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Numerical simulation of the heat transfer process in a corrugated tube

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ABSTRACT

This paper analyses the effect of spirally corrugation in a simple tube on the heat transferred and friction factor using numerical simulations. The simulations have been validated with experimental data available in the literature. The study compares the behaviour of both smooth and spirally corrugated tubes considering turbulent flow at four Reynolds numbers (15-40 × 10^3) and two Prandtl numbers (2.9 and 4.3). The main novelty of this paper is to perform a 3-D simulation because some previous studies using similar geometry were restricted to a 2-D analysis. For the smooth and corrugated tubes, stainless steel tubes with an inner diameter of 18 mm, a length of 6 m and a wall thickness of 1 mm were used. The corrugated tube has a corrugation depth of 0.43 mm and a helical pitch of 15.86 mm. The meshing process was performed using ANSYS Workbench (v.17.0) with an unstructured grid with a refined mesh near the wall to ensure that the laminar viscous sub-layer was captured. Hence, a k-epsilon ($k-\varepsilon$) turbulence model with a near-wall treatment was used in the proposed simulations. Two grids were used to perform a grid sensitivity analysis. The results for the corrugated tube indicate that the numerical model predicts an average Nusselt number within a maximum relative error of 17% compared with the experimental data, and the differences in the Fanning factor are lower than 9%.

1. Introduction

Heat transfer enhancement techniques are a relevant aspect for the design and operation of heat exchangers (HXs). These techniques are particularly interesting in engineering applications that involve high-viscosity fluids, such as oils or fluids from the chemical and food industry, because in those situations, the flow regime tends to be laminar, which is typically associated with low heat transfer coefficients.

One of the simplest methods to improve the thermal performance of HXs involves deforming the geometry of the tubes to transform a smooth and straight tube into a corrugated one. According to several studies, such corrugation promotes turbulence in the flow and augments the heat transfer coefficient by mixing the boundary layers near the heat transfer surfaces [1–3]. As a consequence, HXs with corrugated tubes can be downsized and still satisfy the same load as a conventional HX with a smooth tube but often at the expense of a higher friction factor and in turn a greater pressure drop and increased pumping costs. Corrugated tubes are widely used in the food industry, where the high viscosity of the fluids makes the HX work in the laminar or transition regime because the corrugations reduce the transition Reynolds number. Hence, it is possible to transform a laminar or transition flow in a smooth tube into a fully turbulent flow in a corrugated tube with the subsequent increase in the Nusselt number and friction factor.

Several researchers have experimentally investigated the effect of corrugation on the heat transfer coefficient and friction factor in the laminar regime with Reynolds numbers below 2.3 × 10^3. Rainieri and Pagliarini [4] used HX tubes with different corrugation types (helical and transverse) and a highly viscous Newtonian fluid as the working fluid with Reynolds numbers of 90–800. They observed significant swirl components in the flow with the helical corrugation and negligible variations in the Nusselt number with the corrugation pattern. Vicente et al. [5,6] experimentally investigated the heat transfer rate using different spirally corrugated tubes in the laminar, transition and turbulent flow regimes with Reynolds numbers of up to 90 × 10^3. They tested several corrugated tubes with different corrugation depths and helical pitches and observed that the corrugation could reduce the transition Reynolds numbers up to values below 1.3 × 10^3 with increments in the Nusselt numbers up to 250% in the turbulent regime. Rozzi et al. [7] compared smooth and helically corrugated tubes to test non-Newtonian fluids in the laminar and transitional flow regimes in a shell and tube HX. When they tested the corrugated tubes, they observed a significant enhancement of the heat transfer coefficient for Reynolds

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numbers above 800. García et al. [8] compared corrugated tubes with other heat transfer enhancement techniques, such as dimpled tubes and wire coils. They recommended wire coils in laminar flow with Reynolds numbers of 200-2 × 10^3 and corrugated or dimpled tubes for Reynolds numbers higher than 2 × 10^3 because of the minor increase in the pressure drop observed under these conditions. Laohalertdecha et al. [9] observed that corrugation of the tubes promoted turbulent flow, which increased the heat transfer coefficient without a significant increase in the pressure drop in a two-phase flow through an evaporator. Barba et al. [10] measured higher heat transfer coefficients and friction factors in a corrugated tube than in a smooth one for a single-phase flow at Reynolds numbers of 100–800.

Although the capacity of corrugated tubes to transform a laminar flow into a turbulent one makes them advantageous for low Reynolds numbers, other researchers, such as Pethkool et al. [11], experimentally studied the effect of helically corrugated tubes in fluids with high Reynolds numbers. They reported higher heat transfer rates for corrugated tubes than for smooth tubes when analysing turbulent fluid flow with Reynolds numbers of 5.5 × 10^3-60 × 10^3. Li et al. [12] studied the friction factor and heat transfer coefficient in two-dimensional roughness tubes for Reynolds numbers above 7 × 10^3 and found that the use of corrugation was preferred to enhance the heat transfer rates in fluids with high Prandtl numbers because in such situations, the pressure drop was not augmented to as large of an extent as in fluids with low Prandtl numbers.

All aforementioned studies are based on experimental analysis. The use of computational fluid dynamics (CFD) techniques based on numerical simulations can be considered a new methodology in this field. These techniques have high potential because they can be used to predict the fluid behaviour in detail as additional information to the experimental data. In addition, CFD enables the analysis of operation of non-existing equipment and a comparison of alternative designs.

Thus, several studies have been performed. Agra et al. [13] performed 2-D axisymmetic numerical simulations of helically finned and corrugated tubes in a turbulent flow with Reynolds numbers of up to 54 × 10^3. They analysed two corrugated tubes from Vicente et al. [5], but they did not compare numerical and experimental data, so their numerical model was not validated. Mohammed et al. [14] also obtained numerical results over a similar range of Reynolds numbers for transverse (not helical) corrugated tubes with rectangular roughness and turbulent flow. They studied a 2-D planar geometry and observed that the Nusselt number increased with the depth and width of the corrugation but decreased with the pitch. Stel et al. [15] numerically studied d-type corrugations (roughly square cavities) in a tube with an inner diameter of 25.9 mm. They focused on the effect of the geometry corrugation on the friction factor but did not investigate the heat transfer problem. According to them, the friction factor increased with increases in the groove length. Zachar [16] studied steady heat transfer in helically coiled-tube HXs and compared the simulation results with experimental values. He performed a 3-D simulation of the flow, although it was restricted to the laminar regime. His results showed that the heat rate in smooth helical tubes was lower than that transferred in HX coils with helically corrugated walls. More recently, Haervig et al. [17] numerically studied sinusoidally spirally corrugated tubes with Reynolds numbers above 5 × 10^3. They focused on the re-circulation flows that might appear around the corrugated surfaces and their effect on the heat transfer and pressure drop. They observed that for certain corrugation lengths, there was an optimal corrugation depth that maximized the Nusselt number.

Most of the commercial and industrial corrugated tubes available in the marked are spirally corrugated. This geometry precludes to use a 2-D simulation because the flow and the geometry are fully 3-D. Previous works [13] that numerically simulate corrugated tubes used artificial 2-D geometries, which permit to perform a simpler and low cost 2-D numerical simulations, but these results are not directly extrapolable to a truly 3-D spirally corrugated tube. So, the main novelty of this work is to perform a 3-D simulation of a commercial spirally corrugated tube. The selected geometry is the tube number three of the work by Vicente et al. [5]. The experimental results in the turbulent regime for both the heat transfer coefficient and friction factor obtained by Vicente et al. [5] have been numerically reproduced.

In the following sections, we first describe the tube geometry, meshing process and conservation equations for the mass, momentum and energy balance of the numerical model. Then, we describe the numerical results at different Reynolds and Prandtl numbers and compare them with the experimental values of Vicente et al. [5]. Finally, we discuss and summarize the main conclusions of this work.

2. Geometry description and numerical model

2.1. Case study

Two different tube geometries were studied: a smooth tube and a spirally corrugated tube. The smooth tube was studied as a reference base case to compare with the results of the corrugated tube. The geometrical characteristics and experimental data for each tube were obtained from the experimental investigation of Vicente et al. [5], who analysed several spirally corrugated tubes at different Prandtl numbers. In that study, the pressure drop and heat transfer were analysed for turbulent flows using an ethylene glycol-water solution to cover a wide range of Reynolds numbers of 2 × 10^3-90 × 10^3 with a constant heat flux condition. The results of the proposed simulations can be validated using these experimental results for each studied tube.

For the smooth and corrugated tubes, stainless steel tubes with an inner diameter of 18 mm and a wall thickness of 1 mm were used. The selected corrugated geometry is tube number three of Vicente et al. [5], which had a corrugation depth of 0.43 mm and helical pitch of 15.86 mm. Vicente et al. [5] did not provide the precise geometry of the corrugated tube, so we assumed the corrugation as an ellipse (see Fig. 1).

Fig. 2 shows a schematic of the experimental facility in Vicente et al. [5]. They divided the length of the tubes into different regions: first, a non-heated region of L_1 = 2.16 m in length to guarantee fully developed flow conditions in the second region of 2.7 m in length. In the second region, a constant heat flux condition was imposed on the perimeter of the tube. Finally, a third region with a length of L_2 = 1.14 m was not heated and acted as the outlet section. The total length of the tube was 6 m, although the heat transfer coefficient and Nusselt number (from the standpoint of the heat transfer problem) were measured only in the heated section of the tube, i.e., $L_{\text{heat transfer}}$ = 2.7 m. In contrast, the friction factor and pressure drop (from the standpoint of the pressure drop problem) were determined in the length of $L_{\text{pressure drop}}$ = 5.2 m, as shown in Fig. 2. This length comprises the entire heated section and part of the non-heated entry and outlet regions. Thus, the total length of

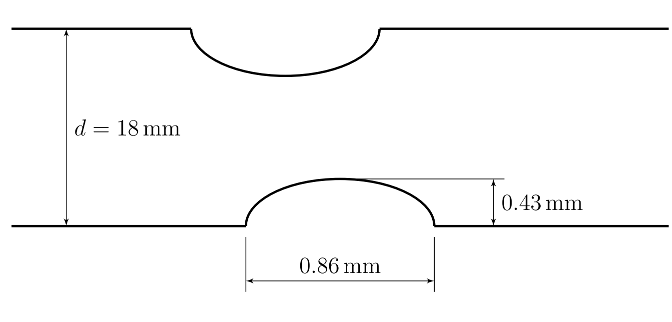

Fig. 1. Detail of the corrugation geometry.