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Experimental estimation of the local heat-transfer coefficient in coiled tubes in turbulent flow regime

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Abstract. Wall curvature is a popular heat transfer enhancement technique since it gives origin to the centrifugal force in the fluid: this phenomenon promotes local maxima in the velocity distribution that locally increase the temperature gradients at the wall by enhancing the heat transfer both in the laminar and in the turbulent flow regime. This geometry produces an asymmetrical distribution of the velocity field over the cross-section of the tube which lead to a significant variation in the convective heat-transfer coefficient along the circumferential angular coordinate: it presents higher values at the outer bend side of the wall surface than at the inner bend side. Although the irregular distribution of the heat transfer coefficient may be critical in some industrial applications, most of the authors did not investigate this aspect, mainly due to the practical difficulty of measuring heat flux on internal wall surface of a pipe. In the present investigation the local convective heat-transfer coefficient is experimentally estimated at the fluid-wall interface in coiled tubes when turbulent flow regime occurs; in particular, temperature distribution maps on the external coil wall are employed as input data of the inverse heat conduction problem in the wall and a solution approach based on the Tikhonov regularisation is implemented. The results, obtained with water as working fluid, are focused on the fully developed region in the turbulent flow regime in the Reynolds number range of 5000 to 12000.

1. Introduction

In order to save in materials and energy use, adopting techniques of heat transfer enhancement is mandatory in the design of commercial heat exchangers. Enhancement techniques can be separated into two categories: passive and active. Passive methods require no direct application of external power and they usually employ special surface geometries, which cause heat transfer enhancement.

On the other hand, active schemes (e.g. electromagnetic fields and surface vibration) require external power for operation [1].

Passive techniques are commercially more attractive because no power is required to facilitate the enhancement and among them treated surfaces, rough surfaces, displaced enhancement devices, swirl-flow devices, surface-tension devices, coiled tubes, or flow additives are found [2].

Wall curvature is one of the most frequently used passive techniques; its effectiveness occurs because it gives origin to the centrifugal force in the fluid: this phenomenon induces local maxima in the velocity distribution that locally increase the temperature gradients at the wall by maximising the heat transfer [3-5]. Dean [6] solved the simplified Navier–Stokes equations for a coiled pipe of small curvature showing that the flow is governed by the Dean number $De = Re \cdot \delta^{1/2}$, where Re is the Reynolds number and δ is the curvature ratio defined as the ratio of the pipe diameter to the coiling diameter. Both in the laminar and the turbulent flow regime the distributions of the velocity field over the cross-section of the tube are asymmetrical and they lead to a significant variation in the convective heat-transfer coefficient along the circumferential angular coordinate: it presents higher values at the outer bend side of the wall surface than at the inner bend side, [3,7,8].

The presence of an irregular distribution may be critical in some industrial applications, such as in those that involve a thermal process. 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. However, most of the papers available in the scientific literature did not investigate this aspect, mainly due to the practical difficulty of measuring heat flux on internal wall surface of a pipe, and they presented the results only in terms of the Nusselt number averaged along the wall circumference. Only a few authors have studied the phenomenon locally, and most of them have adopted the numerical approach. To the Authors's knowledge, only six papers [9-14] have presented experimental results and only three of them reports the real local values of the convective heat transfer coefficient [9-11] while the others, neglecting heat conduction in the tube wall, estimate only the apparent local values [12-14].

Bai et al. [9] 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 estimated by solving the two-dimensional inverse heat conduction problem with 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. [10] focused their investigation on the fully developed region for the laminar flow regime in the Reynolds number range of 135 to 1050 and 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 data are discussed by Seban et al. [11] investigating the laminar flow for 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 which 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 is about four for both the laminar and the turbulent flow case. However, no details about the approach adopted to estimate the punctual convective heat flux are given in this paper.

Xin and Ebadian [12], Janssen and Hoogendoorn [13] and Hadik et al. [14] conducted extensive experimental campaigns with many different fluids on a wide range of curvature ratios and Reynolds

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numbers. However, these Authors processed their data neglecting the circumferential heat conduction in the tube wall so the reported local heat-transfer coefficient are the apparent one and not the real one.

In the present paper, the estimation procedure presented in [10], is applied to estimate the local convective heat-transfer coefficient at the fluid-wall interface in coiled tubes investigating turbulent flow regime; the temperature distributions on the external wall of the coiled tube, which are acquired using the infrared technique, are adopted as input data of the inverse heat conduction problem in the wall of the tube.

The purpose of this paper is presenting results which are representative of a wide range of technical applications; this data could be employed both as a useful benchmark for CFD results as well as in the design of coiled tube apparatuses.

2. Experimental setup

Two different helically coiled stainless steel type AISI 304 tubes were tested. They had smooth wall and they were 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. The helix diameter was of approximately 310 mm and 425 mm, respectively while the pitch was about 200 mm for both the pipes. This geometry yields a coiled pipe length L of approximately 8 and 10 m, respectively and a dimensionless curvature δ of 0.045 and 0.032, respectively.

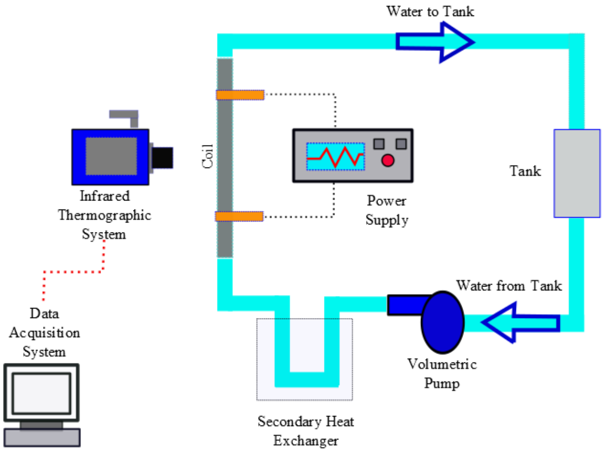

Figure 1. Sketch of the experimental setup.

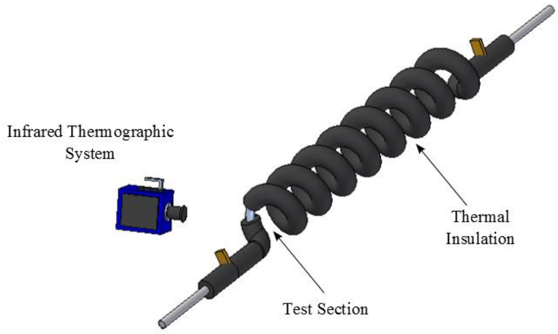

Figure 2. Particular of the test section.

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 sufficiently far from the inlet section, in the region of the heated section where, according to [8,13], the turbulent boundary layers reached the asymptotic profiles. This condition makes the results obtained for this particular segment representative of the thermally fully developed region.

The surface temperature distribution was acquired by means of a FLIR SC7000 unit, with a 640 x 512 pixel detector array. A sketch of the experimental setup is reported in figure 1, and figure 2 shows a particular of the text section.

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. Volumetric flow rates were obtained by measuring the time needed to fill a volumetric flask placed at the outlet of the test section.

To investigate the heat transfer performance of coiled tubes in the turbulent flow regime, water was used as the working fluid. It is currently accepted that the effect of coil curvature is to suppress turbulent fluctuations arising in the flowing fluid, smoothing the emergence of turbulence and increasing the value of the Reynolds number required to attain a fully turbulent flow, with respect to a straight pipe [15]. Considering that the flow in curved pipes remains laminar up to Reynolds numbers higher at least by a factor of two than in straight pipes, in order to be sure that the flow regime was turbulent, in the present investigation the Reynolds number range 5000−12000 was considered. In the temperature range characterising the experimental conditions, the Prandtl number of the working fluid varied in the range of 5-9.

The working fluid was conveyed by a volumetric pump to an holding tank, and it entered the coiled test section equipped with stainless-steel fin electrodes, which were connected to a power supply, type HP 6671A. This setup allowed 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. The heat flux provided to the fluid was selected to make the buoyancy forces negligible compared to inertial ones for the fluid velocity values investigated here. The coiled section was inserted horizontally in a loop completed by a secondary heat exchanger, fed with city water, to keep the working fluid temperature constant at the coil inlet.

3. Estimation procedure

The procedure, presented in [10], is here adopted to estimate the local convective heat-transfer coefficient in the coiled tubes under test. The temperature distribution maps on the external coil wall are 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.

This estimation procedure is based on a simplified 2-D model of the test section (sketched in figure 3) formulated by assuming that the temperature gradient is almost negligible along the axis of the tube.

In the 2-D solid domain, the steady-state energy balance equation is expressed in the (r,α) coordinate system in the form:

$$\frac{k}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)+\frac{k}{r^2}\frac{\partial^2 T}{\partial \alpha^2}+q_g=0 \tag{1}$$

where q_g is the heat generated by the Joule effect in the wall, k is the wall thermal conductivity and α is the angular coordinate.

The following two boundary conditions completed the energy balance equation:

$$k\frac{\partial T}{\partial r}=\frac{\left(T_{env}-T\right)}{R_{env}} \tag{2}$$

which is applied on surface S_2 and where R_{env} is the overall heat-transfer resistance between the tube wall and the surrounding environment with temperature T_{env};

$$-k\frac{\partial T}{\partial r}=q(\alpha) \tag{3}$$

which is applied on surface S_1 and where q is the local convective heat flux at the fluid-internal wall interface, assumed to be varying with the angular coordinate α.

To express the problem in the discrete domain, the convective heat flux distribution can be simplified by considering that it is described by a continuous piecewise linear function. In this way, the heat flux distribution can be defined by the vector $\mathbf{q}=[q_1,q_2,q_3,\dots,q_n]^T$.

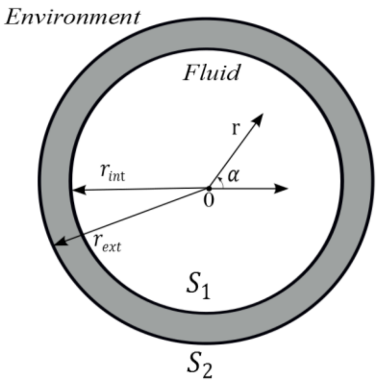

Figure 3. Geometrical domain with a coordinate system.

Proceeding this way the direct problem becomes linear with respect to the heat flux *q(α)* and its discrete version can be described as follows:

$$\mathbf{T} = \mathbf{Xq} + \mathbf{T_{q=0}}, \tag{4}$$

where **T** is the vector of the discrete temperature data at the external coil surface, **q** is the heat flux vector at the fluid-internal wall interface, $\mathbf{T_{q=0}}$ is a constant term and **X** is the sensitivity matrix.
The sensitivity matrix **X** was calculated using the two-point difference approach:

$$X_{i,j} = \frac{T_i\left(q_1, q_2, ..., q_j + \Delta q, ..., q_n\right) - T_i\left(q_1, q_2, ..., q_j, ..., q_n\right)}{\Delta q}, \tag{5}$$

where T_i is the temperature value at the *i* sensor position obtained by solving equations (1-3) with an internal heat flux distribution as defined in equation (5). In the same way, the constant term $\mathbf{T_{q=0}}$ was obtained by imposing a null internal heat flux. This set of equations was easily solved by the finite element method.

The direct formulation of the problem is concerned with the determination of the temperature distribution on the tube external wall when the convective heat flux vector **q** is known. In the inverse formulation considered here, **q** is instead regarded as being unknown, whereas the surface temperature **Y** is measured.

As the inverse problem is ill-posed, in order to cope with the presence of noise in the measured temperature some type of regularisation is required. The Tikhonov regularisation method [16] makes it possible to reformulate the original problem as a well-posed problem that consists of minimising the following objective function:

$$J(\boldsymbol{q}) = \left\|\boldsymbol{Y} - \boldsymbol{Xq} - \boldsymbol{T_{q=0}}\right\|_2^2 + \lambda^2 \left\|\boldsymbol{Lq}\right\|_2^2, \ \lambda>0, \tag{6}$$

where $\left\|\cdot\right\|_2^2$ stands for the square of the 2-norm, λ is the regularisation parameter, **L** is a discrete derivative operator and **T** is the distribution of the external surface temperature derived from a direct numerical solution of the problem obtained by imposing a given convective heat flux distribution on the internal wall side **q**. Often, **L** is the zero, first or second derivative operator: in this work the second-order derivative formulation was chosen to preserve the local variation in the heat-flux distribution. An appropriate choice of λ is a crucial point to find a reliable approximation of the wanted solution and, in this paper, this choice is made by the fixed-point method, [17].

Once the heat-flux distribution at the fluid-wall interface compatible with the experimental temperature data has been determined through the strategy described above, the local convective heat-transfer coefficient can be easily determined, as follows:

$$h_{\text{int}}(\alpha) = \frac{|q_\lambda(\alpha)|}{T(\alpha, r = r_{\text{int}}) - T_b} \tag{7}$$

where $q_\lambda(\alpha)$ is the heat flux distribution estimated under the solution approach based on the Tikhonov regularisation method, T_b is the bulk-fluid temperature on the test section, calculated from the energy balance on the heated section as described in [5,8] and $T(\alpha, r=r_{int})$ is temperature distribution on the tube internal wall efficiently estimated by numerically solving the direct problem by imposing a convective heat flux equal to $q_\lambda(\alpha)$.

4. Results

Figure 4 reports a representative temperature map of a portion of the coil tube with some explicative text labels. The figure clearly reveals that the tube wall is colder at the outer bend side of the coil than at the inner bend side while the temperature gradients are almost negligible along the axis of the tube. This observation confirms that adopting a 2-D numerical model for this type of problem is appropriate for the flow conditions under test.

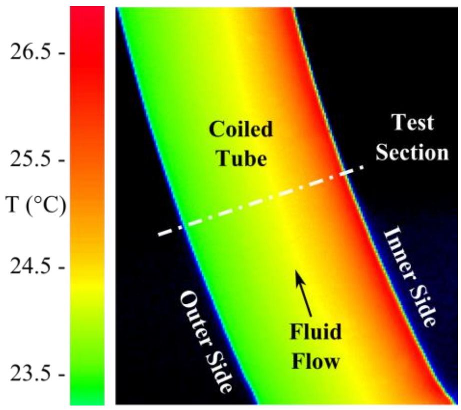

Figure 4: Representative infrared image of the coil wall (Re = 7443, Pr = 8).

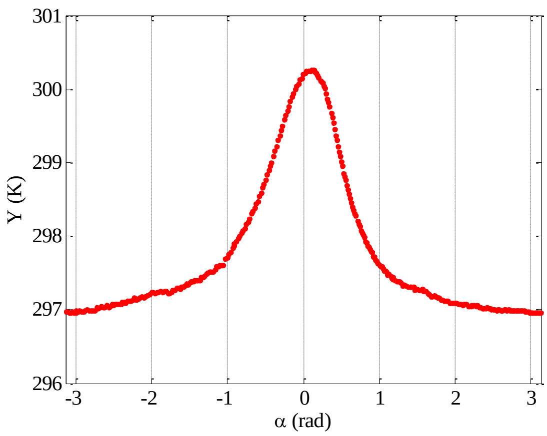

Figure 5. Temperature distribution on the coil external wall at the text section (Re = 7443, Pr = 8).

The temperature values on the test section wall along the whole circumference are reported in figure 5, where the angular coordinate origin is taken at the inner side of the coil. This distribution was obtained processing multiples thermal images of the test section, taken from different point of view around the coil.

For this case, the distribution of the convective heat transfer coefficient restored by the minimisation procedure presented above is reported in figure 6. The 95% confidence interval associated with the estimated values was determined by parametric bootstrap [10]. These data, as expected, highlight that the convective heat transfer coefficient is minimal close to the inner bend side of the coil, and it reaches its maximum at the outer bend side. Moreover, figure 6 shows the effect of torsion induced by the coil pitch: it creates a rotation force that affects the flow pattern. Consequentially, the location of the minimum Nusselt number shifts slightly from zero to higher angular coordinate values.

In figure 6 it is reported also the “apparent” convective heat transfer coefficient, estimated following the procedures suggested by many Authors [12-14] which neglect the heat conduction in the tube wall. Comparing real and apparent heat transfer coefficient values it is clear that neglecting heat conduction in the tube wall misleads the estimated convective heat transfer distribution, increasing the minimum local values and decreasing the maximum ones.

The whole estimation procedure was repeated for various Reynolds number values and representative results for the turbulent regime are plotted in figure 7. To locally compare the Nusselt distributions estimated for the various *Re* values, the shifting effect of the torsion was compensated by introducing a relative angle α^* whose origin was taken where the Nusselt number reaches its minimum.

Figure 8 reports the Nu/Nu_{max} ratio for various Reynolds numbers: by accounting for the experimental uncertainty, it can be stated that this ratio is almost independent of the Reynolds number, analogously to the laminar fluid flow in coils [10]. The best fit of these experimental Nu/Nu_{max} distributions is plotted in figure 9 and compared to the distribution found by Bozzoli et al. [10] for the laminar regime. Some differences between the behaviour in the two different flow regimes can be observed: in the turbulent regime, at the outside surface of the coil, the Nusselt number is about ten times larger than that at the inside surface while in the laminar regime it is only five time larger. Moreover, in turbulent regime the Nu/Nu_{max} pattern shows a typical "V-shape" while in the laminar regime the pattern is more flat near the outside surface of the coil.

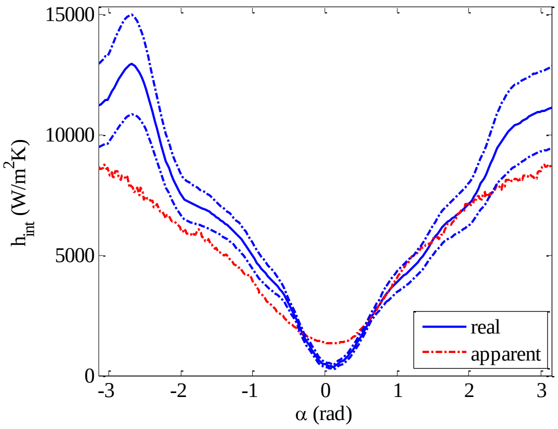

Figure 6. Restored convective heat-transfer coefficient distribution with 95% confidence interval and apparent distribution (*Re* = 7443, *Pr* = 8).

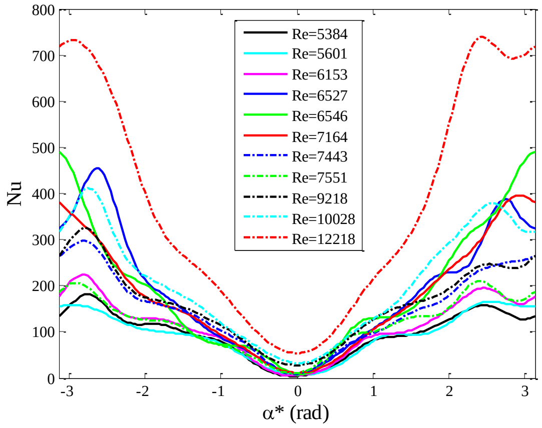

Figure 7. Restored convective heat-transfer coefficient distribution for different Reynolds number values.

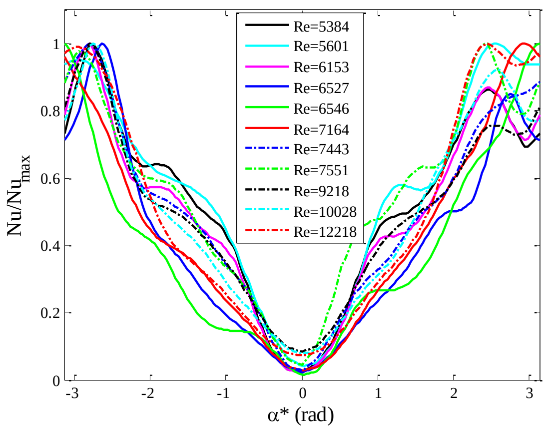

Figure 8. Normalised local Nusselt number for different Reynolds numbers.

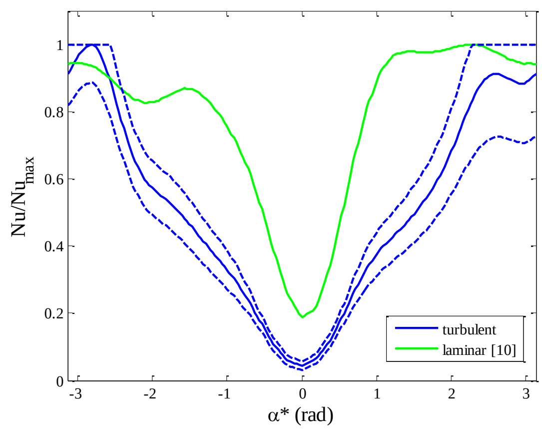

Figure 9. Normalised local Nusselt number and comparison with the data for laminar regime [10].

5. Conclusions

In this paper, it is experimentally investigated the local convective heat transfer coefficient in coiled tubes when turbulent flow regime is present.

The results showed that the variation in the convective heat transfer coefficient along the boundary of the duct section is very significant: at the outside surface of the coil, the Nusselt number is approximately ten times larger than that at the inside surface and this ratio is almost independent of the Dean number.

The purpose of this paper is presenting results which are representative of a wide range of technical applications; this data could be employed both as a useful benchmark for CFD results as well as in the design of coiled tube apparatuses.

Acknowledgments

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