Experimental investigation on the convective heat transfer enhancement for highly viscous fluids in helical coiled corrugated tubes

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Abstract. In the present analysis, the forced convective heat transfer in smooth and corrugated helical coiled tubes was experimentally studied in the Reynolds and Dean number ranges 50÷1200 and 12÷295 respectively, by adopting Ethylene Glycol as working fluid. The primary aim of the investigation is to study the combined effect of the wall curvature and of the wall corrugation in the thermal entrance region for highly viscous fluids. Two coiled tubes with a curvature ratio of about 0.06, one with smooth wall and the other with spirally corrugated wall, were investigated under the uniform heat flux boundary condition. The main conclusion is that in the Reynolds number range analyzed, both curvature and corrugation enhance the heat transfer. For Dean number values lower than about 120 the wall curvature effect prevails, and the heat transfer enhancement reflects Nusselt numbers that are approximately 2-3 times higher than the straight smooth section. For greater Dean number values, the wall corrugation instead prevails. In fact the corrugated coiled tube reaches Nusselt number values which are up to 8 times higher than the ones expected for the smooth straight tube. The smooth coiled tube shows instead thermal performances at maximum 3.6 times over the straight section.

1. Introduction
Both wall curvature and wall corrugation represent widely used techniques to passively enhance the convection heat transfer. This issue is particularly crucial in applications in which the thermal processing of highly viscous fluids is required, such as in the food, chemical, pharmaceutical and cosmetics industries. In fact often in these conditions the momentum transfer mechanism is necessarily laminar and therefore the effectiveness of the heat transfer apparatuses in which the fluids are conveyed is inevitably penalized. The desired augmentation effect is due to several factors. In curved geometries the distortion of the velocity boundary layer and the onset of vortices generated by the centrifugal forces has a positive effect on the convective heat transfer mechanism [1]. This solution, which often produces a swirl or helical type flow, appears very interesting also in the conditions in which the flow persists in the laminar regime [2]. In corrugated wall tubes the most important effect is instead related to the macroscopic mixing of the fluid, activated by the destabilization of the flow which leads to the early onset of the transfer mechanism associated with the transitional/turbulent regime [3-4]. To these heat transfer enhancement effects it is necessarily associated a pressure drop penalty, which has to be accounted for [5]. Recently the combined effect of wall curvature and wall
corrugation was experimentally investigated by Rainieri et al. [6]. The results were obtained in two Reynolds number ranges separately (in particular in the ranges 5–13 and 150-1500). Rainieri et al. [6] concluded that the wall curvature enhances heat transfer at all Re, whereas the wall corrugation enhances heat transfer only in the higher Re range. Moreover the wall corrugation is totally ineffective in the low Re range and, if helical coils are present, it also destroys the benefit induced by the wall curvature. The helical coiled wall corrugated tube was also investigated numerically by Zachár [7].

The aim of the present investigation is to further study the behavior of these geometries in the Reynolds number range in which the wall corrugation superimposed to the wall curvature effect is expected to provide benefic effect to the heat transfer mechanism. Two coiled tubes, one with smooth and the other with spirally corrugated wall, were tested in the Reynolds number range 50÷1200 by adopting Ethylene Glycol as working fluid under the uniform heat flux boundary condition.

2. Tubes geometry and experimental setup

Two different stainless steel type AISI 304 helical coiled tubes, one with smooth and the other with corrugated wall, were considered in the present analysis. Both the smooth and the corrugated tubes were characterized by 8 coils, which followed an helical profile having a diameter $a=0.25$ m and a pitch $s=0.1$ m, with a curvature ratio, $\delta$ of about 0.06. The wall corrugated tube considered in the present investigation is included in the general category usually known as spirally enhanced tubes, exemplified by the section shown in figure 1. It presented an internal helical ridging corresponding to an external helical grooving, obtained by embossing a smooth stainless steel tube. In the present study, tubes having a wall thickness of 1 mm and an internal envelope diameter $D_{env}$ of 14 mm were analyzed, while the corrugation profile had a depth of 1 mm and pitch of 16 mm. The test sections were inserted horizontally in the loop described in [5] which enabled to investigate their behavior under the constant heat flux boundary condition at the fluid-wall interface. The coiled section is preceded by a straight smooth wall section of 0.60 m and it was followed by a straight section of about 1.55 m. The test section was fitted with steel fin electrodes which were connected to a power supply, type HP 6671A, working in the ranges 0–8 V and 0–220 A. The supplied power was considered uniform along the heat transfer section. A secondary heat exchanger, fed with city water, provided a constant working fluid temperature at the inlet tube’s section.

![Figure 1. Tube’s geometry: a)helical coil parameters; b)wall corrugation profile](image1)

![Figure 2. Coiled tube under test.](image2)

The heated section was about 6 m long and it was thermally insulated to minimize the heat exchange to the environment. The temperature of the wall and of the fluid at the inlet and outlet sections was measured by 40 type T thermocouples, previously calibrated and connected to a multichannel ice point reference, type KAYE K170-50C. Regarding the wall temperature the sensors were attached at different circumferential locations to the external tube’s surface and at different axial locations along the heated section. In particular the thermocouples were placed along the external and
internal side of the coil in order to investigate the distortion induced by the centrifugal force to the boundary layers. The inlet fluid temperature was measured by a thermocouple sensor placed on the tube’s wall upstream the starting heating section. The fluid bulk temperature at any location along the heat transfer section was calculated from the energy balance also accounting for the heat losses towards the environment. These were estimated in a preliminary calibration of the apparatus aimed to measure the overall thermal resistance between the internal tube wall and the environment. This procedure was performed as follows: in absence of fluid circulation, a known power rate was supplied to the tube wall and the wall to ambient temperature difference was measured. The value 0.83 K/W was found for the overall thermal resistance that yielded heat losses of about 1% of the supplied power. The outlet fluid temperature was checked also by placing a thermocouple at the end of the heated section. The flow rate was obtained by measuring the time needed to fill a volumetric flask placed at the outlet of the test section. The data acquisition system consists of a high precision multimeter (type HP 3458A) connected to a switch control unit (type HP 3488A) driven by a Personal Computer. Pressure drop throughout the coiled section was measured by two piezometric tubes in isothermal conditions. Ethylene Glycol was used as working fluid in the Reynolds number range 50-1200 which, for the curvature ratio value under investigation corresponded to the Dean number range 12-295. In the temperature range of interest the Prandtl number of the working fluid varied in the range 150-300.

3. Data processing
The performances of each tube were determined in terms of pressure drop penalty and heat transfer enhancement, quantified by means of the friction factor and the Nusselt number distributions respectively. In the data reduction, the maximum internal diameter (the envelope diameter $D_{env}$) was adopted as the characteristic length. The Darcy friction factor is related to the pressure drop $\Delta p$ as follows:

$$f = \frac{\Delta p}{\rho \cdot \frac{D_{env}}{L} \cdot \frac{2}{w^2}}$$

(1)

where $w$ is the mean fluid axial velocity and $L$ is the length of the test section.

The tests were performed by varying the Reynolds number, and consequently the Dean number, which are defined as follows:

$$Re = \frac{w \cdot D_{env}}{v}$$

(2)

$$De = Re \sqrt{\delta}$$

(3)

Regarding the heat transfer performance, the local Nusselt number was computed as follows:

$$Nu_x = \frac{h_x \cdot D_{env}}{\lambda}$$

(4)

where the circumferentially averaged local convective heat transfer coefficient [6] $h_x$ is:

$$h_x = \frac{q}{(T_w - T_b)}$$

(5)

In evaluating the heat exchanged per unit surface area, $q$, the heat transfer area was assumed equal to the envelope cylinder surface area. Moreover, in the data reduction the average bulk temperature between the inlet and outlet sections was used for evaluating all fluid properties.

The average Nusselt number over the heated length $L$ was then evaluated as follows:

$$Nu = \frac{1}{L} \int_0^L Nu_x \, dx$$

(6)

The maximum uncertainty for the friction factor, Nusselt number and the Reynolds number was estimated to be respectively ±6%, ±4% and ±3% [6,8].
4. Results and discussion
The heat transfer enhancement and the pressure drop penalty effects due to the wall curvature only and to the combined effect of wall curvature and corrugation was studied for a given curvature ratio value of about 0.06. This was carried out by testing two helical coiled tubes (with the same curvature ratio and coils number) one with smooth wall and the other with corrugated wall.

4.1 Pressure drop
Figure 3 shows the average Darcy friction factor for both the smooth and the corrugated wall coiled sections versus the Reynolds number. In the same figure the analytical solution for the laminar fully developed flow, holding for the straight smooth wall tube is reported. For the coiled smooth wall tube, the friction factor values are close to the expectations for the straight section for Reynolds number values lower than 200. These results are in agreement with the data available in literature. In fact most of the available correlations holding for helical coiled tubes predict friction factor values similar to that of a straight pipe, for small Dean number [5]. For Reynolds number values in the range 200-400, the curvature effect on the pressure drop is still small, but not negligible: a maximum enhancement over the straight section’s behavior of about 1.2 was found. For the coiled corrugated wall tube, the friction factor values are higher than the values associated with the smooth coiled section. This pressure drop augmentation becomes more significant for Reynolds number values greater than 100. In particular in the Reynolds number range here investigated a maximum enhancement over the straight smooth wall section’s behavior of about 2.2 was found. The augmentation effect due to the wall corrugation with respect to the smooth wall behavior measured for the present coiled sections is similar to the one reported in literature for straight sections. Indeed, for spirally enhanced tubes a friction factor increase with respect to the smooth tube’s behavior, in the range 1.4-1.6 is reported for Reynolds number included within the range 100-800 [4].

4.2 Heat transfer
The wall temperature distribution along the curvilinear coordinate and the corresponding local Nusselt number versus the dimensionless abscissa are reported in figures 4 and 5 for both the smooth wall and the corrugated wall pipe for two runs performed in the low Reynolds number range. As the fluid enters the coiled section it starts in fact to be affected by the centrifugal force due to the curvature of the tube. Therefore the fluid is colder towards the outside wall where the axial velocity is expected to be maximum. For what concerns the Local Nusselt number data, it is necessary to observe that the inside and outside local values have to be considered as apparent values, since in their calculation the nominal heat flux density has been adopted, by de facto disregarding the effect of the heat flux redistribution into the pipe’s wall which occurs due to the significant circumferential wall temperature variation. As expected, as a consequence of the fluid mixing due to the centrifugally induced secondary flow, the local Nusselt number, circumferentially averaged, reaches values higher than the

![Figure 3. Darcy friction factor versus Reynolds number for both the tubes under test and comparison with the analytical solution for the straight smooth wall tube.](image-url)
ones obtained in a straight pipe. The Nusselt number values obtained in the downstream region of the coiled section are in agreement with the asymptotic values predicted by Janssen and Hoogendoorn [9], for laminar convective heat transfer in helical coiled tubes under the boundary condition of uniform wall heat flux. The augmentation over the smooth straight tube’s behaviour [10] is higher for the corrugated coiled tube than for the smooth coiled tube, although the data confirm that in this Reynolds number range the wall curvature effect provides the most important contribution [6].

![Graph 1](image1.png)

**Figure 4.** Wall temperature (a) and local Nusselt number (b) for the coiled smooth wall tube (Re=76) and comparison with the analytical solution for the straight smooth wall tube under uniform heat flux boundary condition (SSW: straight smooth wall).

![Graph 2](image2.png)

**Figure 5.** Wall temperature (a) and local Nusselt number (b) for the coiled corrugated wall tube (Re=80) and comparison with the analytical solution for the straight smooth wall tube under uniform heat flux boundary condition (SSW: smooth wall).

Figures 6 and 7 show wall temperature distribution and the corresponding local Nusselt number for the two geometries under test for Re of about 1000.

The asymptotic Nusselt number values are compared with the correlation suggested by Janssen and Hoogendoorn [9] and Zachár [7] for the smooth and corrugated wall tube respectively. The Nusselt number distribution for the corrugated coiled tube assumes an almost flat behavior along the curvilinear coordinate, by confirming that the wall corrugation is responsible for an early transition from the laminar regime [4,11-13].
For the smooth coiled tube a thermal entrance effect is instead still present in the Nusselt number distribution. The smooth wall coiled tube Nusselt number values are higher that the values predicted by Janssen and Hoogendoorn [9], while the coiled corrugated wall tube thermal performance approaches to values numerically predicted by Zachár [7].

In order to compare the performances of the two wall conformations (coiled smooth and corrugated wall tubes) the average Nusselt number over the heated length is reported versus the Dean number in figure 8. In the same figure the analytical solution for the straight smooth wall tube under uniform heat flux boundary condition, is reported too [10]. The data show that for low Dean number values the wall corrugation effect is almost negligible and the enhancement over the smooth straight tube behavior has to be ascribed to the wall curvature effect that allows reaching a maximum enhancement over the straight section’s behavior of about 3.6 in the Dean number range 20-120. For larger Dean number values the effect of the wall corrugation starts to prevail by augmenting the convective heat transfer mechanism significantly. In the Dean number range 120-300 a maximum enhancement over the straight section’s behavior of about 3.6 and 8 was found for the coiled smooth and corrugated wall tube respectively. These data suggest that a critical Dean number value, above which the wall corrugation effect start to become effective, exists for coiled sections. For the particular geometry considered in the present investigation, the critical Dean number value was about 120, that corresponds to a Reynolds number value of about 500. This behavior is consistent with the results available for straight spirally enhanced tubes for which a critical Reynolds number in the range 500-800 is reported [4].

![Graph](image1.png)

**Figure 6.** Wall temperature (a) and local Nusselt number (b) for the coiled smooth wall tube (Re=1001) and comparison with the analytical solution for the straight smooth wall tube under uniform heat flux boundary condition (SSW: straight smooth wall).

### 5. Conclusion

The forced convective heat transfer in helical coiled tubes having smooth and corrugated wall was studied in the Reynolds and Dean number ranges 50÷1200 and 12÷295 respectively, by adopting Ethylene Glycol as working fluid. The aim of the research was to investigate the effect of the curvature and of the combination of curvature and corrugation as passive convective heat transfer enhancement techniques. The main conclusion is that in the Reynolds number range analyzed both the curvature and the combination of curvature and corrugation enhances the heat transfer. For low $De$ values the wall curvature effect prevails, and the heat transfer enhancement produces Nusselt number that are approximately 2-3 times higher than the straight smooth section for both the corrugated and the smooth coiled tubes. Therefore in this regime the wall corrugation is almost ineffective. For higher $De$ values instead, the wall corrugated coiled tube reaches Nusselt number values which are about 8...
times higher than the one expected for the smooth straight tube. These data suggest that a critical Dean number value, above which the wall corrugation effect starts to become effective, exists for helical coiled tubes. For the particular geometry considered in the present investigation, the critical Dean number value was about 120, that corresponds to a Reynolds number value of about 500. These results hold for the curvature ratio value and corrugation profile here investigated. Further research is needed in order to generalize the results since interesting applications of these geometries are expected.

![Figure 7](image1.png)

**Figure 7.** Wall temperature (a) and local Nusselt number (b) for the coiled corrugated tube (Re=1004) and comparison with the analytical solution for the straight smooth wall tube under uniform heat flux boundary condition (SSW: straight smooth wall).

![Figure 8](image2.png)

**Figure 8.** Average Nusselt versus Dean number for both the tubes under test and comparison with the analytical solution for the straight smooth wall tube under uniform heat flux boundary condition.

**Nomenclature**

- $a$: Coil diameter (m)
- $c_p$: Specific heat (J/kg·K)
- $De$: Dean number
- $D_{env}$: Tube envelope diameter (m)
- $f$: Darcy friction factor
- $h$: Convective heat transfer coefficient (W/m²·K)
- $Nu$: Nusselt number
- $Re$: Reynolds number
- $\Delta T$: Temperature difference
- $\rho$: Density
- $\mu$: Dynamic viscosity
- $L$: Length
- $w$: Corrugation wavelength

The Dean number is defined as $De = Re\sqrt{\delta}$.
Length of the test section \( L \) m

Nusselt number \( Nu \)

Prandtl number \( Pr \)

Heat flux exchanged per unit surface \( q \) W/m\(^2\)

Reynolds number \( Re \)

Helix pitch \( s \) m

Curvilinear coordinate along the tube’s axis starting from the beginning of the heated length \( x \) m

Dimensionless coordinate \( x^* \)

Mean fluid axial velocity \( w \) m/s

Curvature ratio \( \delta \)

Fluid thermal conductivity \( \lambda \) W/m·K

Dynamic viscosity \( \mu \) Pa·s

Kinematic viscosity \( \nu \) m\(^2\)/s

Subscripts

Bulk \( b \)

Wall \( w \)

Local value along the curvilinear coordinate \( x \)

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