Evaluation of the minimum required length to study improved heat exchangers via computational fluid dynamics

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Abstract. Heat exchangers are fundamental for energy plants and the operation of industrial processes. As a result, active and passive techniques are needed to increase heat transfer and develop efficient and reliable designs. The design of heat exchangers has been based on iterative methods that require experimentation to be validated. These circumstances, added to the development of powerful microprocessors, promote the use of numerical methods to study flow distribution and pressure drop, as well as thermal analyses applying different turbulence models and several coupling speed-pressure schemes. This paper examines flow development through curved and twisted tube heat exchanger by means of a computational fluid dynamics numerical model developed in ANSYS® in order to establish the minimum length required to infer conclusions for the complete exchanger. This work tries to reduce the computational cost while maintaining the balance between desired accuracy and available computational resources. Finally, this study shows that the flow can be considered fully developed after it has traveled 360° inside the helix; after that point, secondary flows and re-circulation regions improve the heat transference in these devices can be verified.

1. Introduction
Heat exchangers are fundamental for energy plants and the operation of industrial processes [1]. As a result, active and passive techniques are needed to increase heat transfer and develop efficient and reliable designs [2].

The design of this type of devices has been based on iterative methods that require experimentation to be validated. These circumstances, added to the development of powerful microprocessors, promote the use of numerical methods to study flow distribution and pressure drop, as well as thermal analyses applying different turbulence models (e.g., k-ε, k-ω) and several coupling speed-pressure schemes (e.g., simple, simplec, piso, etc.). Computational fluid dynamics (CFD) is the branch of fluid mechanics that employs numerical methods and programming languages to implement algorithms in the analysis of problems and phenomena of all kinds of fluids. Computers are used to carry out millions of calculations related to equations and mathematical models associated with the phenomenon in a short time. Commercial CFD packages have been used to predict mass and thermal flow development, and their results have been coherent with those of different experimental configurations, such as concentric spiral tube heat exchangers, shell and tube spiral exchangers, rectangular spiral tubes, biphasic flow spiral tubes, monophasic flow spiral tubes, and spiral twisted tubes. Such predictions have been obtained using the computer programs FLUENT®, CFX®, PHOENICS®, and ANSYS® [3]. The quality of the
predictions has fallen within the acceptable interval for the conditions of experimental or real processes, thus showing that CFD is an efficient tool to predict the behavior and performance of a wide variety of heat exchangers [4,5].

This work examines flow development through curved heat exchangers by means of a CFD numerical model developed in ANSYS®. Its main goal is to establish the minimum length of the heat exchanger that will allow us to draw applicable conclusions for the complete exchanger, thus reducing computational cost and maintaining a balance between desired accuracy and available computational resources. Additionally, a mesh independence study was carried out to establish the minimum length and mesh density required for the development of external flow correlations in exchangers with double passive improvement. This study shows that the flow can be considered fully developed after it has traveled 360° inside the helix; after that point, the formation of secondary flows and re-circulation regions improve the heat transference in these devices.

2. Method

2.1. Geometric development
The simulation process is divided into three stages: pre-processing, processing, and post-processing. In the first stage, the geometry is defined; that is, the limits of the physical system are established. In this project, a mass of water inside the external tube of a helical concentric tube exchanger receives a heat flow from the inside wall, and we assume that the outer one is adiabatic. The exchanger was developed in SpaceClaim®, and all the procedure described in this section was applied to the helix in Figure 1.

2.2. Discretization of the control volume
The volume occupied by the fluid is discretized using cells (meshing) in ICEM®. The resulting mesh is optimized by applying an inflation to the surfaces over which the hydrodynamic limit layer (internal and external walls) and the thermal limit layer (the inside wall) are developed. Table 1 details the metrics of the mesh employed in this process. The second column indicates the minimum size established in the refinement strategy; the fourth column lists the average value of the density along the exchanger in terms of mesh elements per millimeter. An average is calculated for each maximum size and the effect of the densification strategy and its length independence can be verified. Such effect can also be noted in Figure 2, where representative images of the maximum mesh sizes under evaluation are presented.

2.3. Models, materials, and boundary conditions
The settings module was enabled for the simulations in CFX®. We used the turbulence model k-ε to represent the turbulent flow inside the exchanger. Additionally, the materials and their properties were defined for the simulations. Liquid water was configured with thermo-dependent properties at a standard
temperature; however, they were modified to an average temperature after some pilot tests. The values of the properties of the materials at the estimated temperature were calculated using the polynomial Equation (1) to Equation (4) proposed by Zachár in [6,7] and Jayakumar et al. in [8]:

$$
\rho = 998.25 - 0.123261T - 1.31119x10^{-3}T^2 - 1.21406x10^{-5}T^3
$$

$$
\mu = 1.66167x10^{-3} - 4.10857x10^{-5}T + 4.64802x10^{-7}T^2 - 1.90559x10^{-9}T^3
$$

$$
c_p = 4222.62 - 6.94932x10^{-1}T + 6.24126x10^{-3}T^2 + 8.29448x10^{-6}T^3
$$

$$
k = 5.68733x10^{-1} + 1.96461x10^{-5}T - 9.77855x10^{-6}T^2 + 1.2432x10^{-8}T^3
$$

Where, T: temperature characterizing the fluid (°C); ρ: characteristic density (kg/m3); μ: characteristic dynamic viscosity (Pa.s); c_p: characteristic specific heat (J/kgK); k: Characteristic thermal conductivity (W/mK).

**Table 1.** Metrics in the mesh independence study of a curved smooth tube.

| Mesh number | Element maximum size (mm) | Number of elements | Average mesh density (elements/mm²) | Minimum skewness | Minimum OQ | Mesh quality |
|-------------|--------------------------|--------------------|-------------------------------------|------------------|-----------|-------------|
| 1           | 5                        | 207680             | 0.1834                              | 0.859-0.992      | 0.8272    | very good   |
| 2           | 4                        | 308352             | 0.2723                              | 0.766-0.925      | 0.8335    | very good   |
| 3           | 3                        | 571136             | 0.5043                              | 0.727-0.944      | 0.8318    | very good   |
| 4           | 2                        | 1345536            | 1.1881                              | 0.727-0.944      | 0.8318    | very good   |
| 5           | 1                        | 6257664            | 5.5254                              | 0.421-0.981      | 0.5835    | good        |
| 6           | 0.9                      | 8118144            | 7.1681                              | 0.377-0.986      | 0.5315    | good        |

We specified the behavior and flow properties inside the limits of the problem to define boundary conditions. The study was carried out under a constant temperature of 90 °C in the inner wall (Dirichlet’s condition) and an adiabatic condition (no heat flux) in the outer wall. Opening type of boundary was established for the outlet. A constant temperature of 20 °C and variable speeds were established for the input. The simulations were conducted at velocities of 0.1 m/s, 0.5 m/s and 1.0 m/s.

**2.4. Results obtention and datums analysis**

To obtain the results, tangential projections of the output speed were used to reveal secondary flows and eddy formations. This also helped to shed light on the way they evolve along their travel through the exchanger. The analysis model evaluated relative error in the Equation (5):

$$
\text{Error} [\%] = 100 \left| \frac{Nu_c - Nu_i}{Nu_i} \right|,
$$

where Nu stands for Nusselt number, and subscripts c and i refer to current and ideal values, respectively. Ideal results were obtained with the finest mesh and a higher computational cost during the mesh independence study. Such results were achieved with the longest exchanger and, also, with the highest consumption of computation time during the length independence study or flow development study.

**3. Results and discussion**

Initially, the heating of the fluid due to the increase in the average temperature was evaluated using a weighted average integral function at the outlet; it was verified that heat transfer and Dean number rise as mass flow grows. The improvement in the heat transfer is the result of the secondary flow formations identified by Dean in [9,10]. These effects become evident in Figure 3, where the plotted velocities are
tangential components at the outlet. The increase in heat transfer associated with a higher Dean number can be observed by comparing the magnitudes of the vector velocities in Figure 3(a) and Figure 3(b), which can be interpreted as more contact between fluid particles and the hot surface that transfers energy to them. Moreover, those magnitudes can explain the pressure drop along the heat exchanger, because some energy is necessary to produce this tangential velocity field. [11,12].

![Figure 3](image)

**Figure 3:** Tangential projection of speed vectors at the exchanger outlet with a (a) low and (b) high Dean number.

To verify the mesh independence for the calculated Nusselt number, we studied the latter’s behavior in the simulation process using different mesh densities. This density is defined as the ratio between the number of elements in the control volume and the length of the heat exchanger’s central path. That analysis was carried out with a velocity of 1 m/s, and the results are shown in Figure 4. The vertical axis indicates the average Nusselt number, and its horizontal counterpart shows the angular position inside the heat exchanger; finally, the colors represent the number of elements in the mesh. It can be observed that the Nusselt number barely changes from mesh to mesh. The mesh that contains 571136 elements was selected for the simulations. Figure 5 presents the behavior of the Nusselt number at three velocities: 0.1 m/s, 0.5 m/s, and 1 m/s. In those cases, the minimum length necessary to study heat exchangers is 360°, when the Nusselt number is almost constant.

![Figure 4](image)

**Figure 4.** Nusselt number as function of exchanger length and number of elements.
Figure 5. Nusselt number as function of mesh density and exchanger length with a low Dean number (0.1 m/s).

If we compare the results obtained with the high mesh densities of 0.5043 elements/mm$^3$ and 7.1681 elements/mm$^3$, the differences in the Nusselt number remain under 1%; however, the computation time increases between 40% and 75% with the densest mesh. Therefore, the optimal mesh density to analyze a heat exchanger in the cases under study is 0.5043 elements/mm$^3$. This parameter can be a reference value for meshes of different lengths or types of heat exchanger. Nevertheless, mesh independence must be verified in each case to validate the quality of the results.

4. Conclusions

This study was able to establish a minimum length and mesh density for the heat exchangers simulated in this work. The results of the analysis can be used to infer same parameters in other cases, in order to reduce the computational cost and maintain the balance between desired accuracy and available computational resources. The ideal conditions established in the simulations were a two-spin length and a mesh density of 0.5043 elements/mm$^3$, which constitute a starting point to continue the process of numerical study of double passive improvement in heat exchangers. Fluid heating, the effects of the mass flow increase on outlet temperature, and Dean and Nusselt numbers (which characterize heat transfer in the device) were verified. The increase in the Nusselt number associated with the rise in the Dean number could establish the base for future works on the development of correlations. Finally, since secondary flows and vortex formations have been identified in similar systems, the use of passive methods to improve the performance of heat exchangers was verified and justified.

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