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Numerical Study of the Effect of Passive Techniques in Tube-In-Tube Helical Heat Exchanger

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This research presents a Computational Fluid Dynamic (CFD) study of a tube-in-tube helical heat exchanger evaluating two passive technique: i) ridges and ii) torsion in the internal tube. The effects of the fluid flow rate on the heat transfer were studied in the internal and annular flow. A commercial CFD package was used to predict the flow and thermal development in a tube-in-tube helical heat exchanger. The simulations were carried out in counter-flow mode operation with hot fluid in the internal tube side and cold fluids in the annular flow. The internal tube was modified with a double passive technique to provide high turbulence in the outer region. The numerical results agree with the reported data, the use of one passive technique in the internal tube increases the heat transfer up to 28.8 % without torsion in the tube.

1. Introduction

According to Le et al. (2015), two types of techniques have been proposed to improve the heat transfer of heat exchangers: the active and the passive. The active techniques require external power, such as vibration or magnetic fields, whereas the passive techniques require deformations on the tube surface, without external power. The passive technique was widely recommended by several authors because it considers bent tube and its ability to compact the heat exchanger. Pan et al. (2014) described the heat transfer improvements assuming passive techniques, it produced by secondary flows and studied by Dean since 1927.
The simple improvement techniques consist of: the insertion of coils (Kurnia et al., 2015), twist tape (Le et al., 2015), staggered tapes in straight tubes, the corrugation of the tube and the curved tube (García et al., 2012). Another technique is the double improvement, which is a combination of two simple improvements (Rainieri et al., 2012), such as a curved tube with springs, corrugated or baffles inserts; or a straight tube with tape inserts and corrugated (Zachár, 2010).
This work focuses on the Computational Fluid Dynamic (CFD) study of a tube-in-tube helical heat exchanger evaluated with a passive technique. Therefore, the objective of this research is present the CFD results to help design engineers in the field of heat exchangers.
The effects of the fluid flow rate on the heat transfer were studied in the annular flow (flow generated between the tubes). Previous research emphasizes the CFD study in the heat exchangers with the aim of validating the numerical results and optimization (Salpingidou et al., 2016) and control strategies of tubular heat exchangers using neural-networks-based method (Bakosova et al., 2017).
This work has been organized as follows: first, the simulation code was developed, and it was carefully verified, whenever possible, with numerical results presented by Kumar et al. (2006). Second, the simulation of the heat exchanger with four ridges in the internal tube, and the simulation of three heat exchangers increasing the number of twists in the internal tube from one to five. Finally, the Nusselt number was calculated for each case with the aim of assessing the effect of each passive technique on the heat transfer.

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2. Methodology

The heat exchanger model consists of two helical concentric tubes, tube-in-tube, according to dimensions and suggestions by Kumar et al. (2006). The hot fluid flows in the internal tube and the cold fluid flows in the annular section in counter-flow using water as working fluid in both flows. The simulation was carried out with ANSYS CFX 17 in a computer of 16 core, 32 GB RAM at the Modelling Laboratory of Materiales Avanzados y Energía (MATyER) research group of Instituto Tecnológico Metropolitano at Medellín-Colombia.

2.1 Modelling of heat exchanger

A concentric tube-in-tube heat exchanger was modeled as shown in Figure 1 (A), considering dimensions and boundary conditions given in Table 1. As can be seen in Figure 1 (B), four ridges were aggregated in the internal tube conserving it hydraulic diameter. The internal tube was modified increasing the torsion, Figures 1 (C), (D) and (E) illustrate the heat exchanger with one, three and five turns, respectively.

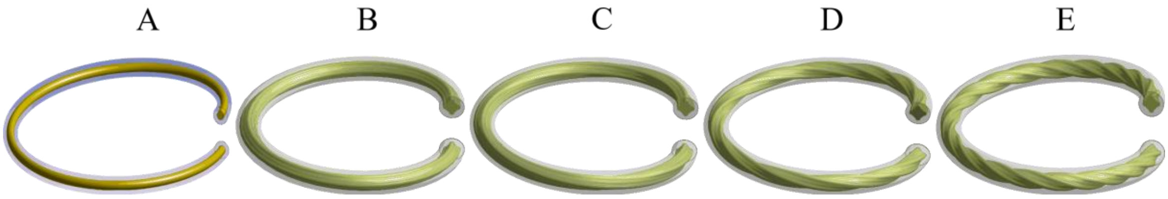

Figure 1: Geometries modelled for simulations

Table 1: Geometry and boundary conditions

	Internal tube	External tube
Outer diameter (m)	0.0254	0.0508
Helical diameter (m)	0.762	0.762
Pitch (m)	0.1	0.1
Velocity (m/s)	0.073	0.32-0.44
Dean number (dimensionless)	1000	4410-6030
Prandtl number (dimensionless)	7	7

The mesh was generated for five heat exchanger geometries described above with element size of 1 mm. Several meshes were created with the aim of comparing the Nusselt number and selecting adequate mesh for simulations. The error between two mesh sizes was calculated according to $\frac{|Nu_U - Nu_L|}{Nu_U} \times 100$, where Nu_U and Nu_L mean the Nusselt number for upper mesh and lower mesh, respectively. According to the numerical results, the number of elements of each mesh was selected between 40 x10^6 and 57x10^6 with the objective of showing independent results of the mesh and errors lower than 1 %.

2.2 Numerical computation

Five Dean numbers were simulated: 4,410, 4,880, 5,340, 5,810 and 6,030 for an external tube keeping constant the design and increasing the Reynolds number. The inner tube had a constant Dean number of 1000 for all simulations. The inlet and outlet temperatures in the internal tube were fixed at 300 K and 380 K, respectively. The numerical computation was set up at convergence criterion less than 10^{-6} and the total number of iterations were varied from 500 to 600. The Dean number for the external tube was calculated as $De = \frac{\rho v D_h}{\mu\sqrt{\delta}}$, where-$D_h = \frac{4A}{P_h}$ is the hydraulic diameter, $\delta = \frac{D_c}{D_i}$ is the curvature diameter, ρ is the density and μ is the absolute viscosity, A is the cross-sectional area and P_h is the hydraulic diameter.
Equations suggested by Zachár et al. (2010) for physical-thermal properties of water as density, specific heat, thermal conductivity and dynamic viscosity were programmed and used in this research.
The differential equations governing the turbulent flow that describe the chaotic movement of the fluid inside heat exchanger can be written in the tensor form according to Eq(1-8).
Continuity balance:

$$\frac{\partial \rho u_i}{\partial x_i} = 0 \quad (1)$$

Momentum balance equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} + \frac{\partial P}{\partial x_i} - \frac{\partial}{\partial x_j}\left[(\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] - F_i = 0 \quad (2)$$

Energy balance equation:

$$\frac{\partial(\rho E u_j)}{\partial x_j} + \frac{\partial(P u_j)}{\partial x_j} - \Phi - \frac{\partial}{\partial x_j} k\left(\frac{\partial T}{\partial x_j}\right) - s_E = 0 \quad (3)$$

Turbulent kinetic energy:

$$\frac{\partial(\rho u_j k)}{\partial x_j} - \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] - P_k + \rho\varepsilon = 0 \quad (4)$$

Turbulent dissipation energy:

$$\frac{\partial(\rho u_j \varepsilon)}{\partial x_j} - \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] - \frac{\varepsilon}{k}(C_{\varepsilon 1} P_k - C_{\varepsilon 2}\rho\varepsilon) = 0 \quad (5)$$

Where Φ is the viscous heating term in energy equation represented by Eq(6).

$$\Phi = \mu \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \quad (6)$$

And P_k is the parameter to calculate the generation of turbulent kinetic energy due to the mean velocity gradient and is represented by Eq(7).

$$P_k = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\frac{\partial u_i}{\partial x_j} \quad (7)$$

The $k - \varepsilon$ model assumes that the turbulence viscosity is coupled to the governing equations via the relation in Eq(8).

$$\mu_t = C_\mu \rho \frac{k^2}{\varepsilon} \quad (8)$$

The empirical constants for the turbulence model are assigned the following values: C_μ = 0.09, $C_{\varepsilon 1}$ = 1.47, $C_{\varepsilon 2}$ = 1.92, σ_k = 1.0 and σ_ε = 1.3.

3. Results and discussion

The computer model has been carefully verified using, whenever possible, the numerical results of Kumar et al. (2006).

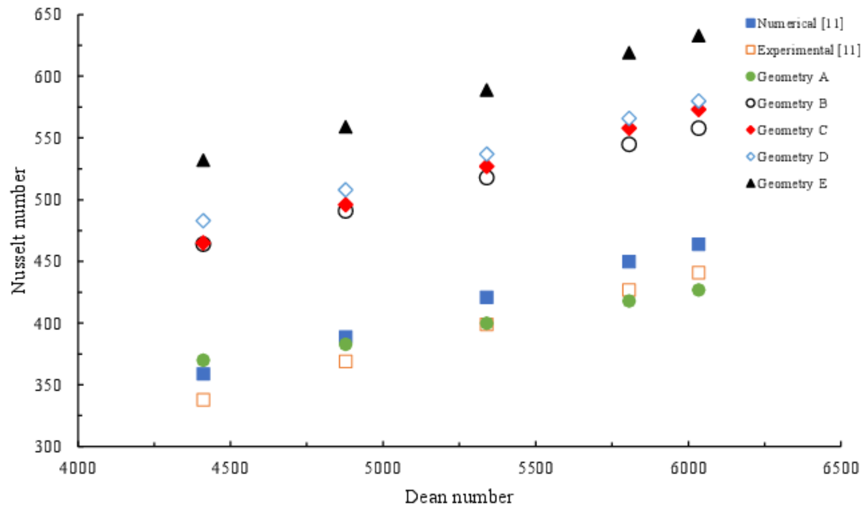

Figure 2: Nusselt number vs Dean number for outer tube

Figure 2 shows the Nusselt number reported by Kumar et al. (2006) and our numerical results for tube-in-tube heat exchanger without passive technique improvements (Geometry A). As can be seen, the model agrees with the numerical and experimental data reported by Kumar et al. (2006). The geometry B includes the passive technique without turns into the internal tube, an increment of 28.8 % in the Nusselt number was calculated when the Dean number was increased. It is interesting to note that, the increases of the Nusselt number from one to three turns of torsion in the internal tube were minor to 3%, nevertheless, the biggest increase was observed when the number of turns changes from three to five in 9.7 %.

The velocity and temperature contours in the internal and annular flow were analyzed in this section considering a Dean number equal to 4,411 in the external fluid. Figure 3 illustrates the velocity contours for the internal fluid of five geometries of the heat exchanger. The geometries were arranged in rows, the rows (A), (B), (C), (D) and (E) show the velocity contours considering: a smooth tube, a geometry modified with passive technique without twist, a geometry modified with passive technique with one, three and five turns, respectively. The columns report different cross-section (30°, 60°, 90°, 180°, 270° and 360°). Arcs located to the left of every field indicate the exterior of the heat exchanger.

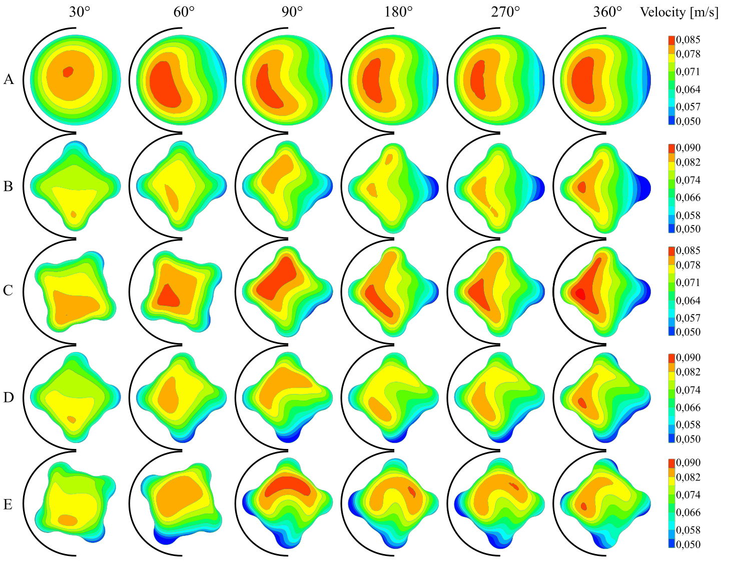

Figure 3: Velocity Contours for the inner fluid of five heat exchangers with Dean number 1,000 for the inner tube and with Dean number 4,411 for the outer tube

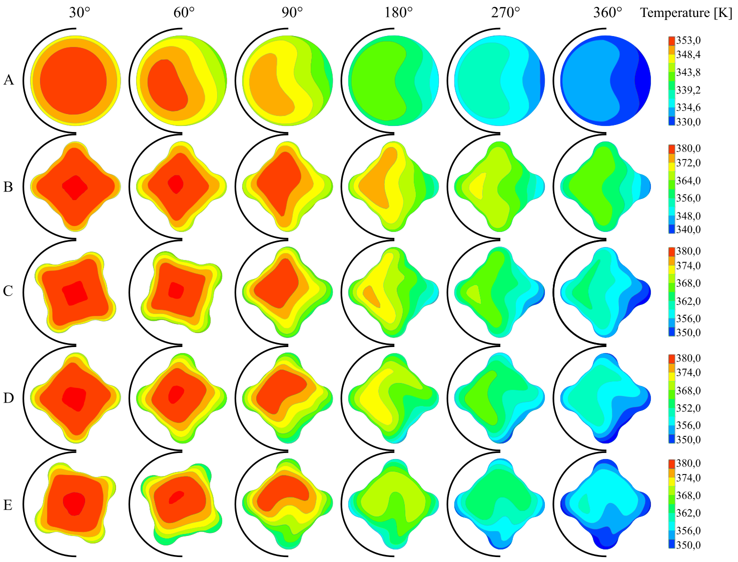

Figure 4: Temperature Contours for the inner fluid with Dean number 1,000 for inner tube and with Dean number 4,411 for outer tube

As can be seen in the numerical results of model (A), the velocity contours practically remain the same. This behavior is similar to results reported by Kumar (2006). In four geometries (B)-(E), the velocity contour changed from the position of 90° appreciating different velocity profiles, however for angles from 180° to 360° small variations are observed in the velocity fields, this is evidence of total developed flow. For the models (A)-(D), the maximum velocity was observed towards the outside of heat exchangers, this effect is attributable to the centrifugal force and secondary flows caused by the curvature of the helical coil, the effect of these forces over the velocity is the same that reported by Dean (1927). As can be seen, for the numerical results of models (D) and (E) it is not possible to observe a pattern in the velocity contour possibly attributed to the effect of twist along the tube.

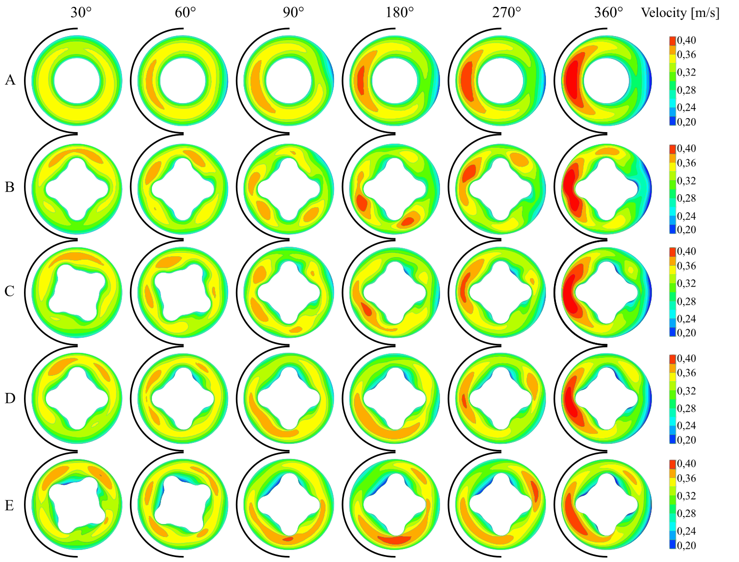

Figure 5: Velocity Contours fields for the outer fluid with Dean number 1,000 for inner tube and with Dean number 4,411 for outer tube

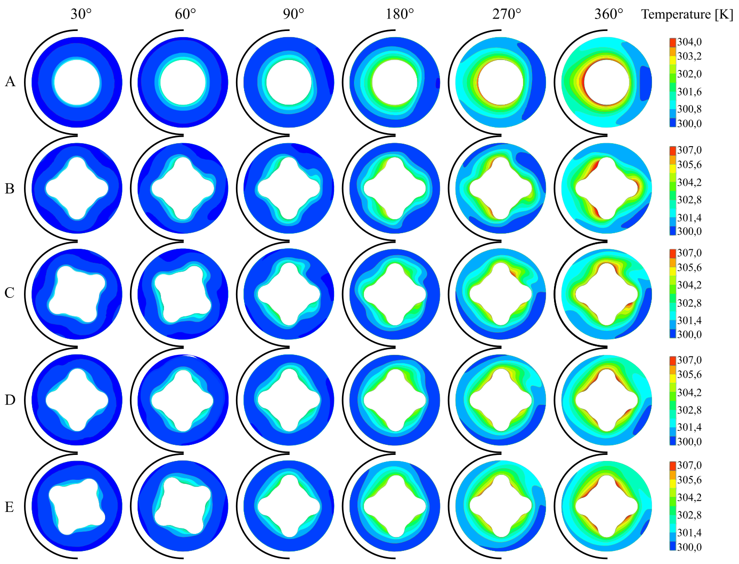

Figure 6: Temperature fields for the outer fluid with Dean number 1,000 for inner tube and with Dean number 4,411 for outer tube

Figure 4 shows the temperature contours for the five models described previously. In geometries A-D the maximum temperature was observed towards the exterior of the heat exchanger, this is due to the center of the vortex formed by the secondary flows. In addition, the movement of fluid in the form of vortex produces an increase in the heat transfer mechanism, possibly caused for the increase of velocity contours in these sections (see Figure 3). For the models (D) and (E), the temperature contours were very similar from 30° to 360°, this

behavior may be the reason why the Nusselt number presents an insignificant increase when the torsion increases from three to five (Figure 3).
Figure 5 shows the velocity contour for annular flow. For models from (B) to (E), a pattern was not appreciated in velocity fields, small variations were observed in the velocity fields from 270° to 360°, but this is not enough to affirm that there is a developed flow. The secondary flows observable in models from (B) to (E) were more turbulent than model (A), the passive technique benefits the turbulence and consequently the heat transfer.
Figure 6 illustrates the temperature contour for annular flow. For models from (B) to (E), the temperature increases uniformly, there are no significant temperature differences between the models in the cross-section selected. The temperature contours correlate with the velocity contours, as previously showed in Figure 5.

4. Conclusions

The numerical model using Computational Fluid Dynamic of a helical tube-in-tube heat exchanger with and without passive techniques was successfully carried out. The main contributions of the present research are the following:

- The numerical model of Kumar et al. (2006) was reproduced considering the assumptions and experimental information described in that research. This evidence gives confidence about the evaluation when the passive techniques were added.
- With reference to the first passive technique applied the addiction of four ridges in the inner tube. Shows an increment up to 28.8 % in the Nusselt number were calculated for all cases under study. Then, when the Dean number was increased from 4,500 to 6,000 the Nusselt number increases linearly. This increment can be caused by the velocity contours generated by the addiction of ridges and its influence on heat transfer. In the annular section of heat exchanger, the ridges decrease the centrifugal forces generated by the action of a helical coil.
- When the torsion of the internal tube was added from one to three turns, an increase up to 3 % in the Nusselt number was calculated. The biggest increase, up to 9 % was calculated when nine turns were simulated.
- The numerical results of this research will be considered to design other passive techniques without torsion in tube. The previous suggestion is supported by the increase in the heat transfer and without compromising the mechanical integrity of the tubes caused by the torsion.

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