NUMERICAL ANALYSIS OF THE LAMINAR CONVECTIVE HEAT-TRANSFER COEFFICIENT IN COILED TUBES AND EXPERIMENTAL VALIDATION

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INTRODUCTION:

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.

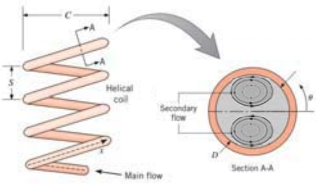

Figure 1: Cylindrical helicoidal heat exchanger

GOVERNING EQUATIONS:

The governing equations are integrated numerically in a steady state condition with the assumption of fully developed flow for both what concerns the hydrodynamic and the thermal problem. Under the assumption of Newtonian, incompressible and constant thermophysical properties fluid, these equations, in case of negligible viscous dissipation, are:

$$\nabla \mathbf{u} = 0$$

$$\rho \frac{D\mathbf{u}}{Dt} = -\nabla p + \mu \nabla^2 \mathbf{u}$$

$$\frac{DT}{Dt} = \alpha \nabla^2 T$$

To reduce the computational cost, translational *periodic boundary conditions* for the inlet and outlet sections were used. The pressure difference Δp along each module is selected according to expected mass flow rate while the temperature difference ΔT is related to the wall heat flux per unit surface q as follows:

$$\Delta T = \frac{q \cdot A_w}{\dot{m} \cdot c_p}$$

Boundary conditions:

- Initial condition:
  - *Inlet:*
    - Mass flow rate: $\dot{m} = 0.004\ [^{kg}/_{s}] \rightarrow Re = 364$
    - Constant temperature profile: $T_{(\varphi,r)} = 294.9[K]$
  - *Outlet:*
    - Out flow
- *Wall:*
  - Stationary wall
  - No slip condition: $\overrightarrow{v_{(\varphi,R_{in})}} = 0\ [^{m}/_{s}]$
  - Constant heat flux: $q_{(\varphi,R_{in})} = 1272.58\ [^{W}/_{m^2}]$

The heat transfer results are evaluated in terms of the average Nusselt number described as follows:

$$Nu = \frac{h \cdot D}{\lambda} \qquad \text{where:} \qquad h = \frac{q}{\left(\overline{T_w} - \overline{T_b}\right)}$$

the pressure drop is described in terms of the friction factor defined as follows:

$$f = \frac{\rho \cdot A_c^2 \cdot D \cdot \Delta p}{\dot{m}^2 \cdot l}$$

MESH CONVERGENCE ANALYSIS:

Table 1: Convergence analysis: Meshes tested

	Mesh 1:	*Mesh 2:*	*Mesh 3:*	*Mesh 4:*
Elem. Size [mm]:	1.067	0.8	0.6	0.45

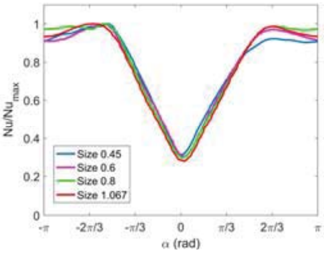

Figure 2: Local convergence analysis

RESULTS:

The centrifugal force due to the geometry of the coil deform significantly the velocity field. These effects produce two secondary flow called *Dean's vortices.* This vortices push the fluid from the inner bend of the coil to the outer bend and this cause a non-uniform temperature distribution at the fluid-wall interface.

Figure 3: Dean's vortices

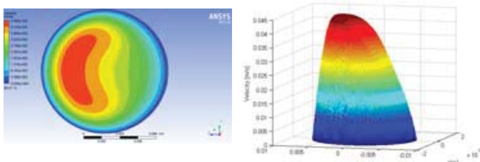

Figure 4: Velocity profile

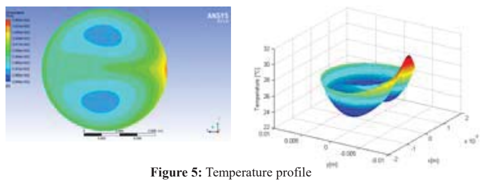

Figure 5: Temperature profile

Darcy friction factor:

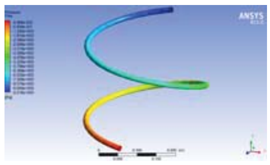

Figure 6: Pressure drop

Table 2: Darcy friction factor

	Cylindrical	*Shah* [1]	*CFD*
f	0.176	0.21	0.26

Since the velocity filed is modified by the Dean's vortices, the Darcy friction factor is greater than the value expected for the straight tube:

$$f_{lam} = \frac{64}{Re}$$

Nusselt Number:

Table 3: Average Nusselt number

	Shah [1]	*Xin* [2]	*Experimental* [3]	*CFD*
$\overline{Nu}$	10.61	10.1	10.53	10.70

[1] Shah, R. K., and S. D. Joshi, "Handbook of Single-Phase Convective Heat Transfer" , WileyInterscience

[2] Xin R.C and Ebadian M.A, "The effect of Prandtl number on local and average heat transfer coefficient in helical pipe" . Heat transfer Journal, vol(119) pp. 463(1997).

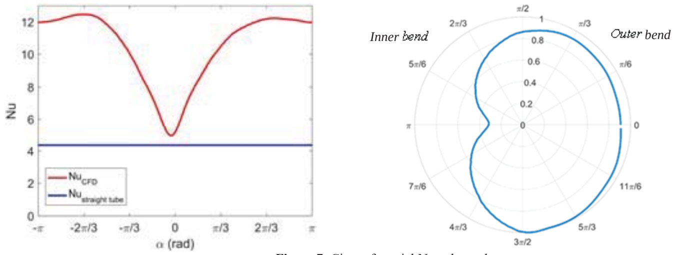

Figure 7: Circumferential Nusselt number

EXPERIMENTAL PROCEDURE:

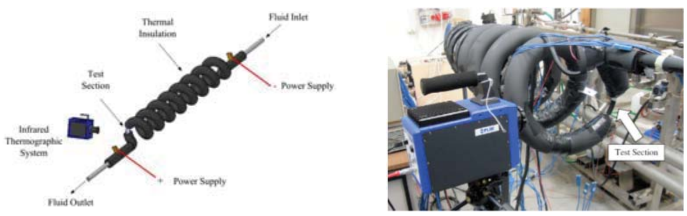

Figure 8: Experimental apparatus

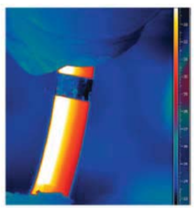

Figure 9: Acquired thermographic image

The experimental investigation has been performed by applying the Inverse Heat Conduction Problem approach: the internal convective heat transfer coefficients was estimated by measuring the external wall temperature using an infrared thermographic camera and applying the Tikhonov regularization technique.

CONCLUSIONS:

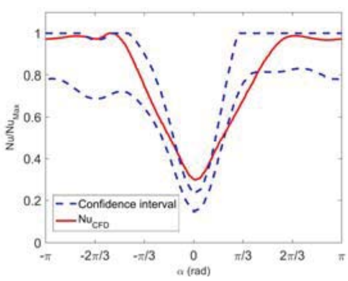

Figure 10: Nusselt number ratio, experimental validation

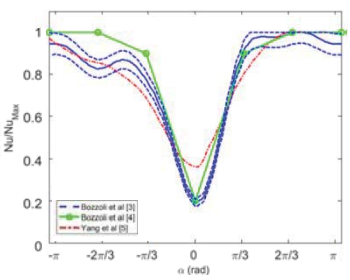

Figure 11: Nusselt number ratio, experimental results

The comparison between the numerical results and the confidence interval of the experimental data shows a good agreement, thus validating the CFD model (Figure 10) . The experimental data are also in good correspondence with the data available in literature, as shown in Figure 11.

The high variation in the circumferential Nusselt Number causes an overheating of the fluid nearby the inner bend while at the outer bend it's not heated as well. This observation is very important if the fluid evolving inside the het exchanger is food, such as milk; in this case the sterilizing effect is not uniform in the cross-sectional area. For this reason, to guarantee a lower level of the total bacteria charge, the fluid have to be overheated reducing the organoleptic property of the food and increasing the energy consumption.

Figure 12: Helicoidal heat exchanger for food industry

[3] Bozzoli et al: Estimation of the local heat-transfer coefficient in the laminar flow regime in coiled tubes by the Tikhonov regularisation method

[4] F. Bozzoli, L. Cattani, S. Rainieri, G. Pagliarini, Estimation of local heat transfer coefficient in coiled tubes under inverse heat conduction problem approach, Exp. Therm. Fluid Sci. (2013)

[5] G. Yang, F. Dong, M.A. Ebadian, Laminar forced convection in a helicoidal pipe with finite pitch, Int. J. Heat Mass Transfer 38 (5) (1995) 853–862.

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