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Numerical investigation of heat transfer and pressure drop in helically coiled tube with spherical corrugation

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ABSTRACT

Heat transfer and pressure drop in helically coiled tube with spherical corrugation are investigated by a three-dimensional numerical simulation. Different geometrical parameters of spherical corrugation in helically coiled-tube heat exchangers are studied to improve the heat transfer rate. Calculated results have been compared to experimental tests and existing empirical formulas to study the validity of the numerical results. The simulation results indicate that the secondary flow induced by the centrifugal force has significant ability to enhance the heat transfer rate, the eddy caused by the corrugation structure destroys the flow boundary layer and increases the turbulence intensity of the flow and strengthens the heat transfer process. With the increase of the corrugation height (H), the augmentation on heat transfer performance is about 1.05–1.7 times as compared to the smooth helically coiled tube, while friction factor sharply increases 1.01–1.24 times. With the increase of the corrugation pitch (P), the enhancement on heat transfer is in the range of 1.37–1.66 times and the friction factor increases 1.18–1.28 times compared to the smooth helically coiled tube. The overall heat transfer performance of helically coiled tube with corrugation is better than that of smooth helical tube under the same condition. The value of performance evaluation criterion (PEC) could be up to 1.56.

1. Introduction

Due to the compact structure, high heat transfer coefficient and thermal stress adaptability, helically coiled tube heat exchangers are extensively used in various industrial applications ranging from refrigeration and air-conditioning systems, power engineering, chemical and food industries, environmental engineering, and many other engineering applications [1–5]. In the heat exchanger, heat is transferred from the warm fluid to the cold fluid via the solid walls. The increasing in the heat transfer coefficient often leads to a higher flow resistance, thereby reducing energy efficiency. Heat transfer efficiency of helically coiled heat exchanger is very high because of the secondary flow caused by the centrifugal force compared to the straight pipes. A survey of the literature indicates that a large number of papers have been published to illustrate the flow and heat transfer characteristic in helically coiled tube including laminar and turbulent flow regimes.

Table 1 summarizes some previous studies in helically coiled tube heat exchanger. Although the structure of helically coiled tube heat exchanger was proposed several decades ago, many researchers still employ this structure in their studies to enhance the heat transfer. The key geometrical properties of the helically coiled tube heat exchanger and the data are selectively reported. Zachár [6] numerically studied the flow and temperature fields of helically coiled tube with spirally corrugated wall, the results indicated that the heat transfer rate increases 80–100% due to the additionally developed swirling motion while the relative pressure drop is 10–600% larger compared to the common helically coiled heat exchangers. Naphon [7] experimentally investigated the thermal performance and pressure drop of the helical coil heat exchangers with and without helically crimped fins. It was found that Inlet hot and cold water mass flow rates and inlet hot water temperature have significant effect on the heat exchanger effectiveness. Ghorbani et al. [8] experimentally investigated the mixed convection heat transfer in a coil-in-shell heat exchanger for various Reynolds numbers, various tube to coil diameter ratios and different dimensionless coil pitch. The result showed that the equivalent diameter of shell is the best characteristic length. Dravid et al. [9] numerically studied the effect of secondary flow on laminar flow heat transfer in helically coiled tubes. It was found that the results acquired from predictions were validated with those obtained from experiments and the asymptotic correlation of the Nusselt

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Nomenclature

A	cross sectional area (m^2)
C_p	constant pressure specific heat capacity ($J\ kg^{-1}\ K^{-1}$)
C_v	constant volume specific heat capacity ($J\ kg^{-1}\ K^{-1}$)
D	the helix diameter of helically coiled tube (mm)
De	Dean number ($=Re(d/D)^{0.5}$)
d	the diameter of smooth tube (mm)
d_e	hydraulic diameter (mm)
f	friction factor, dimensionless
G_k	the generation of turbulent kinetic energy
H	the height of spherical corrugation (mm)
h	the convective heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
k	turbulent kinetic energy ($J\ kg^{-1}$)
L	the length of test tube (mm)
Nu	Nusselt number, dimensionless
P	spherical corrugation pitch (mm)
Pe	wetted perimeter (m)
ΔP	pressure drop (Pa)
p	the pitch of double spiral spring (mm)
Pr	Prandtl number, dimensionless
Re	Reynolds number, dimensionless
S	the screw pitch of helically coiled tube (mm)
T	temperature (K)
ΔT	temperature difference (K)
u, v	the flow velocity ($m\ s^{-1}$)
w	winding number dimensionless

Greek symbols

λ	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
μ	dynamic viscosity ($kg\ m^{-1}\ s^{-1}$)
ρ	density ($kg\ m^3$)
ε	turbulence dissipation rate ($m^3\ s^{-2}$)

Subscripts

i, j	direction of coordinate
c	helically coiled tube with spherical corrugation
s	smooth helically coiled tube

Abbreviations

PEC	performance evaluation criterion

Table 1
The summarized research results of helically coiled tube heat exchanger.

Authors	Working fluid	Condition	Configuration	Correlations
Zachár [6]	Water–ethylene glycol	$30 \le De \le 1400$ $3 \le Pr \le 30$	h/d = 0.1–0.5 p/d = 1	$Nu = 0.5855De^{0.6688}Pr^{0.408}(h/d)^{0.166}(p/d)^{-0.192}$
Ghorbani et al. [8]	Urban water	$1.2 \times 10^7 \le Ra \le 3.2 \times 10^8$ $120 \le Re \le 1200$	p/D = 0.1856–0.2651 N = 16.25,23.25	$Nu = 0.0041Ra^{0.4533}Re^{0.2}Pr^{0.3}$
Dravid et al. [9]	Water	$50 \le De \le 200$ $5 \le Pr \le 175$	d/D = 0.1–0.5	$Nu = (0.65De^{0.5} + 0.76)0.5855Pr^{0.175}$
Havas et al. [11]	Air	$3500 \le Re \le 27000$ $124 \ge Pr \ge 2.7$	d/D = 0.03–0.05	$Nu = 0.187Re^{0.688}Pr^{0.36}(\mu/\mu_w)^{0.11}(d/D)^{0.62}$
Yildiz et al. [12]	Air	$1265 \le De \le 2850$ $Pr = 0.7$	H_s/d_s = 8–24	Empty helical pipes: $Nu = 0.0551De^{0.864}Pr^{0.4}$ Helical pipes with spring insert: $Nu = 4.02De^{0.785}Pr^{0.4}(H_s/d_s)^{-1.008}$
Xin and Ebadian [13]	Air, distilled water, ethylene glycol	$20 \le De \le 2000$ $0.7 \le Pr \le 175$	d/D = 0.0267–0.0884, p/D = 0.2–2.56	$Nu = (0.318De^{0.643} + 2.153)Pr^{0.177}$
Xin and Ebadian [13]	Air, distilled water, ethylene glycol	$5 \times 10^3 \le Re \le 10^5$ $0.7 \le Pr \le 175$	d/D = 0.0267–0.0884, p/D = 0.2–2.56	$Nu = 0.00619Re^{0.92}(1 + 3.455(d/D))Pr^{0.4}$
Guo et al. [18]	Steam water	$6 \times 10^3 \le Re \le 1.8 \times 10^5$	p/D = 0.2343	$Nu = 0.328Re^{0.58}Pr^{0.4}$
Rahul et al. [19]	Propyleneglycol	$7 \times 10^3 \le Re \le 5.5 \times 10^4$	$1.1275 < p/d_o < 1.8575$	$Nu = 0.5186Re^{0.595}Pr^{0.408}(p/d)^{0.857}$
Ju et al. [20]	Steam water	$11.6 \le De$	D/d = 8.0–9.3	$f = [1 + 0.11Re^{0.23}(d/D)^{0.14}](0.316/Re^{0.25})$
Guo et al. [21]	Steam water	$5 \times 10^4 \le Re \le 3.5 \times 10^5$	$0.04296 \le d/D \le 0.07575$	$f_c = 2.552Re^{-0.15}(d/D)^{0.51}$
Naphon and Wongwises [22]	Air	$300 \le De \le 2200$ $Pr \ge 5$	$0.0602 \le P/D \le 0.2023$ $0.0345 \le d/D \le 0.1162$	$Nu = 27.358De^{0.287}Pr^{-0.949}$
Naphon and Wongwises [23]	Air	$200 \le De \le 3000$ $Pr \ge 5$	$0.0248 \le d/D \le 0.06$	Dry-surface conditions: $Nu = 4.0De^{0.464}Pr^{-0.755}$ Wet-surface conditions: $Nu = 19.0De^{0.464}Pr^{-0.755}$

numbers was proposed. Yang and Ebadian [10] numerically analyzed the fully developed turbulent convective heat transfer in a circular cross-section helicoidal pipe with finite pitch. The results revealed that as the pitch of the coil increased, the temperature distribution in the cross-section was asymmetrical, the pitch effect would be augmented as the flow rate increased.

Besides that, Havas et al. [11] further experimentally studied the heat transfer coefficients of helical coils in agitated vessels. A modified heat transfer equation obtained from experimental data was proposed. Yildiz et al. [12] experimentally investigated the heat transfer and pressure drop in a heat exchanger constructed by placing spring shaped wire with varying pitch. The results showed that the Nusselt number increased with decreasing pitch/wire diameter ratio. The Nusselt number correlations were presented based on the experimental data for both empty helical pipes and helical pipes with springs installed inside. Xin and Ebadian [13] experimentally investigated the effects of the Prandtl number and geometric parameters on the local and average convective heat transfer characteristics in helical pipes. The results indicated that the Nusselt number in the laminar flow region changed significantly as the Prandtl and the Dean numbers increased. Some new empirical correlations for the average fully developed were proposed based on the present data. Rozzi et al. [14] experimentally studied the heat treatment of fluid foods in a shell and tube heat exchanger between smooth and helically corrugated wall tubes using Newtonian and non-Newtonian. It is found that heli-