has the smallest heat transfer. This result highlighted the importance of the secondary flow on the overall heat transfer performance. Another important finding is that, at high temperature, heat transfer enhancement in in-plane spiral tubes as compared to straight tube is more pronounce. For example, heat transfer in in-plane spiral tube is 11% higher than that of straight tube at wall temperature of 323.15 K (50°C) and it increases to 15 % when the wall temperature increase to 573.15K (300°C).

Another point of interest to assess the performance of a heat exchanger design is the pressure drop induced inside the tube. It is found that pressure drop increase as the wall temperature increase. This is mainly attributed to the fact that viscosity of air increase as the temperature increase. Hence, airflow at high temperature induced higher pressure drop. As expected, higher pressure drop is required to drive the flow through the coiled tube than that of straight tube.

As can be inferred from the table 2, straight tube has the highest FoM, progressively followed by in-plane spiral, conical spiral and helical spiral. This is justifiable looking at significantly lower pressure drop induced in straight tube as compared to coiled tubes. Another point of interest is that FoM increases as the temperature increases. On average, FoM for all configuration at wall temperature of 373.15 K (100°C) is more than two times higher as that for wall temperature of 323.15 K (50°C) and FoM at 573.15 K (300°C) is approximately two times that that at 373.15 K (100°C). This is mainly attributed to the fact that increase in pressure drop is not as significant as increase in heat transfer when the wall temperature increase. In addition it is found that the difference in FoM for straight tube and coiled tubes decreases as the wall temperature increase. For example at Re 100, FoM for straight tube is 20 % higher than that of in-plane spiral tube at low temperature. However, as the temperature increase, the gap decrease, i.e. at

573.15 K (300°C), FoM for straight tube become 17% higher than in-plane spiral tube. This indicates that coiled tube is potential to be applied in high temperature heat exchanger as it offers high heat transfer with relatively high FoM.

Table 2: Total heat transfer, pressure drop and FoM for various configurations at different wall temperature.

Geometry	Total heat transfer (Watt)				Pressure drop (Pa)				Figure of Merit (FoM)			
	$T_{wall,1}$	$T_{wall,2}$	$T_{wall,3}$	$T_{wall,4}$	$T_{wall,1}$	$T_{wall,2}$	$T_{wall,3}$	$T_{wall,4}$	$T_{wall,1}$	$T_{wall,2}$	$T_{wall,3}$	$T_{wall,4}$
Straight	2.11	6.33	14.65	22.74	6.56	7.68	11.18	14.32	2.89×10^3	7.42×10^3	1.18×10^4	1.43×10^4
Conical spiral	2.23	6.69	15.54	24.32	12.16	14.73	20.87	26.44	1.65×10^3	4.08×10^3	6.70×10^3	8.28×10^3
Helical spiral	2.19	6.55	15.17	23.64	13.92	16.49	23.02	28.66	1.42×10^3	3.57×10^3	5.93×10^3	7.42×10^3
In-plane spiral	2.23	6.69	15.55	24.32	12.30	14.52	20.60	26.10	1.63×10^3	4.14×10^3	6.79×10^3	8.39×10^3

N.B. $T_{wall,1}$ =323.15 K (50°C), $T_{wall,2}$ =373.15 K (100°C), $T_{wall,3}$ =473.15 K (200°C) and $T_{wall,4}$ =573.15 K (300°C)

Conclusion

A 3D computational model for laminar air flow have been developed to investigate heat transfer performance of various coiled non-circular tube designs (conical, helical and in-plane) when subjected to large temperature difference between the inlet and wall. Their performance is evaluated and compared to that of a straight tube with identical cross-section. The effect of wall temperature is examined. The result indicates that higher heat transfer performance obtained at high wall temperature, indicated by higher Figure of Merit. On average, FoM for all configurations FoM is doubled when the wall temperature is doubled. Straight tube offer higher FoM due to lower pressure drop as compared to coiled tubes. Among coiled tubes, in-plane spiral provide the highest FoM. It has roughly 2% higher FoM as compared to conical spiral and 14% higher FoM as compared to helical spiral. Two phenomena are observed inside the tube that considerably affects the flow and heat transfer: temperature-induced buoyancy flow and curvature-induced secondary flow. For straight tube the former is dominant while for coiled tube the latter is more pronounced.

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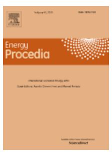

Biography

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