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# Thermal and hydraulic numerical study for a novel multi tubes in tube helically coiled heat exchangers: Effects of operating/geometric parameters

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A B S T R A C T

Compactness with high performance heat exchangers are main challenges in a lot of engineering applications. Thus, this research reports CFD simulation of a novel MTTHC (multi tubes in tube helically coiled) heat exchanger using ANSYS-FLUENT 14.5. The aim of the work is to investigate the thermal and hydraulic performance of the MTTHC for turbulent flow. The effects of the operating and geometrical parameters of the coil on the cold/hot water Nusselt numbers, heat transfer coefficients, pumping power, effectiveness, and thermal-hydraulic index are studied and presented. The results show that, the largest heat transfer coefficient is found at N = 3 and $\beta = 0^\circ$ & $90^\circ$, and the pumping power (P) rises with ≅20 times if $N$ changed from 1 to 5 at any $\beta$. Moreover, the effectiveness of the coil ($\varepsilon$) has the largest values at $\beta = 0^\circ$ & $90^\circ$ and $N=3$, and it enhances with 8.5%, 9% and 7% if $N$ increased from 1 to 3 at $\beta = 0^\circ$, 45° and 90°, respectively. In addition, thermal-hydraulic index ($\xi$) improves with 5%, 8% and 6% if $N$ increased from 1, to 3 at $\beta = 0^\circ$, 45° and 90°, respectively. Finally, Numerical correlations for P, $\varepsilon$ and $\xi$ are correlated and presented within reasonable errors.

## 1. Introduction

Heat exchangers in the shape of helical coils are broadly used in several engineering applications such as energy conversion systems, refrigeration and air conditioning systems, chemical processing, thermal power plants, nuclear reactors, solar energy concentrator receivers, and medical equipment, due to their higher thermal performance and compact size. The flow field and the overall heat transfer coefficient in a helically coiled tube are complex as compared with the conventional heat exchanger and this is due to the dependence of the secondary flow behavior on curvature of tubes. Furthermore, a centrifugal force is generated within fluid flow because of the curvature of the tubes, so the rate of heat transfer is enhanced significantly as the induced of secondary flow. Double tubes and shell and tube helically coils heat exchangers were numerically and experimentally investigated. Owing to the complication of studying the heat transfer processes and fluid flow field in the helically coiled tubes heat exchangers, experimental investigations are costly, limited study parameter ranges and consuming time and the numerical investigations are replacement tool by using CFD packages for this concern.

The Effects of the Prandtl number and geometrical parameters on both the average and local Nusselt numbers for flow in helical pipes was investigated experimentally by Xin and Ebadian [1]. New empirical correlations for the average Nusselt number have been regressed and presented and no noticeable effect of the coil pitch existed. Xin et al. [2] investigated experimentally the effects of the coil geometry and fluid flow rates for both single-phase and two-phase (air/water) flow on helical annular pipes pressure drop for vertical and horizontal coil orientations. Different pressure drop correlations for single-phase and two-phase flow were established and presented. Rennie and Raghavan [3] reported experimentally the heat transfer in a double-pipe heat exchanger comprised one loop. Two heat exchangers with different sizes for both parallel and counter flow configurations were examined. The heat transfer coefficients in the inner tube and the annulus were obtained with different fluids flow rates. A small difference between the overall heat transfer coefficients for the parallel flow and counter flow configurations were found in spite of the higher heat transfer rates that appeared in counter flow configuration. Kumar et al. [4] performed experimental and numerical studies of tube-in-tube helical heat exchanger at the pilot plant scale. The hydrodynamics and heat transfer characteristics were investigated with different inner tube and annulus mass flow rates for counter flow configuration. A commercial CFD package (FLUENT 6.0) was used to predict the flow and thermal profiles in the coil. It was found that the overall heat transfer coefficient

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**Nomenclature**

| | |
|---|---|
| $A_o$ | Cross sectional area of annulus $A_o = \pi(D^2 - N\ d_o^2)/4$, m$^2$ |
| $A_i$ | Cross section area of inner tubes, $A_i = \frac{\pi}{4} N d_i^2$, m$^2$ |
| $Bo_i$ | Hot water buoyancy parameter, $Bo_i = \frac{Gr_i}{\mathrm{Re}_i^{3.5}\mathrm{Pr}_i^{0.8}}$ |
| $Bo_o$ | Cold water buoyancy parameter, $Bo_o = \frac{Gr_o}{\mathrm{Re}_o^{3.5}\mathrm{Pr}_o^{0.8}}$ |
| $C_o$ | Cold water heat capacity, $C_o = m_o^{\bullet} C_{p,o}$, W/K |
| $C_i$ | Hot water heat capacity, $C_i = m_i^{\bullet} C_{p,i}$, W/K |
| $\mathrm{C_{p,o}}$ | Cold water average specific heat, J/kg.K |
| $\mathrm{C_{p,i}}$ | Hot water average specific heat, J/kg.K |
| $D$ | Outer tube diameter, m |
| $D_c$ | Diameter of helical coil, m |
| $De_i$ | Hot water Dean number, $De_i = \mathrm{Re}_i\sqrt{\frac{d_i}{D_c}}$ |
| $De_o$ | Cold water Dean number, $De_o = \mathrm{Re}_o\sqrt{\frac{D_h}{D_c}}$ |
| $D_h$ | Annulus hydraulic diameter, $D_h = \frac{4A_o}{\pi(D + N\ d_o)}$, m |
| $d_i$ | Internal diameter of inner tubes, m |
| $d_o$ | External diameter of inner tubes, m |
| $H$ | Coil pitch, m |
| $h_i$ | Hot water average heat transfer coefficient, W/m$^2$.K |
| $h_o$ | Cold water average heat transfer coefficient, W/m$^2$.K |
| $K_o$ | Cold water average thermal conductivity, W/m.K |
| $K_i$ | Hot water average thermal conductivity, W/m.K |
| $L$ | Length of helical coil, $L = Z(H^2 + (\pi D_c)^2)^{0.5}$, m |
| $\dot{m}_o$ | Cold water mass flow rates, $m_o^{\bullet} = \rho_o v_o A_o$, kg/s |
| $\dot{m}_i$ | Hot water mass flow rates, kg/s |
| $\mathrm{Nu_o}$ | Cold water average Nusselt number |
| $\mathrm{Nu_i}$ | Hot water average Nusselt number |
| $N$ | Inner tubes number |
| $Pr_i$ | Hot water Prandtl number |
| $Pr_o$ | Cold water Prandtl number |
| $Q_o^{\bullet}$ | Cold water heat transfer rates $Q_o^{\bullet} = C_o \Delta t_o$, W |
| $Q_i^{\bullet}$ | Hot water heat transfer rate, $Q_i^{\bullet} = C_i \Delta t_i$, W |
| $Q_{avg}^{\bullet}$ | Average heat transfer rate, W |
| $P$ | Pumping power of the coil, W |
| $Re_i$ | Hot water Reynolds number, $\mathrm{Re}_i = \frac{\rho_i v_i d_i}{\mu_i}$ |
| $Re_o$ | Cold water Reynolds number, $\mathrm{Re}_o = \frac{\rho_o v_o D_h}{\mu_o}$ |
| $Re_{cr,o}\ T_{s,avg}$ | Critical Reynolds number of cold side, $\mathrm{Re}_{cr,o} = 2000\left(\frac{d_i}{D_c}\right)^{0.32}$ |
| $Re_{cr,i}\ T_{s,avg}$ | Critical Reynolds number of hot side, $\mathrm{Re}_{cr,i} = 2000\left(\frac{D_h}{D_c}\right)^{0.32}$ |
| $t_{s,avg}$ | Inner tubes average surface temperature, °C |
| $t_{o,avg}$ | Cold water average temperature, °C |
| $t_{i,avg}$ | Hot water average temperature, °C |
| $t_{o,in}$ | Temperature of inlet cold water, °C |
| $t_{o,out}$ | Temperature of outlet cold water, °C |
| $t_{i,in}$ | Temperature of inlet hot water, °C |
| $t_{i,out}$ | Temperature of outlet hot water, °C |
| $U_o$ | Overall heat transfer coefficient, W/m$^2$.K |
| $Z$ | Number of coil turns |
| $\Delta P_o$ | Cold water pressure drop, Pa |
| $\Delta P_i$ | Hot water pressure drop per inner tube, Pa |
| $\Delta P_{i,\ avg}$ | Hot water average pressure drop for all inner tubes, $\Delta P_{i,avg} = \frac{\sum \Delta P_i}{N}$, Pa |
| $\Delta t_{i,s,avg}$ | Temperature difference between the surface temperature of the inner tubes and the average hot water temperature, $\Delta t_{i,s,avg} = t_{i,avg} - t_{s,avg}$, °C |
| $\Delta t_{o,s,avg}$ | Temperature difference between the surface temperature of the inner tubes and the average cold water temperature, $\Delta t_{o,s,avg} = t_{s,avg} - t_{o,avg}$, °C |
| $Gr_i$ | Hot water Grashof number, $Gr_i = \frac{g\beta_i(t_{i,avg} - t_{s,avg})d_i^3}{\nu_i^2}$ |
| $Gr_o$ | Cold water Grashof number, $Gr_o = \frac{g\beta_o(t_{s,avg} - t_{o,avg})D_h^3}{\nu_o^2}$ |

*Greek symbols*

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|---|---|
| $\beta_i$ | Coefficient of volume expansion of hot water, (1/K) |
| $\beta_o$ | Coefficient of volume expansion of cold water, (1/K) |
| $\varepsilon$ | effectiveness of the coil |
| $\mu_w$ | Water dynamic viscosity, Pa.s |
| $\nu_i$ | Hot water kinematic viscosity, (m$^2$/s) |
| $\nu_o$ | Cold water kinematic viscosity, (m$^2$/s) |
| $\rho_w$ | Water density, kg/m$^3$ |
| $\xi$ | Thermal-hydrodynamic performance index [W/Pa] |

increases with increasing the Dean number in the inner-coiled tube for a constant annulus flow rate. Kumar et al. [5] studied numerically tube-in-tube helically coiled (TTHC) heat exchanger using renormalization group (RNG) k–ε model for modeling the turbulent flow and heat transfer. The fluid flow and heat transfer characteristics were investigated for different inner (compressed air) and outer (cooling water) tube fluid flow rates for both parallel and counter flow configurations. New empirical correlations for the hydrodynamic and the heat-transfer were developed. Jayakumar et al. [6] presented experimental and CFD (Fluent 6.2) theoretical analysis of a helically coiled heat exchanger considering fluid-to-fluid heat transfer. The effects of the actual fluid properties instead of constant values on the heat transfer characteristics were presented and empirical correlation for inner heat transfer coefficient was developed. Piazza and Ciofalo [7] predicted numerically the turbulent flow and heat transfer in helically coiled heat exchangers using the k–ε, SST k–ω and RSM–ω that compared with DNS results and experimental data available of pressure drop and heat transfer. It was observed that the standard k–ε model, with a near-wall treatment presents under prediction of both friction coefficient and Nusselt number. Colorado et al. [8] carried out numerical study and experimental validation to describe the heat transfer and fluid dynamic behavior of a helically coiled steam generator using transient analysis one dimensional model. The proposed model includes subcooled liquid, two-phase flow, and superheated vapour regions. Zhou et al. [9] developed a novel thermodynamic optimization model based on minimizing the work loss for tube-in-tube helically coiled heat exchangers. The effects of main design parameters of the heat exchanger on the available work loss were discussed and presented and the optimal design parameters were also obtained. The results of optimization model provided useful guidance for using such heat exchangers in Joule-Thomson refrigerators. Nada et al. [10] conducted an experimental study of the performance and compactness enhancement of helical-coil in a shell by the attachment of radial fins on the outer surface of the coils. Experimental correlations of Nusselt number in terms Re, Gr, and shell diameter were developed for finned and unfinned coils. Amori [11] investigated experimentally the thermo-fluid characteristics of helically coiled heat exchanger immersed in cold water. Two types of coils were tested; a conventional vertical coil and a new triple vertical coil in parallel connection i.e. meshed coils. The effect of hot water flow rates inside the tubes that varied from 2.67 to 7.08 l/min, and the inlet temperatures, namely 50, 60, 70 and 80 °C were tested. Enhancements of heat transfer and pumping power saving for meshed coils compared to single coil were notices.

Nada et al. [12–14] and Fouda et al. [15] investigated experimentally and numerically the heat transfer and pressure drop characteristics in annulus formed by multi hot rods in tube helically coiled heat exchanger for laminar flow. The effects of the geometric parameters and fluid flow parameters; number of inner tubes, annulus hydraulic