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Design of a device to induce swirling flow in pipes: A rational approach

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ABSTRACT

In this study, a rational approach is proposed to design a device for inducing swirling flow in heat exchanger pipes, for improved efficiency in the laminar regime. First, 2D computational fluid dynamics results lead to select, among four profiles, the blade profile with the most favorable lift to drag ratio. Then, the fluid flow in the swirler made with the selected blade profile is simulated in 3D, for Reynolds numbers ranging from 50 to 1600. Based on the simulation results, an analytic approximation of the evolution of the tangential fluid velocity is proposed as a function of the Reynolds number.

1. Introduction

Heat exchangers are used in many types of applications with the aim of transferring heat from one fluid to the other in an efficient manner. The shell and tube type heat exchangers are among the most common heat exchangers. In most cases, the fluid flowing inside the pipes of these kinds of heat exchangers is in the turbulent regime, since it provides heat transfer rates much higher than in the laminar regime (see, e.g., [1]). However, there are some specific applications where laminar flow can be expected, for example when the working fluids are highly viscous or when the transporting pipe diameter is small. Consequently, low heat transfer rate and increased fouling rate occur.

Different methods for increasing heat transfer in laminar and mildly turbulent flows in tubes have been investigated for years. One particular technique leading to wall shear stress enhancement is to induce rotating or swirling flows inside the pipe. Swirling flows in circular or annular tubes have been widely investigated for the turbulent regime [2–4] but less for the laminar regime [5–7]. These flows are formed supplementing the axial velocity with a tangential velocity component, turning the flow either clockwise or counter-clockwise and giving it a helical motion, or the so-called corkscrew pattern. One of the most widely used methods consists in installing helical spirals or other fins, either inside or outside heat exchanger tubes [8–10]. This method greatly enhances heat transfer and, since it increases the wall shear stress, the fouling rate is also globally reduced. But the drawbacks of such an installation are a really high increase in pressure drop and the presence of local fouling in areas close to the fins where low velocities can be expected. It also may complicate cleaning procedures.

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Confined swirling flows can be classified into two main categories: the first one is the continuous swirling flow, for which the swirl pattern is quite constant along the pipe. The second category is the decaying swirling flow. Most continuous swirling flows are produced by inserting full-length twisted-tapes or helical screw tapes into the pipes [11,12]. Such a way to induce swirling flows is particularly efficient in increasing heat transfer, but it usually generates a too large pressure drop, which is a drawback. This is even worse if an inner rod is installed to hold the tapes on location, which is equivalent to an annular tube configuration. Swirl can also be driven by an axially rotating pipe wall [13]. This technique is mainly used for fundamental investigations and validation of numerical models developed to predict swirling flows.

For decaying swirling flows, the flow which is axial is given an additional angular component at a specific cross section, usually at the inlet of the tubes. The helical movement continues along the pipe for a certain distance, but residual swirl effects might still be visible after a considerable greater distance [14]. One advantage of inducing the swirl motion at a single cross section is that a large portion of the downstream pipe volume is free of any inserted objects that could trap deposits (local fouling) or a hindrance when cleaning the tubes.

Decaying swirling flows can be generated by rotating a section of the pipe, by using tangential injectors [15,16], or by the use of a so-called swirl inducing device, whose appellation in the literature can slightly differ. Among others, the names are: swirling device, swirler device or even shorter: swirler. Various swirler types exist and their applicability can be forced rotating, freely rotating, or fixed (static), to name a few. Forced swirlers, as for example rotating honeycombs, must be avoided in tubular heat exchangers because of excessive pressure drop, power consumption and installation costs. For the purposes of swirling flows in heat exchangers, they should be considered as a source of flow structure information. Static swirlers in circular tubes can be short-length twisted tapes, short twisted-fins or axial guide vanes [17–19].

Two main conclusions can be drawn from the already cited works. Firstly, studies of the profiles of the radial component of the velocity show that it does not play an important role in the evolution and decay of the swirling motion. Secondly, the main contribution to the swirl is related to the tangential component of the velocity. The tangential velocity field can generally be divided into three regions: i) the core region near the axial centerline where the tangential velocity is proportional to the radius, as in a forced vortex; ii) the annular region, where a free vortex occurs and the tangential velocity is inversely proportional to the radius—note that at the intersection of the first two regions, the tangential velocity reaches a maximum; iii) the wall region, which corresponds to a boundary layer. A swirling flow can thus be considered as a combined vortex. The maximum of the tangential velocity decays in the axial downstream direction and the position of this maximum moves towards the center line. So the size of the core region decreases along the axis.

In most studies dealing with short static swirlers, the effect of the variation of different parameters such as the twist angle of the blades, the number of blades, the distance between blades or the length of the blades are presented. To the authors' knowledge no study presents the swirler design process as is done in [20] or [21] for turbine blades; the focus of this paper is the presentation of such a process. It should however be added that a static swirler does not exactly resemble a turbine, nor a compressor, but rather something in between, since no mechanical energy is involved when the fluid flows through the swirler. This design process is three-fold and is followed in the present study. The first step is the selection of a 2D profile, based on the lift-to-drag ratio of the blades. The second step is the computation of 3D flow fields in the domain affected by the swirling flow. The last step is the determination of the Reynolds number range for which the assumption made in the first step is valid. Thus, in the first part of this paper, a blade design process is presented, which is based on hydrodynamic considerations in order to get a desired tangential velocity profile. The second part of the paper presents the resulting flow properties that are obtained using Computational Fluid Dynamics (CFD), for Reynolds numbers in the range 50 to 1600, ensuring a laminar flow regime. The latter is based on the tube diameter and on the mean axial velocity upstream the swirler.

2. The blade design

A general design concern of a swirler is to maximize the swirling effect of the internal flow in a tube, without increasing pressure drop excessively. Two sources of pressure drop are most important here: i) friction because of insertion of blades (hydrofoils). ii) The disturbance of the otherwise smooth flow profiles in a low Reynolds number tube flow. The first source can be related to the number of blades chosen and to their thickness, as well as the structure of the swirler (e.g., how blades are connected). The second source is related to the blade shape and how well they direct the flow into a smooth swirling motion, preferably without generating local vortices. Fig. 1 shows an example of a static swirler having eight blades.

The purpose of the blades is to force the fluid to deviate from its initial direction. This deviation, which represents the amount of swirling in the tube, is defined by the so-called "flow exit angle" ϕ. Fig. 2 shows the definition of the parameters that define the shape of the blades in this study. The blades have zero thickness and are therefore only defined by the camber line shape. Note that as the paper focuses on the geometrical definition of the swirler, it is more adapted to define the swirler in terms of geometrical parameters than to define the swirler in terms of fluid dynamics parameters such as the swirl number.

The blade exit angle θ, at a given radial location determined by a diameter D, differs from the actual flow exit angle ϕ of the velocity by a deviation angle δ due to the fact that the fluid will not "follow" the blade profile exactly. A rule of thumb is

$$\delta = m\frac{(90-\theta)}{\sigma^{n}}$$