Numerical and experimental study of local heat transfer enhancement in helically coiled pipes. Preliminary results.

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Abstract In the last years, the attention of heat transfer equipments manufacturers turned toward helically coiled-tube heat exchangers, especially with regards to applications for viscous and/or particulate products. The recent progress achieved in numerical simulation motivated many research groups to develop numerical models for this kind of apparatuses. These models, intended both to improve the knowledge of the fundamental heat transfer mechanisms in curved geometries and to support the industrial design of this kind of apparatuses, are usually validated throughout the comparison with either theoretical or experimental evidences by considering average heat transfer performances. However, this approach doesn't guarantee that the validated models are able to reproduce local effects in details, which are so important in this kind of non-standard geometries. In the present paper a numerical model of convective heat transfer in coiled tubes for laminar flow regime was formulated and discussed. Its goodness was checked throughout the comparison with the latest experimental outcomes of Bozzoli et al. [1] in terms of convective heat flux distribution along the boundary of the duct, by ensuring the effectiveness of the model also in the description of local behaviours. Although the present paper reports only preliminary results of this simulation/validation process, it could be of interest for the research community because it proposes a novel approach that could be useful to validate many numerical models for non-standard geometries.

1. Introduction

Helically coiled-tube heat exchangers are used in several industrial applications because, for given operating conditions, heat transfer rate in this kind of devices is significantly larger than in straight pipes [1]. The effectiveness of wall curvature in enhancing thermal performances primarily occurs because it gives origin in the fluid to the centrifugal force that induces local maxima in the velocity distribution and, as a consequence, it locally increases the temperature gradients at the wall [2]. This kind of passive heat transfer enhancement is particularly interesting when highly viscous fluids are treated because, with these fluids, the laminar flow regime is usually encountered and convective heat transfer rate is limited.

In the last years the attention of heat transfer equipment manufacturers turned toward helically coiled-tube heat exchangers and the progress in the numerical simulation have motivated many research groups to develop numerical models for this kind of apparatuses. These models are intended both to improve the knowledge of the fundamental heat transfer mechanisms in curved geometries and to

support the industrial design of this kind of apparatuses. Some representative examples of numerical models for the laminar fluid flow in curved tubes are proposed by Mitsunobu et al [3], Kozo and Yoshiyuki [4] Yang et al [5] while the turbulent flow regime was deeply investigated by Di Piazza and Ciofalo [6] and Di Liberto and Ciofalo [7].

In [3] a finite-difference solution using a combination of line iterative method and boundary vorticity method is presented for the hydrodynamically and thermally fully developed laminar forced convection in curved pipes. The method was applied to single out the effect of Dean number on local Nusselt for two different fluids having Prandtl number equal to 0.7 and 100 respectively.

Kozo and Yoshiyuki [4] carried out a theoretical study on the effect of the secondary flow on heat transfer in uniformly heated helically coiled tube. Both the centrifugal and buoyancy forces are taken into account in the numerical analysis. The solutions cover a range of Prandtl numbers between 1 and 100. The velocity and temperature profiles, the local friction factor and heat transfer coefficient distribution are obtained.

Yang et al. [5] presented a numerical investigation on the fully developed laminar convective heat transfer in a helicoidal pipe, with particular attention to the effects of torsion on the local heat-transfer coefficient. In particular, the authors reported the Nusselt number distribution varying the coil pitch, and they showed that, due to torsion, the local heat-transfer coefficient, compared to the case of an ideal torus, is increased on half of the tube wall while it is decreased on the other half.

Di Piazza and Ciofalo [6] analysed the heat transfer in turbulent flow regime in curved pipes with curvature ratio, defined as the ratio between the pipe radius and the coil radius, ranging from 0.03 to 0.3. The authors applied to the problem the code ANSYS CFX that employs a coupled technique that, at the same time solves all the transport equations in the whole domain through a false time-step algorithm. The domain under numerical study was a short segment of toroidal pipe of circular cross section. The Authors employed and compared different turbulence models as the standard k-ε with wall functions, the SST k-ω and a second-order RSM. The study was performed by varying the Reynolds number and the Prandtl number in the range 1.4×10^4- 8×10^4 and 0.7-5.6 respectively.

Di Liberto and Ciofalo [7] studied the turbulent heat transfer in curved ducts by numerical simulation considering two different curvature ratios (0.1 and 0.3). The simulations were performed at Prandtl number equal to 0.86 and at Reynolds number varying between 12630 and 17350.

It is universally accepted that numerical models have scientific relevance only if they are validated by the comparison with theoretical or experimental evidences, but, in some cases, the applications of this principle can be ambiguous. Regarding heat transfer models of coiled tubes, they are usually validated only considering average heat transfer performances. However, this approach doesn't guarantee that the so validated models are able to reproduce local effects in details. Moreover most of the available studies regard low Prandtl number fluids, while the experimental data frequently refer to medium and highly viscous fluids. Since the flow velocity and temperature fields in coiled ducts are strongly asymmetrical over the cross-section of the tube, local effects have to be accounted for when a numerical model has to be developed. Moreover, local behavior can affect the performances of many industrial processes. For instance, in food pasteurisation, the irregular temperature field induced by the wall curvature could reduce the bacteria heat-killing or could locally overheat the product. On the other hand, the scarcity of available experimental data regarding local quantities makes this kind of model validation difficult to be performed. Regarding coiled heat exchangers, most of the papers available in the scientific literature did not investigate local thermal behavior, mainly due to the practical difficulty of measuring heat flux on the internal wall surface of a pipe, and they presented the results only in terms of the Nusselt number averaged along the wall circumference. To the Authors's knowledge, only five papers [8-12] present experimental results in terms of local Nusselt number.

Bai et al. [8] experimentally studied the turbulent heat transfer in helically coiled tubes using deionised water as the working fluid. The working fluid was heated by applying alternating current in the tube wall and, in each cross section eight thermocouples were placed on the external surface of the tube wall. The local heat-transfer distribution on the internal wall of the tube was found by solving the two-dimensional inverse heat conduction problem whit the least-square method. As expected, they

found that the local heat-transfer coefficient was not evenly distributed along the periphery of the cross section and that, in particular, at the outside surface of the coil, it was three or four times higher than that at the inside surface.

Bozzoli et al. [9] focused their investigation on the thermally fully developed region for the laminar flow regime in the Reynolds number range of 135 to 1050 and in the Prandtl number range of 170 to 200. The temperature distribution maps on the external coil wall were employed as input data of the linear inverse heat conduction problem in the wall under a solution approach based on the Tikhonov regularisation method with the support of the fixed-point iteration technique to determine the proper regularisation parameter. The results showed that, at the outside surface of the coil, the Nusselt number is approximately five times larger than that at the inside surface and this ratio, in the conditions under test, is constant.

Regarding local heat-transfer coefficient, some experimental were discussed by Seban et al. [10] investigating the laminar flow of oil and the turbulent flow of water in tubes coils. These Authors correctly drew the attention to the difference between apparent and true local values: apparent heat transfer coefficient are obtained neglecting the circumferential heat conduction in the tube wall, that means considering the average value of the convective heat flux instead of the punctual value. In terms of true heat transfer coefficient, the ratio of the outside to the inside coefficient found in this experimental campaign was about four for both the laminar flow case and the turbulent one. However, no details about the approach adopted to estimate the punctual convective heat flux were given in this paper.

Both Xin and Ebadian [11] and Janssen and Hoogendoorn [12] developed an extensive experimental campaign with many different fluids on a wide range of curvature ratios and Reynolds numbers. However, these Authors processed their data neglecting the circumferential heat conduction in the tube wall, therefore the reported local heat-transfer coefficient values were the apparent ones and not the real ones.

In the present paper a numerical model of convective heat transfer in coiled tubes for laminar flow regime was formulated and discussed. The validation process was developed through the comparison of the numerical results with the latest experimental outcomes of Bozzoli et al. [1] in terms of convective heat flux distribution along the boundary of the duct. This procedure verifies the robustness of the model in the description of local effects in typical thermal treatments of medium viscous fluids .

2. Mathematical formulation

This section provides the basic equations that must be solved to describe the velocity field and the temperature distribution inside the coiled heat exchanger and the temperature distribution in the coil wall. The physical problem is a time independent three dimensional heat and fluid flow process.

The momentum equation of the fluid inside the heat exchanger coil is expressed by the three dimensional Navier-Stokes equations and the conductive heat transport in the wall of the coil is described by the Laplace equation. A SIMPLE like method is applied to solve the momentum and continuity equations. The dependent variables that describe the present problem are the temperature T, for the fluid and solid wall regions of the studied problem, velocity components U_i in the x_1, x_2 and x_3 directions respectively and the pressure field P.

2.1. Equations

The following set of partial differential equations for U_1, U_2, U_3, P and T as a function of x_1, x_2, x_3 describes the flow and temperature field inside a helically coiled heat exchanger. The conservation equations are formulated in the Cartesian coordinate system because the applied flow solver (Ansys CFX 11.0) uses the Cartesian system to formulate the conservation equations for all quantities (vector U_i and scalar T, P). Temperature dependency of the different physical properties (ρ, c_p, η, λ) of the working fluid are neglected.

In the fluid domain the continuity equation is formulated in the following manner in Cartesian coordinate system:

$$\frac{\partial U_1}{\partial x_1} + \frac{\partial U_2}{\partial x_2} + \frac{\partial U_3}{\partial x_3} = 0. \tag{1}$$

The following equation system is the formulation of the momentum equations in Cartesian coordinate system where $i \in \{1,2,3\}$:

$$\rho\left(U_1 \frac{\partial U_i}{\partial x_1} + U_2 \frac{\partial U_i}{\partial x_2} + U_3 \frac{\partial U_i}{\partial x_3}\right) = -\frac{\partial P}{\partial x_i} + \eta\left(\frac{\partial^2 U_i}{\partial x_1^2} + \frac{\partial^2 U_i}{\partial x_2^2} + \frac{\partial^2 U_i}{\partial x_3^2}\right) \tag{2}$$

η is the dynamic viscosity and ρ is the density of the working fluid.
The energy equation, under the assumption of negligible viscous dissipation, is then solved to calculate the temperature field of the studied flow process for the fluid domain

$$\rho\, c_P\left(U_1 \frac{\partial T}{\partial x_1} + U_2 \frac{\partial T}{\partial x_2} + U_3 \frac{\partial T}{\partial x_3}\right) = \lambda\left(\frac{\partial^2 T}{\partial x_1^2} + \frac{\partial^2 T}{\partial x_2^2} + \frac{\partial^2 T}{\partial x_3^2}\right), \tag{3}$$

where λ and c_p are the thermal conductivity and the specific heat at constant pressure of the fluid respectively. The transport equations have been formulated in a conservative form which is more suitable for numerical treatment.

In the solid domain the steady state energy balance equation is expressed as follows:

$$\left(\frac{\partial^2 T}{\partial x_1^2}\right) + \left(\frac{\partial^2 T}{\partial x_2^2}\right) + \left(\frac{\partial^2 T}{\partial x_3^2}\right) + \frac{q_g}{k} = 0 \tag{4}$$

where q_g is the heat generated by Joule effect in the wall, k is the wall thermal conductivity. The solid and the fluid domain are coupled by imposing continuity of temperature and heat flux on the interior wall boundary.

A constant mass flow rate is assumed at the inlet section of the coiled tube. The gradient of the velocity profile and of the temperature field is assumed to be zero at the outlet section. An adiabatic boundary condition (fully insulated) is specified at the outer wall of the helical tube.

2.2. Domain of discretization
The domain of the numerical computation consists of two separate calculation domains: the fluid and the solid pipe wall domain. The geometry has been drawn separately with 3D geometry modeler software (Inventor 11), as shown in figure 2. The developed geometry has been imported into the ICEM CFD software to generate the numerical grids and, after that, the generated grids have been imported by the CFX task preparation environment. A solid-fluid domain interface has been used to connect the fluid and the solid region of the investigated heat and fluid flow problem. So called prismatic cells have been used on the inner surface of the helical pipe and on the near wall region of the fluid domain. Prismatic cells are generally used to discretize boundary layers of different kind of fluid flow problems near solid surfaces. A prismatic cell is a specially constructed finite volume element that is similar to a thin triangular plate. The reason of using this kind of volume element type is to resolve the rapid changing of the normal directional values of the flow variables near the solid surface in the boundary layer.

2.4. Numerical solution of the transport equations
The equations with the appropriate boundary conditions have been solved with a commercially available CFD code (Ansys CFX 11.0). A "High Resolution Up-Wind like" scheme is used to discretize the convection term in the transport equations. The resulting large linear set of equations is solved with an algebraic multi-grid solver.

3. Model validation
The validation process was developed throughout the comparison of the numerical results to the latest experimental outcomes of Bozzoli et al. [1] in terms of convective heat flux distribution along the boundary of the duct. This approach ensures the robustness of the model in the description of local effects.

Bozzoli et al. in [1] tested a helically coiled stainless steel type AISI 304 tube in terms of convective heat flux and heat transfer coefficient distribution along the boundary of the duct. The tube had smooth wall and it was characterised by eight coils following a helical profile along the axis of the tube. The tube internal diameter was 14 mm, and the wall thickness measured 1.0 mm, while the helix diameter and the pitch were approximately 310 mm and 200 mm, respectively.

The working fluid (i.e. ethylene glycol) entered the coiled test section equipped with stainless-steel fin electrodes, which were connected to a power supply. This setup allowed the investigation of the heat transfer performance of the tube under the prescribed condition of uniform heat flux generated by the Joule effect in the wall.

To minimise the heat exchange with the environment, the heated section was thermally insulated. A small portion of the external tube wall, near the downstream region of the heated section, was made accessible to an infrared imaging camera by removing the thermally insulating layer, and it was coated by a thin film of opaque paint of uniform and known emissivity. Therefore, the test section was taken approximately 9 m downstream the inlet section, in the region of the heated section where the laminar boundary layers reached the asymptotic profiles. This condition makes the results obtained for this particular section representative of the thermally fully developed region.
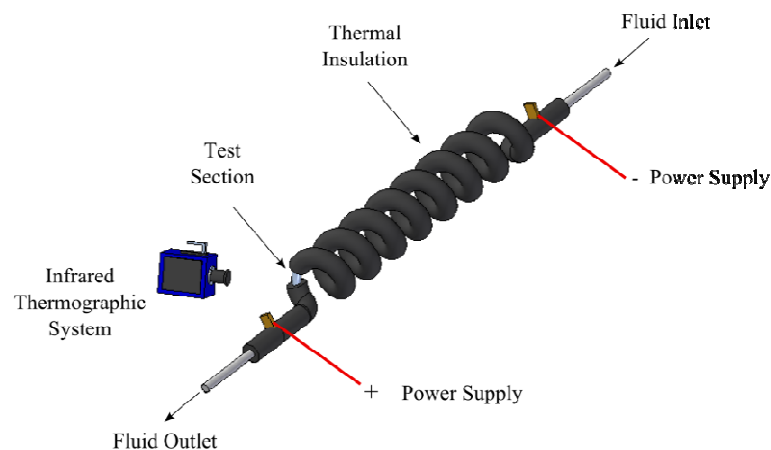

Figure 1. Sketch of the experimental setup.

The surface temperature distribution was acquired by means of a FLIR SC7000 infrared camera. The inlet and the outlet fluid bulk temperatures were measured with type-T thermocouples. The bulk temperature at any location in the heat transfer section was then calculated from the power supplied to the tube wall. The temperature distribution maps on the external coil wall were employed as input data of the linear inverse heat conduction problem in the wall under a solution approach based on the

Tikhonov regularisation method. This procedure is allowed to estimate the local convective wall heat flux and the local convective heat transfer coefficient along the boundary of the duct section.

The whole coiled tube was simulated in ANSYS environment by the numerical model above described. Figure 2 reports a view of the geometrical model of the investigated coiled tube. The applied grids, as shown in figure 3, are strongly non-uniform in order to resolve the wall boundary layers effects. Prismatic cells have been used on the inner surface of the helical pipe and on the near wall region of the helical fluid domain. Fifteen layers of prismatic cells have been generated with 1.15 increment factor to discretize the near wall boundary of the fluid domain inside the helical tube. The inner side boundary region of the solid tube wall (made of stainless steel) domain is created with five layers of prismatic cells. The total sum of finite volumes of the four separately generated grids is 43´664´780 finite volumes.
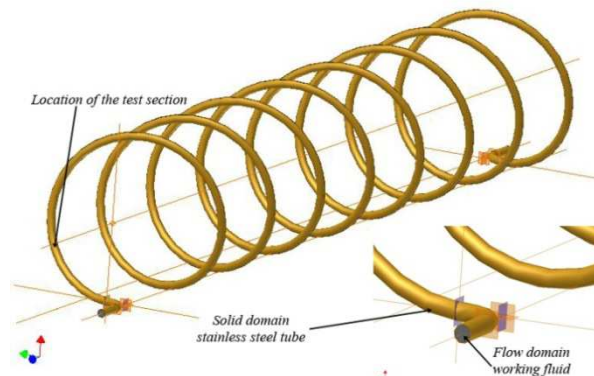

Figure 2. 3D geometrical model of the investigated coiled tube.
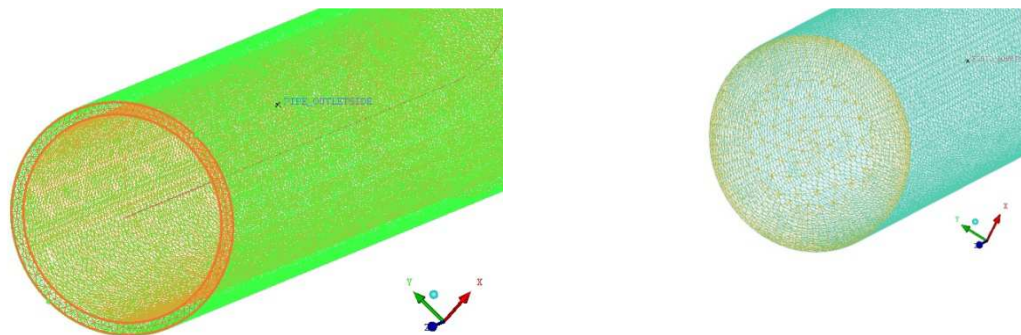

Figure 3. Generetated grid for the coiled tube wall (left) and for the working fluid (right)

To evaluate the local heat transfer performance of the coiled tube, 36 points were defined on the inner surface of the test section.
Numerical values of the local wall heat flux q_w and of the local wall temperature T_w around the test point are collected by sampling the corresponding field variables on each test points.

4. Results

The coiled tube was numerically simulated for two different test conditions for which local measurements are available in [1]. In particular Bozzoli et al. [1] investigated the laminar flow regime in the Reynolds number range of about 130–1100 and they concluded that, by accounting for the experimental uncertainty, for this range the profile of the local Nusselt number is almost independent of the Reynolds number. The two test conditions considered in this paper are representative of the

extreme conditions of the experimental data available in [1] and they correspond to Reynolds number values equal to 135 and 1006, respectively.

The fluid thermo-physical properties considered in the two simulations are reported in Table 1 and they correspond to the values published in [1] for the considered experimental conditions. In particular, they correspond to Prandtl number values of 173 and 190. Following the same approach the values of the heat generated in the wall are taken equal to the experimental values.

Table 1. Test conditions and working fluid thermo-physical properties considered in the numerical simulations.

Re	q_g (W/m^2)	λ (W/m·K)	ρ (kg/m^3)	η (Pa·s)	c_p (J/Kg·k)
135	$2.7 \cdot 10^6$	0.252	$1.12 \cdot 10^3$	0.0181	$2.41 \cdot 10^3$
1006	$4.8 \cdot 10^6$	0.251	$1.12 \cdot 10^3$	0.0199	$2.40 \cdot 10^3$

Mesh quality was fully investigated to evaluate the validity of the numerical results. Table 2 compares the results, reported in terms of the higher and the lower temperature values in the test section for different mesh distributions; the data confirm that the adopted mesh ensures a satisfactory solution.

Table 2. Mesh refinement study.

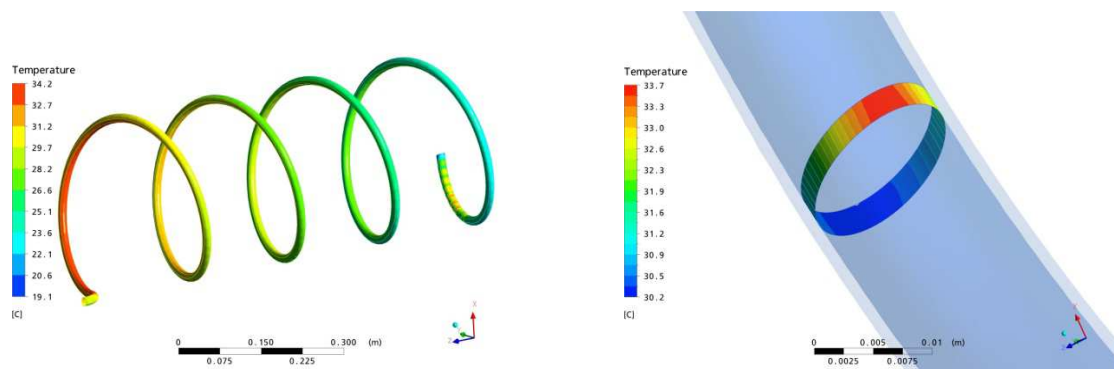|	Re=135		Re=1006	
No. of elements	23'758'136	43'664'780	23'758'136	43´664´780
$T_{w,max}$ **(K)**	307.1	307.0	302.5	302.1
$T_{w,min}$ **(K)**	302.3	302.3	297.5	297.3

Figure 4. Wall temperature for Re=135 .

Figure 4(left) shows the obtained outer side temperature field on the heated section of the coiled tube. The inner side peripheral temperature distribution is shown in figure 4(right) at the same test section. These data highlight that the temperature is not uniform along the periphery of the coil cross section and that, in particular, at the outside surface of the coil (extrados), it was lower than that at the inside surface (intrados).

In figure 5 the numerical results are compared to experimental ones in terms of heat flux density along the angular coordinate α, whose origin was taken at the extrados.

The data in figure 5 show that the numerical model works quite well in reconstruction of the heat flux at the fluid-wall interface: in particular it is possible to notice that for the case of Reynolds number equal to 1006 the two distributions, the numerical and the experimental one, are very close.

The matching of the two heat flux distributions is less accurate for Reynold number equal to 135: in particular the numerical model tends to overestimate the magnitude of the heat flux in the outside region of the coil. In any case the data confirm that the heat flux presents a great variation along the angular coordinate, by showing in particular values at the outside surface of the coil much higher than the ones at the inside surface as a consequence of the complex flow pattern that establishes due to the wall curvature.
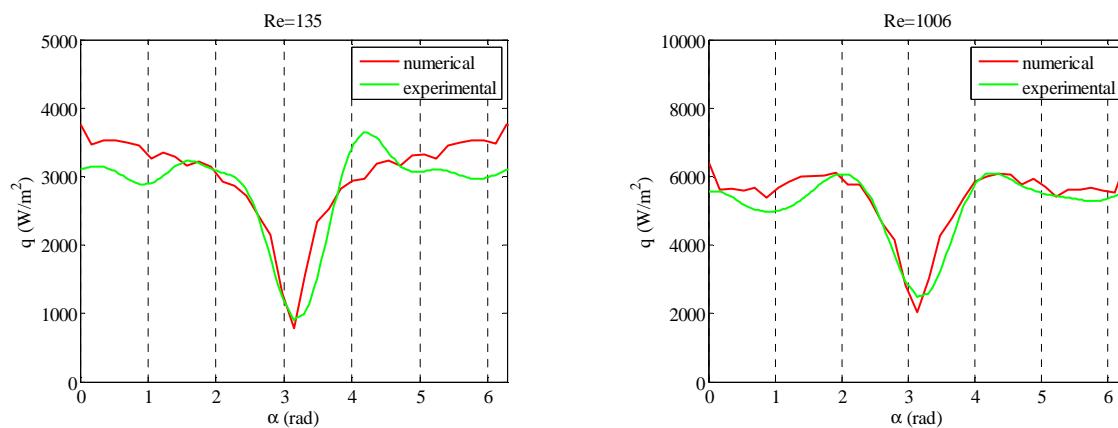

Figure 5. Comparison between experimental and numerical wall heat flux distributions.

In figure 6 the relative wall temperature, defined as the difference between the wall temperature and the fluid bulk temperature T_{bulk} , is reported.

These data show that for both Reynolds number values the numerical model underestimates the relative temperature value with respect to the one obtained experimentally.
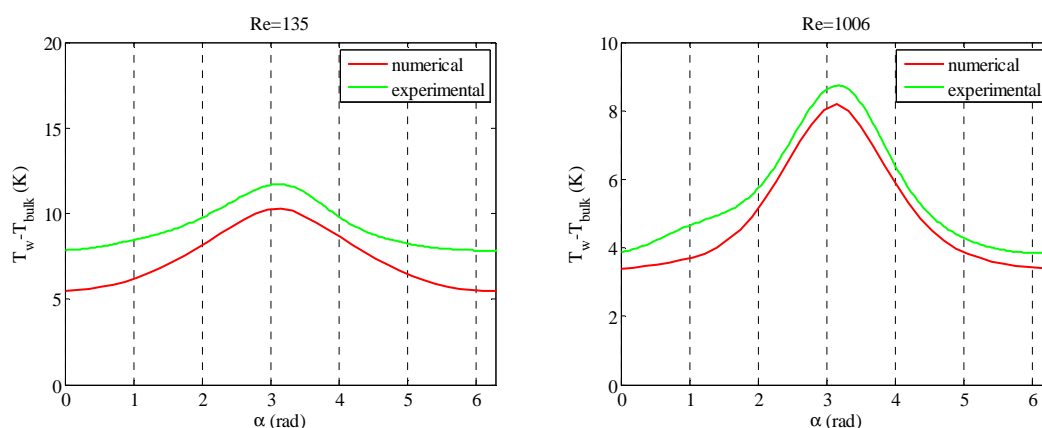

Figure 6. Comparison between experimental and numerical wall relative temperature distributions.

From the quantities obtained by the numerical simulation, the Nusselt number values were calculated as follows:

$$Nu = \frac{q \cdot D_{int}}{(T_w - T_{bulk}) \cdot \lambda_f}$$

(5)

where D_{int} is the tube internal diameter and λ_f is the fluid thermal conductivity, evaluated at the bulk temperature. Figure 7, reporting Nusselt number distributions, confirms that numerical model tends to overestimate the heat transfer in the outside region of the coil.
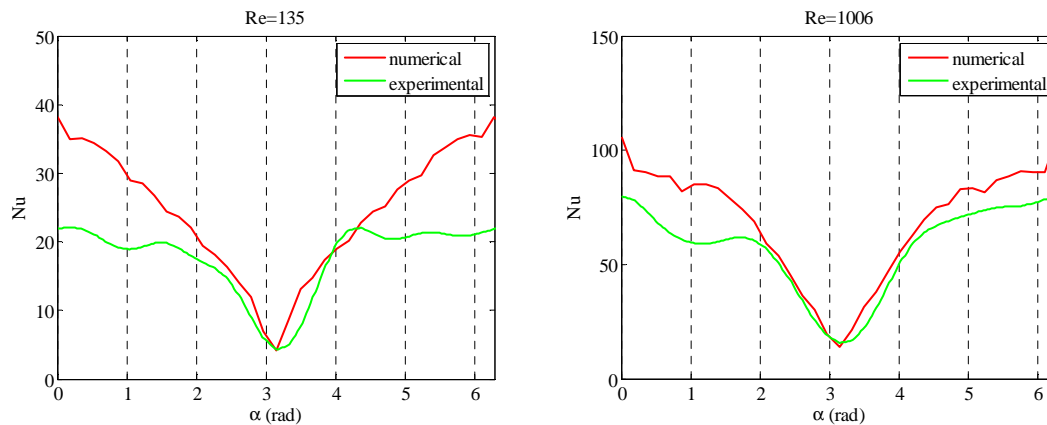

Figure 7. Comparison between experimental and numerical Nusselt number distributions.

Finally, the normalized numerical Nusselt number distribution was compared with the experimental results. The data, reported in figure (8) are in good agreement with the cinclusions of Bozzoli et al. [1]: in this flow regime the local Nusselt number (Nu/Nu_{max}) profile is almost independent on the Reynolds number and its minimum value, located at the inner coil bend, reaches about 0.2.
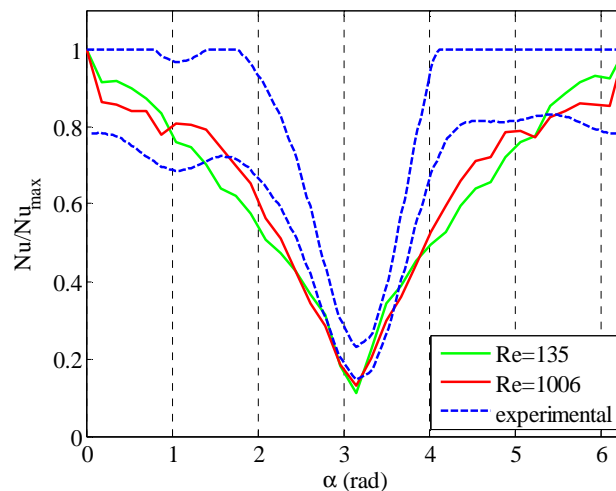

Figure 8. Normalized Nusselt number distribution: comparison between numerical and experimental (95% confidence interval) results.

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5. Conclusion

In the present paper a numerical model of convective heat transfer in coiled tubes under the laminar flow regime is presented and discussed with regard to its application to a high Prandtl number fluid. The numerical approach was checked throughout the comparison with the latest experimental outcomes of Bozzoli et al. [1] in terms of convective heat flux distribution along the boundary of the duct. The aim was to verify the effectiveness of the model in the description of the local heat transfer behavior. The preliminary results so far obtained show that the model works well in reconstruction of the heat flux at the fluid-wall interface, by confirming its significant variation along the angular coordinate as a consequence of the complex flow pattern that establishes due to the wall curvature.

The numerical relative temperature does not match the experimental distribution. Possible reasons of this mismatch could be found in the constant fluid properties approximation. In fact, although the fluid adopted in the experimental investigation shows properties that strongly vary with temperature, in the present study the simplified constant properties approach was adopted to limit the computational cost of the simulations. Moreover this simplified approach could affect the thermal development length in comparison to the one observed experimentally. Although this, the local normalized Nusselt number resulting from the numerical simulations is in good agreement with the experimental outcomes [1].

Further analysis is needed to deepen the conjugate heat transfer problem in coiled tubes, where the effect of the wall curvature produce a complex flow behavior. Furthermore, it has to be stressed that the present paper reports only preliminary results of this simulation/validation process but it could be of interest for the research community because it proposes a novel approach that could be useful to validate many numerical models for non-standard geometries.

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