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Effect of entrance region and curvature on heat transfer in mini scale curved tubing at constant wall temperature

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ABSTRACT

An experimental study on heat transfer enhancement in short curved tubes for the constant wall temperature boundary condition is presented. Copper tubing with four different radii of curvature (1 cm, 2 cm, 4 cm, and 8 cm) and in four lengths were used. Tubing radii were chosen, so that a constant tube length occurs in different curvatures. Hence, the effect of curvature on heat transfer at a constant length can be studied as well as the effect of heat transfer enhancement by changing the length in a constant curvature. Experimental results are also compared with an existing model for curved tubing. Water and three different silicone oils (0.65 cSt, 1 cSt and 3 cSt) were used in the experiments to examine the effect of Prandtl number on curved tubing heat transfer augmentation.

1. Introduction

Using curved geometry is one of the most effective passive heat transfer enhancement methods which fit several heat transfer applications. Curved tubes are an essential component of nearly all industrial processes, ranging from power production, chemical and food industries, electronics, environment engineering, waste heat recovery, manufacturing, air-conditioning, refrigeration, and space applications. The use of curved tubes in continuous processes is an attractive alternative to conventional agitation since similar and sometimes better performance can be achieved at lower energy consumption and reduced maintenance requirement because of no moving parts. For heat and mass-transfer applications, they have the twofold advantages of increasing the transfer rate due to secondary flow and providing high heat and mass-transfer area per unit volume of space. Curved tubes can provide homogenization of feed streams with a minimum residence time and are available in most materials of construction. Early investigations on curved geometries were conducted by Thomson [1], Williams [2], Grindley and Gibson [3] and Eustic [4].

Spiral geometry is one of the applications of curved geometry. The idea of using spiral geometry was first used in spiral plate heat exchangers in late 19th century [5] and further investigations conducted in the 1930s in Sweden [6,7].

The concept is also being used in coiled tubes. 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 [8,9]. From a heat transfer point of view, helically coiled exchangers offer certain advantages. Compact size provides a distinct benefit. Higher film coefficients, the rate at which heat is transferred through a wall from one fluid to another, and more effective use of available pressure drop result in efficient and less-expensive designs. True counter-current flow fully utilizes available LMTD (logarithmic mean temperature difference). Helical geometry permits 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 added to the exchanger's advantages. Neglecting the effect of coiled tube pitch Rennie [10] examined the double-pipe helical heat exchangers numerically and experimentally. Kumar et al. [11] studied a tube-in-tube helically coiled heat exchanger for turbulent flow regime numerically. Shokouhmand et al. [12] experimentally investigated the shell and helically coiled tube heat exchangers in both parallel-flow and counter-flow configuration. They used the Wilson plot method [13] to calculate the overall heat transfer coefficient.

Unlike the flow in straight pipes, fluid motion in a curved pipe is not parallel to the curved axis of the bend, owing to the presence of a secondary motion caused by secondary flow. As the flow enters a curved bend, centrifugal forces act outward from the center of curvature on the fluid elements. Because of the no-slip condition at the wall, the axial velocity in the core region is much greater than

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Nomenclature

A	area, m^2
C_p	specific heat capacity, J/kgK
D	diameter of Tubing,m
De	Dean number, $\equiv Re\sqrt{(D/2R)}$
h	thermal convection coefficient, W/m^2K
k	thermal conductivity, W/mK
L	length of curved tubing, m
$L^\star$	dimensionless length, $\equiv L/H$
$\dot{m}$	mass flow rate, (kg/m^3)
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, $N\ s/m^2$
ν	kinematic viscosity, m^2/s
ρ	fluid density, kg/m^3
θ	curve angle

Superscripts

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

Subscripts

i	inlet
o	outlet
w	wall

that near the wall. To maintain the momentum balance between the centrifugal forces and pressure gradient, slower moving fluid elements move toward the inner wall of the curved tube. This leads to the onset of a secondary flow such that fluid near the wall moves along the upper and lower halves of the torus wall while fluid far from it flows to the outer wall. The curvature, $1/R$, affects the flow patterns, and even slight curvature was observed to modify the critical velocity of the fluid [14].

Dean [15,16] conducted the first analytical studies of fully developed laminar flow in a curved tube. 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 a 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 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 high curvatures, the secondary flow intensity is very high while for lower curvatures, the secondary flow intensity is much less [14].

White [17] numerically studied the pressure drop through a coil of constant curvature. A complete investigation on heat transfer has been provided by Akiyama and Cheng [18,19] 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 regimes. The effect of Dean number on velocity and temperature field was also studied in their work.

Mori et al. [20] studied the effect of viscosity and secondary flow. At the central part of the flow passage, when the intensity of the secondary flow is strong, it is possible to neglect the effect of the viscosity and heat conduction compared with a stress analogous to the Reynolds stress and heat convection due to the secondary flow components. According to their paper, the effect of the viscosity and heat conduction are confined within a thin layer along a wall of the passage, where the intensity of the secondary flow is weakened. They proved that because of this effect one can consider the existence of the boundary layer along the wall of the passage for a flow which is strongly affected by the secondary flow. They divided the flow and temperature fields in their curved channel into two regions: (i) the core region about the central part of the passage where the effect of the secondary flow is dominant over viscosity and (ii) the boundary layer region along the wall where the effect of the viscosity and heat conduction cannot be ignored. They proved the existence of such a boundary layer experimentally. Their method is also applicable to non-circular cross sections as well as circular cross sections.

Nigam [14] 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 this 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 paper 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. Both the first and second parts also review the well established models available for pressure drop, heat transfer, mixing, and mass-transfer in the curved tubes. Methods for estimating the performance of curved tubes from experimental data, empirical correlations, and computational fluid dynamics are also discussed. The third part discusses the development of a new class of curved chaotic geometries (bent coils) as a course of technological development. The hydrodynamics, heat transfer, and mass-transfer performance in curved chaotic configurations have been discussed and compared against curved tubes. More experimental [21,22] and numerical [23,24] studies have been conducted since this time.

Cheng [25] 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