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Fully developed heat transfer in mini scale coiled tubing for constant wall temperature

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ABSTRACT

An experimental study on heat transfer enhancement in mini scale coiled tubing for constant wall temperature conditions is conducted. Copper coils with three different radii of curvature (1 cm, 2 cm, and 4 cm), and in four lengths (1, 2, 3 and 4 coil turns) were used. The tube length is long enough to consider the flow to be hydro-dynamically fully developed. Hence, the effects of varying curvatures and lengths on heat transfer are studied. The pitch of the coil is restricted to diameter of the tube to minimize the effect of coiling. Dean number is used instead of Coiled number (modified Dean number), and hence, the results can be expanded to spiral and curved tubing. Water and two different silicone oils (0.65 cSt, 1 cSt) were used in the experiments to examine the effect of Prandtl number on coiled tubing heat transfer augmentation. Prandtl number from 5 to 15 is covered in this paper. A new asymptotic correlation is proposed to calculate Nusselt number in fully developed coiled tubing based on the current results. In addition, dimensionless mean wall flux and dimensionless thermal length are also considered besides Nusselt, Reynolds, and Dean numbers.

1. Introduction

In general, there are two methods to improve heat transfer rates: active and passive techniques [1,2]. Active techniques are based on external forces such as electro-osmosis [3], magnetic stirring [4], bubble induced acoustic actuation [5] and ultrasonic effects [6] to perform the augmentation. Active techniques are effective; however, it is not always easy to perform the compatible design with other components in a system. They also increase the total cost of the system manufacturing.

On the other hand, passive techniques employ fluid additives or special surface geometry. Using the surface geometry approach is easier, cheaper and does not interfere with other components in the system. Surface modification, adding additives to the fluid or additional devices, incorporated in the stream are three passive augmentation techniques. In these techniques, the existing boundary layer is disturbed and the heat transfer performance is improved; however, pressure drop is also increased [7]. Consequently, their net effectiveness depends upon the balance between the increase in heat transfer augmentation and the pressure drop penalty.

Curved geometry is one of the passive heat transfer enhancement methods that fits several heat transfer applications such as power production, chemical and food industries, electronics, environment engineering, etc. Centrifugal force generates a pair or two pairs of cross-sectional secondary flow (based on the Dean number), which are known as the Dean vortices, improves the overall heat transfer performance with an amplified peripheral Nu variation. The velocity near the outer wall is greater than the flow in straight pipes, whereas, to maintain the momentum balance between the centrifugal force and pressure gradient, slower moving fluid elements move toward the inner wall of the curved tube.

Dean [9,10] conducted the first analytical studies of fully developed laminar flow in a curved tube of circular cross section. He developed a series solution as a perturbation of the Poiseuille flow in a straight pipe for low values of Dean number ($De < 17$). He reported that the relation between pressure gradient and the rate of flow is not dependent on the curvature to the first approximation. In order to show its dependence, he modified the analysis by including the higher-order terms and was able to show that reduction in flow due to curvature depends on a single variable (now known as the Dean number), equal to $Re\sqrt{(D/2R)}$ (where Re is the Reynolds number, D is the tube diameter, and R is the radius of curvature of the curved tube in Deans notation). Various other authors have used different definitions of the Dean number for curved tube studies. It was reported that, for low Dean numbers, the axial-velocity profile was parabolic and unaltered from the fully developed straight tube flow. As the Dean number is increased, the maximum velocity begins to be skewed toward the outer periphery. Similarly, for low values of curvature ratio,

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Nomenclature

b	coil pitch, m
C	experimental correlation constant
C_p	specific heat capacity, J/kg K
D	tube diameter, m
De	Dean number, $\equiv Re\sqrt{(D/2R)}$
Gr	Graetz number, $\equiv \frac{Pe}{L/D}$
h	thermal convection coefficient, W/m^2 K
k	thermal conductivity, W/m K
L	length of curved tubing, m
$L^\star$	dimensionless thermal length
m_1	experimental correlation constant
m_2	experimental correlation constant
$\dot{m}$	mass flow rate, (kg/s)
n	asymptotic correlation constant
Nu	Nusselt number, $\equiv hD/k$
P	tube perimeter, m
Pe	Peclet number $\equiv UD/\alpha$
Pr	Prandtl number, $\equiv \nu/\alpha$
$\bar{q}$	mean heat flux, W/m^2
$q^\star$	dimensionless wall heat flux
R	radius of curvature, m
$1/R$	curvature, 1/m
Re	Reynolds number, $\equiv UD/\nu$
T	temperature, K
U	average liquid velocity, m/s

Greek symbols

α	thermal diffusivity, m^2/s
μ	dynamic viscosity, N s/m^2
ρ	fluid density, kg/m^3

Superscripts

$\star$	dimensionless
$\overline{(\cdot)}$	mean value

Subscripts

c	curved
i	inlet
o	outlet
w	wall

the secondary flow intensity is very high while for high values of the curvature ratio the secondary flow intensity is much less [11].

Due to volumetric constraints, a wide range of curved tubing applications are in coiled shape format. The most popular use for coiled tubing is circulation or deliquification. Coiled tube heat exchangers, logging and perforating, and drilling are other applications of coiled tubing [12,13]. From a heat transfer point of view, helically coiled exchangers offer certain advantages. Compact size provides a distinct benefit. True counter-current flow fully utilizes available LMTD (logarithmic mean temperature difference). Helical geometry permits the handling of high temperatures and extreme temperature differentials without high induced stresses or costly expansion joints. High-pressure capability and the ability to fully clean the service-fluid flow area add to the heat exchanger's advantages. Neglecting the effect of coiled tube pitch, Rennie [14] examined the double-pipe helical heat exchangers numerically and experimentally. Nigam et al. [15] studied a tube-in-tube helically coiled heat exchanger for turbulent flow regime numerically. Shokouhmand et al. [16] experimentally investigated the shell and helically coiled tube heat exchangers in both parallel-flow and counter-flow configuration. They used Wilson Plots [17,18] to calculate the overall heat transfer coefficient.

A complete investigation on heat transfer has been provided by Akiyama and Cheng [19,20] on curved rectangular channels and curved pipes with uniform heat flux and a peripherally uniform wall temperature. Using a numerical method, they found solutions up to a reasonably high Dean number for different aspect ratios. They also showed that the perturbation method is only applicable for a relatively low Dean number region and a boundary-layer technique is valid only for high Dean number regime. The effect of Dean number on velocity and temperature field was also studied in their work.

Nigam et al. [11] accomplished a comprehensive review on the papers related to the application of curved geometry in the process industry up to 2008. The primary objective of their work is to provide an overview of the perspective on evolution of curved tube technology and introduction to a new class of curved tubes. The Nigam's paper [11] is divided into four parts. The first part discusses the fundamentals involved in enhancing the mixing performance and local phenomena in curved tubes to better understand how mixing and heat- and mass-transfer proceed for single-phase flow. In the second part, the chemistry of two-phase flow in curved tubes is discussed and reported. The third part discusses the development of a new class of curved chaotic geometries (bent coils) as a course of technological development. The hydrodynamics and heat and mass transfer performance in curved chaotic configurations have been discussed and compared against curved tube.

Cheng et al. [21] reported that the number of the vortices inside a curved channel depend on the Dean number. Two vortices have been reported for $De < 100$, whereas for $100 < De < 500$, four vortices appear in a square curved channel. These extra vortices vanish at about $De \approx 500$. The channel aspect ratio determines the exact value of the disappearance.

Burmeister and Egner [22] numerically studied laminar flow heat transfer in three-dimensional spiral ducts of a rectangular cross section with different aspect ratios. The boundary conditions were assumed to be axially and peripherally uniform wall heat flux and a peripheral uniform wall temperature. They presented average Nusselt number as a function of distance from inlet and Dean number. Their results are applicable to spiral plate heat exchangers.

Chang et al. [23] conducted an experimental study that examines the detailed Nusselt number (Nu) distributions, pressure drop coefficients (f) and thermal performance factor (TPF) for a square spiral channel with two opposite end walls roughened by in-line 45 degree ribs.

Hashemi and Akhavan-Behabadi [24] experimentally investigated the heat transfer and pressure drop characteristics of nanofluid flow inside horizontal helical tube under constant wall flux. They studied effect of different parameters such as Reynolds number, fluid temperature and nanofluid particle weight concentration on heat transfer coefficient and pressure drop of the flow.

Ghobadi and Muzychka [25] studied the heat transfer augmentation inside short curved mini tubes. They showed that increasing the curvature, $1/R$, leads to heat transfer enhancement. They showed that the entrance effect dominates the augmentation in short lengths, however the secondary flow effect becomes greater by increasing the curved tubing length. It has been shown that the lower the Prandtl number is, the better enhancement is observed.

Manlapaz and Churchill [26] developed a general correlation for fully developed laminar convection from a helical coil for all Prandtl and Dean numbers. Their model was constructed by joining