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Estimation of local heat transfer coefficient in coiled tubes under inverse heat conduction problem approach

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A B S T R A C T

Wall curvature is a widely used technique to passively enhance convective heat transfer and it is particularly effective in the thermal processing of highly viscous fluids. These geometries produce a highly uneven convective heat flux distribution along the circumferential coordinate, impacting on the performance of the fluid thermal treatment. Although many authors have investigated the forced convective heat transfer in coiled tubes, most of them have presented the results only in terms of the Nusselt number averaged along the wall circumference. Moreover, regarding the laminar flow regime, few data about the local heat transfer coefficient are available. In this paper a procedure to estimate the local convective heat flux in coiled tubes is presented and tested: the temperature distribution maps on the external coil wall are employed as input data for the inverse heat conduction problem in the wall under a solution approach based on Tikhonov regularization method. The investigation was particularly focused on the laminar flow regime.

1. Introduction

In the thermal processing of highly viscous fluids, wall curvature represents a widely used technique to passively enhance convective heat transfer. Since in these applications the momentum transfer mechanism usually falls under the laminar flow regime, the effectiveness of the heat transfer tools is inevitably penalized and the use of coiled tubes appears particularly interesting. A comprehensive literature review of flow in curved pipes has been presented by Berger et al. [1] and Naphon and Wongwises [2].

In coiled tubes the enhancement effect is due to the fact that the fluid experiences the centrifugal force, whose intensity depends on the local axial velocity and on the radius of curvature of the coil. This causes the fluid to be pushed from the core region towards the outer wall by producing a thinning of the boundary layer. This phenomenon also generates counter-rotating vortices, generally called secondary flows, that increase both heat transfer and pressure drop when compared to the straight tube [3–5].

As a consequence of the highly asymmetrical fluid velocity field, both the wall temperature and the wall heat flux strongly vary along the circumferential coordinate: convective heat flux shows values at the outer wall surface much higher than the ones at the inner wall surface. This unevenness could impact on the performance of the fluid thermal treatment. For instance, in food pasteurization processes the fluid temperature is particularly crucial because, if the whole product is not heated to a specific temperature for a specific period of time, some undesired microorganisms could survive. Moreover, local overheating of the treated fluid could deteriorate the organoleptic properties of the whole final product.

Although many authors have investigated the forced convective heat transfer in coiled tubes, most of them have presented the results only in terms of the Nusselt number averaged along the wall to fluid interface: only few authors have studied the phenomenon locally, most of them by the numerical approach [6–11] and only a few experimentally [12].

Jayakumar et al. [7] numerically analyzed the turbulent heat transfer in helically coiled tubes and presented the local Nusselt number at various cross sections. The results showed that, on any cross section, the highest Nusselt number is on the outer side of the coil, while the lowest one on the inner side. Moreover, the authors proposed a correlation for the prediction of the local Nusselt number as a function of average Nusselt number and angular location for both constant temperature and constant heat flux boundary conditions.

Di Piazza and Ciofalo [10] numerically simulated the convective heat transfer in coiled pipes by different turbulence models; comparing these approaches in terms of predicted local Nusselt number, the Authors concluded that, considering this particular aspect, the Shear Stress Transport and Reynolds Stress turbulence models provide the most satisfactory predictions.

Fugatami and Yoshiyuki [11] carried out a theoretical and experimental study on the effect of the centrifugal and buoyancy

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Nomenclature

a	coil diameter (m)
D	Tube diameter (m)
Dn	Dean number (Eq. (8)) (–)
h	convective heat transfer coefficient (W/m^2 K)
g	gravitational acceleration (m/s^2)
Gr	Grashof number $\mathrm{Gr} = \frac{g \cdot \beta_f \cdot (T_w - T_b) \cdot D_i^3 \cdot \rho_f^2}{\mu_f^2}$ (–)
Nu	Nusselt number (Eq. (7)) (–)
q_g	internal heat generation per unit volume (W/m^3)
r	radial coordinate (m)
Re	Reynolds number ($Re = w \cdot D_i \cdot \rho_f / \mu_f$) (–)
T	temperature (K)
w	mean fluid axial velocity (m/s)
α	overall heat transfer coefficient (W/m^2 K)
β	volumetric thermal expansion coefficient (K^{-1})
ϕ	angular coordinate (rad)
λ	thermal conductivity (W/m K)
μ	dynamic viscosity (Pa s)
ρ	density (kg/m^3)

Subscripts, superscripts

b	bulk
e, ext	external
f	fluid
i, int	internal
w	wall

forces on laminar convective heat transfer in helically coiled tubes. The Authors reported several local heat transfer distributions covering a wide range of Rayleigh and Prandtl number.

Bai et al. [12] studied experimentally the turbulent heat transfer in horizontal helically coiled tubes using deionized water as working fluid. They found out that the local heat transfer coefficient was not evenly distributed along the periphery on the cross section and, at the outer surface of the coil it was from three to four times that at the inner surface.

To the Authors' knowledge, no detailed experimental data are available with regards to local heat transfer in coiled tubes for the laminar flow regime.

One of the most promising way to estimate the local convective heat transfer coefficient on the interior surface of a tube is found by solving the inverse heat conduction problem (IHCP) in the solid domain from the temperature distribution acquired on the exterior wall surface [13]. The well-known difficulties involved in inverse heat conduction problems stem primarily from the fact that they are often ill-posed and, consequently, they are very sensitive to small perturbations in the input data. The majority of the solution strategies for this kind of problems consists in reformulating it as a well-posed problem by minimizing an objective function, which generally expresses the squared difference between measured and estimated temperature discrete data. When the signal to noise ratio is particularly critical, the objective function obtained in that way is not adequate to overcome the problem's instability, while the regularization scheme suggested by Tikhonov and Arsenin [14] proved to be very successful, although the selection of the regularization parameter requires some care [15,16].

In this paper a procedure to estimate the local convective heat flux in coiled tubes is presented and tested: the temperature distribution maps on the external coil wall are employed as input data in inverse heat conduction problem under a solution approach based on Tikhonov regularization method. The experimental investigation was focused on the laminar flow regime, which is often found in coiled tube heat exchangers applications.

2. Experimental setup

In the present investigation a smooth wall helically coiled stainless steel type AISI 304 tube was tested. It was characterized by eight turns following a helical profile along the axis of the tube where the helix diameter and pitch were of about 230 mm and 100 mm respectively, yielding a coiled pipe length L of about 6 m. The tube internal diameter was 14 mm and wall thickness was measured as 1.0 mm.

The working fluid was conveyed by a volumetric pump to a 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 investigating the heat transfer performance of the tube under the prescribed condition of uniform heat flux generated by Joule effect in the wall.

The coiled section was inserted horizontally in a loop completed by a secondary heat exchanger, fed with city water, which enabled keeping the working fluid temperature constant at the tube inlet.

In order to minimize the heat exchange to the environment, the heated section was thermally insulated.

A small portion of the external tube wall was made accessible to a thermal imaging camera by removing the thermally insulating layer and it was coated by a thin film of opaque paint of uniform and known emissivity. The surface temperature distribution was then acquired by means of a FLIR SC7000 unit, with a 640 $\times$ 512 pixels sensor. Its thermal sensitivity, as reported by the instrument manufacturer, is 20 mK at 303 K, while its accuracy is ±1 K. A sketch of the experimental setup is reported in Fig. 1, while Fig. 2 shows the laboratory facility.

In the present experimental investigation the thermally fully developed condition was considered. Therefore, the test section was taken near the end of the heated section, at 5.70 m downstream the inlet section, where, according to [3,4,6,17], the boundary layers reached the asymptotic profiles. The heat flux provided to the fluid was selected in order to make negligible, for the fluid velocity values here investigated, the buoyancy forces compared to inertial ones. This condition makes the results obtained for this particular section representative of the thermally fully developed region.

The inlet and outlet fluid bulk temperatures were measured through type T thermocouples, previously calibrated and connected to a multichannel ice point reference, type KAYE K170-50C. 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.

Ethylene Glycol was used as working fluid in the Reynolds number range 160-1115. In the temperature range characterizing the experimental conditions, the Prandtl number of the working fluid varied in the range 140-180, while the wall to bulk fluid temperature difference was limited within 5 K and, therefore, the fluid property variation effect was considered negligible.

3. Results and discussion

The temperature maps acquired on the external coil wall were employed as input data of the inverse heat conduction problem in the wall under the solution approach based on Tikhonov

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