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Effect of fluid thermal properties on the heat transfer characteristics in a double-pipe helical heat exchanger

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Abstract

Heat transfer characteristics of a double-pipe helical heat exchanger were numerically studied to determine the effect of fluid thermal properties on the heat transfer. Two studies were performed; the first with three different Prandtl numbers (7.0, 12.8, and 70.3) and the second with thermally dependent thermal conductivities. Thermal conductivities of the fluid were based on a linear relationship with the fluid temperature. Six different fluid dependencies were modeled. Both parallel flow and counterflow configurations were used for the second study.

Results from the first study showed that the inner Nusselt number was dependent on the Prandtl number, with a greater dependency at lower Dean numbers; this was attributed to changing hydrodynamic and thermal entry lengths. Nusselt number correlations based on the Prandtl number and a modified Dean number are presented for the heat transfer in the annulus. Results from the second part of the study showed that the Nusselt number correlated better using a modified Dean number. The counterflow configuration had higher heat transfer rates than the parallel flow, but the ratio of these differences was not different when comparing thermally dependent properties and thermally independent properties.

Keywords: Double-pipe; Heat exchanger; Dean number; Laminar flow; Prandtl number

1. Introduction

Using correct and proper values for fluid properties are important when designing a heat exchanger for a particular process, especially for the heating and cooling of complex fluids, such as food products. These properties are a factor in both the rate of heat transfer (and hence the required size of the heat exchanger) as well as the pressure drop across the heat exchanger, which is important for assuring correct pump selection. Temperature dependent properties complicate the design of heat exchangers for specific processes. The prediction of the developing hydrodynamic and thermal boundary layers can be quite complex and difficult. Numerical methods and/or experiments are often required.

Recently there has been significant investigation into the benefits of using helical heat exchangers in food processing applications [1–7], with one of the benefits being higher heat transfer coefficients compared to straight tube helical heat exchangers due to tube curvature [8,9]. The curvature results in secondary flow patterns, perpendicular to the main axial flow, which increase fluid mixing. A reduction in the residence time distribution is obtained with helical heat exchangers [3], which can be important in maximizing quality retention of processed foods.

For the most part, heat transfer characteristics for tube flow are described using dimensionless numbers, with the Nusselt number expressed as a function of the Reynolds number and the Prandtl number. However, for helical coils, the Reynolds number is often replaced by the Dean number [8,9], which is defined as the Reynolds number multiplied by the square root of the curvature ratio (the ratio of the radius of the tube to the radius of curvature). The Dean number is used to represent the strength of the secondary flows; however, not all correlations for heat transfer in helical coils utilize the Dean number [10]. Other correlations have been based on friction factors [11], the Graetz number (in the case of developing flow) [12], and/or the curvature ratio [13].

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Nomenclature

a	constant	
A	constant	$\mathrm{W\,m^{-1}\,K^{-1}}$
b	constant	
B	constant	$\mathrm{W\,m^{-1}\,K^{-2}}$
c	constant	
c_p	specific heat	$\mathrm{J\,kg^{-1}\,K^{-1}}$
d	diameter of inner tube	m
D_o	outer diameter of annulus	m
D_i	inner diameter of annulus	m
De	Dean number ($Re\sqrt{\frac{d}{2R}}$)	
De^*	modified Dean number for the annulus ($\frac{\rho V}{\mu}(D_o-D_i)(\frac{D_o-D_i}{R})^{1/2}$)	
$De^\dagger$	modified Dean number for the annulus ($\frac{\rho V}{\mu}(D_o-D_i)(\frac{D_o}{2R})^{1/2}$)	
Gz^*	modified Graetz number ($De^* Pr\frac{d}{x}$)	
$Gz^\dagger$	modified Graetz number ($De^\dagger Pr\frac{d}{x}$)	
h_i	inner heat transfer coefficient	$\mathrm{W\,m^{-2}\,K^{-1}}$
h_0	outer heat transfer coefficient	$\mathrm{W\,m^{-2}\,K^{-1}}$
k	thermal conductivity	$\mathrm{W\,m^{-1}\,K^{-1}}$
Nu_i	inner tube Nusselt number ($\frac{h_i d}{k}$)	
Nu_o	annulus Nusselt number ($\frac{h_o(D_o-D_i)}{k}$)	
Pr	Prandtl number ($\frac{\mu/\rho}{k/\rho c_p}=\frac{\mu c_p}{k}$)	
r	cylindrical-polar coordinate	m
R	radius of curvature	m
T	temperature	K
T_{abs}	absolute temperature	K
V	average velocity	$\mathrm{m\,s^{-1}}$
x	axial distance	m
z	cylindrical-polar coordinate	m
Greek symbols		
θ	cylindrical-polar coordinate	m
ρ	density	$\mathrm{kg\,m^{-3}}$
μ	viscosity	$\mathrm{kg\,m^{-1}\,s^{-1}}$

The majority of these cases are for thermal boundary conditions of constant wall temperature or constant wall flux, which are different from the boundary conditions found in a fluid-to-fluid heat exchanger [14]. Thermal boundary conditions can affect the Nusselt number; it is well known that for fully developed flow in straight pipes the Nusselt number differs based on the thermal boundary conditions. Under these conditions, changing the flow rate, fluid properties or fluid temperature on one side of the heat exchanger can affect the heat transfer and fluid flow characteristics on the other side of the heat exchanger. For example, increasing the flow rate tends to increase heat transfer rates, which result in an increase/decrease of the average temperature of the fluid on the other side of the barrier. Thus the fluid properties, such as thermal conductivity, density, and viscosity, may change. If the viscosity decreases, the average pressure drop will also decrease. Depending on the type of pump used, this could result in an increased flow rate. Thus it is important to investigate and understand the effects of thermally dependent fluid properties, and how flow rates and geometry can affect the heat transfer characteristics when dealing with thermally dependent properties, as often found in food processing applications.

2. Objective

The objective of this work is to study the effects of fluid thermal properties on the heat transfer characteristics for double-pipe helical heat exchangers. The work was performed using a computational fluid dynamics package (PHOENICS 3.3). The set goals were achieved in two stages:

(1) Determination of the effects of the Prandtl number;
(2) Determination of the effects of thermally dependent thermal conductivity.

3. Materials and methods

3.1. CFD modeling

Geometries for the heat exchanger were created in AutoCAD 14 and exported as stereolithography files. Three different coils were created, one for the annulus, with inner and outer diameters of 0.1 and 0.115 m, respectively, and with a pitch of 0.115 m. The two other coils were created with outer diameters of 0.04 and 0.06 m, both with a pitch of 0.115 m and with a wall thickness that was 15% of the respective outer diameters. Each of the coils had a length of 2π (one full turn).

The geometry files were imported into a computational fluid dynamics software (PHOENICS 3.3), which is based on a control volume-finite difference formulation. A schematic of the cross-sectional view of the heat exchangers, a long with the coordinate system, is shown in Fig. 1. A cylindrical-polar coordinate system (r–θ–z) was used with a mesh size of $30 \times 40 \times 80$ in the axial (θ), horizontal (r), and vertical directions (z), respectively. In cylindrical-polar coordinates (r–θ–z), the continuity equation, momentum equations, and energy equation, respectively, can be written as:

$$\frac{\partial \rho}{\partial t} + \frac{1}{r}\frac{\partial(\rho r v_r)}{\partial r} + \frac{1}{r}\frac{\partial(\rho v_\theta)}{\partial \theta} + \frac{\partial(\rho v_z)}{\partial z} = 0 \tag{1}$$

$$\frac{\partial v_r}{\partial t} + v_r\frac{\partial v_r}{\partial r} + \frac{v_\theta}{r}\frac{\partial v_r}{\partial \theta} + v_z\frac{\partial v_r}{\partial r} - \frac{v_\theta^2}{r} = -\frac{1}{\rho}\frac{\partial p}{\partial r} + \frac{\mu}{\rho}\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial v_r}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 v_r}{\partial \theta^2} + \frac{\partial^2 v_r}{\partial z^2} - \frac{v_r}{r^2} - \frac{2}{r^2}\frac{\partial v_\theta}{\partial \theta}\right) \tag{2}$$