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Experimental analysis of heat transfer enhancement in shell and helical tube heat exchangers

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HIGHLIGHTS

- ► Experimental setup and Wilson plot technique is used for this study.
- ► Optimum coil pitch, coil diameter and flow rates are proposed for heat exchanger.
- ► By decreasing coil diameter and increasing pitch, shell Nusselt number increase.
- ► By increasing coil diameter and decreasing pitch, tube Nusselt Number increases.
- ► By maximum coil diameter and pitch, the highest heat transfer rate is obtained.

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ABSTRACT

In the present work attempts are made to enhance the heat transfer rate in shell and coiled tube heat exchangers experimentally. Hot water flows in helical tube and cold water flows in the shell side. Tube and shell side heat transfer coefficients are determined using Wilson plots. Experimental apparatus and Taguchi method are used to investigate the effect of fluid flow and geometrical parameters on heat transfer rate. After experiments, Taguchi method is used for finding the optimum condition for the desired parameters in the range of $0.0813 < D_c < 0.116$, $13 < P_c < 18$, tube and shell flow rates from 1 to 4 LPM. Then the optimum condition according to the overall heat transfer coefficient for the whole heat exchanger is found. Results indicate that the higher coil diameter, coil pitch and mass flow rate in shell and tube can enhance the heat transfer rate in these types of heat exchangers. Contribution ratio obtained by Taguchi method shows that shell side flow rate, coil diameter, tube side flow rate and coil pitch are the most important design parameters in coiled heat exchangers.

1. Introduction

In industrial applications pipelines and tubes are widely used. Coiled tubes are used in compact heat exchangers, condensers and evaporators in the food, pharmaceutical, modern energy conversion and power utility systems, heating ventilating and air conditioning (HVAC) engineering and chemical industries [1–3]. In coiled tubes centrifugal force make a pair of longitudinal vortices and these secondary flow increases the heat transfer coefficient.

Dravid et al. [4] numerically investigated the effect of secondary flow on laminar flow heat transfer in helically coiled tubes both in the fully developed and in the thermal entrance regions. They presented a correlation for the asymptotic Nusselt number.

$$Nu = \left(0.65\sqrt{De} + 0.76\right) Pr^{0.175}, \quad 5 < Pr < 175, \; 5 < De < 200 \qquad (1)$$

Patankar et al. [5] discussed the effect of the Dean number on friction factor and heat transfer in the developing and fully developed regions of helically coiled pipes. Good agreements were obtained in comparison with the experimental data. Kubiar and Kuloor [6] studied experimentally the heat transfer rate and pressure drop of glycerol flowing inside a vertical helical coil at constant wall temperature. The flow regime was laminar and new correlations were proposed. Rahul et al. [7] presented a correlation for outside Nusselt number of a helical tube. Their results indicated that the pitch of coil significantly affects the outside heat transfer coefficient. Helical and straight tubes were compared by Prabhanjan et al. [8]. The results showed that a helical coil heat exchanger increases the heat transfer coefficient and the temperature rise of fluid depends on the coil geometry and the flow rate. Xin and Ebadian [9] studied the effect of Prandtl number and geometric

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Nomenclature

c_p	Specific heat capacity, J/kg K
d	Tube diameter, m
D	Shell diameter, m
D_c	Coil diameter, mm
D_h	Shell hydraulic diameter, m
De	Dean number, $De = \mathrm{Re}\sqrt{d/D_c}$
h	Heat transfer coefficient, W/m^2 K
k	Thermal conductivity, W/m K
L	Tube length, m
$\dot{m}$	Mass flow rate, kg/s
Nu	Nusselt number, hd/k
n	Number of repetition
P_c	Coil pitch, mm
Q	Flow rate, LPM
q	Heat transfer rate, W
Re	Reynolds number, $4\dot{m}/\pi d\mu$
SN	Signal to noise ratio
T	Temperature, °C
ΔT_{LM}	Logarithmic mean temperature difference, °C
V_i	Tube side fluid velocity, m/s
Y_i	Quality characteristics

Subscripts

i	inlet
o	outlet
h	hot
c	cold
w	wall
cr	critical

Greek symbols

ρ	Density, kg/m s
μ	Viscosity, kg m^{-3}

parameters on Nusselt number and the resulted correlations are as the followings:

$$Nu = \left(2.153 + 0.318De^{0.643}\right)Pr^{0.177}, \quad 0.7 < Pr < 175, \quad 20 < De < 200, \ 0.0267 < d/D < 0.0884 \tag{2}$$

$$Nu = (1 + 3.455d/D)Re^{0.92}Pr^{0.4}, \quad 0.7 < Pr < 5, \ 5\times10^3 < Re < 10^5, \ 0.0267 < d/D < 0.0884 \tag{3}$$

Ko and Ting [10] studied numerically the entropy generation in helical coils at constant wall flux. In their analysis of second thermodynamic law, they found that optimum Reynolds number and curvature ratio are related to the amount of wall heat flux. Naphon and Wongwises [11] reviewed the flow and heat transfer characteristics in curved tubes and tabled the proposed correlations at applicable ranges. Jamshidi et al. [12] studied the optimum shape and flow parameters in helical tube under constant wall temperature by using Taguchi method numerically. Shokouhmand and Salimpour [13] and Salimpour [14] used Wilson plot in order to investigate helical coil heat exchangers experimentally. They presented correlations for inner and outer Nusselt numbers as follows:

$$Nu_i = 0.112De^{0.51}\gamma^{-0.37}Pr^{0.72} \tag{4}$$

$$Nu_o = 5.48Re_o^{0.511}\gamma^{0.546}Pr^{0.226} \tag{5}$$

According to the literature review no experimental work has been done to predict the effect of different design parameters on heat transfer rate in shell and helically coiled heat exchanger. In the present work the heat transfer coefficients in shell and helical tubes are determined experimentally. The effect of shell and tube side flow rate, coil diameter and coil pitch on heat transfer rate in coiled tube heat exchangers are studied by the use of Wilson plot and Taguchi method.

2. Experimental apparatus

2.1. Test section

The test section comprising helically coiled heat exchanger is shown in Fig. 1. Copper helical tubes have 9 mm inner diameter and 12.7 mm outer diameter. The coil diameter (D_c) and pitch (P_c) are depicted in Fig. 1. The plexiglass shell has 14 cm inner and 15 mm outer diameters and 25 cm length. Helical tube in this study has 10 turns. The experimental setup and its schematic diagram are shown in Fig. 2. The setup is a well instrumented heat exchanging system in which a hot water stream flowing inside the coiled tube is cooled by a cold stream flowing in the shell side.

Two 2000 W parallel electric heaters were placed in the hot water storage tank; reach the hot water temperature to the desired value. The hot water is then pumped to the helical tube which is placed in the heat exchanger. The mass flow rate is measured by two flow meters placed in the way of hot and cold waters. As the hot water exits the heat exchanger, its temperature reduces so the hot flow returns back to the hot water storage tank to have the constant hot water temperature at the entrance of helical tube. The cold water has the same closed cycle but the difference is that the cold water temperature increases as it passes through the heat exchanger so a cooling unit placed in cold water cycle fixes the cold water temperature at the entrance of shell. The inlet and outlet temperatures of hot and cold water were recorded manually using 4 k type thermocouples inserted in the small holes made in the inlet and outlet tubes of each heat exchanger and sealed to prevent any leakage. Tests were conducted with varying different parameters such as different flow rates in tube and shell side, different coil and pitches of helical coil to study the effect of these parameters in heat transfer rate. For calculating the heat transfer coefficients in helical tube and shell side, Wilson plot method [15] is used.

2.2. Calculation of heat transfer coefficients

The overall heat transfer coefficient, U_o, is calculated from the temperature data and the flow rates using the following equations [16]: