

Experimental investigation on the convective heat transfer enhancement in tubes with cross-helix profile wall corrugation

Luca Cattani^b, Fabio Bozzoli^{a,b}, Sara Rainieri^{a,b}, Giorgio Pagliarini^{a,b}

^a Department of Industrial Engineering, University of Parma,
Parco Area delle Scienze 181/A I-43124 Parma, Italy

^b SITEIA.PARMA Interdepartmental Centre, University of Parma,
Parco Area delle Scienze 181/A, I-43124 Parma, Italy

E-mail address of the corresponding Author (Luca Cattani): luca.cattani@nemo.unipr.it

Abstract. Wall corrugation is a popular heat transfer enhancement technique since it acts as a disturbance source in the flow that significantly enhances the thermal performance of the tube section with a limited pressure drop augmentation if compared to other passive techniques, such as the ones based on insert devices. In the hereby presented study nine pipes characterised by a cross-helix type corrugation were tested: this kind of corrugation, that was obtained by rolling twice the same tube with two helical corrugations evolving along opposite directions, showed a performance that exceeded the single helix-type corrugation behaviour and moreover presented an earlier transition to unstable regime. In the analysis the effect of the cross-helix type corrugation profile on the forced convection heat transfer mechanism was experimentally investigated in the Reynolds and Prandtl number range $25 \div 1000$ and $115 \div 150$ respectively. In particular, the effect of both the corrugation depth and the corrugation pitch were analysed. The results were compared with other types of wall corrugation and with the predictions for the smooth tube in order to point out the achieved heat transfer enhancement.

Key words: Convective heat transfer enhancement, cross-helix corrugated tubes, passive heat transfer enhancement technique.

1. Introduction

Among passive heat transfer enhancement techniques, wall corrugated tubes certainly represent the most widely adopted solution in heat exchangers industrial applications. In fact, wall corrugation acts as a disturbance source in the flow that significantly enhances the thermal performance of the tube section by limiting the pressure drop augmentation if compared to other passive techniques, such as the ones based on insert devices [1]. The main augmentation mechanisms activated by wall corrugation are: the interruption of the thermal boundary layer, the flow mixing due to the onset of secondary swirl flow components, the increase of the heat transfer surface area and, above all, the early onset of a transitional turbulent flow regime [2,3]. Practically these phenomena are able to activate a significant heat transfer enhancement only if a critical condition can be reached [4-6]. Under this critical value, they normally don't present a significant heat transfer augmentation over the smooth wall behaviour. On the other hand, an important increase is achieved with the transition to the unsteady regime. This is a results of complex interactions between the core fluid and the boundary layer fluid through shear-layer self-sustained oscillations [7]. Guzman and Amon [8] investigated the transition from laminar to unsteady



state for converging-diverging conducts by numerical simulations in Reynolds number range from 10 to 850. They stated that the transition occurred through a sequence of intermediate state of self-sustained periodic, quasi-periodic and finally aperiodic or chaotic regime.

Wang and Vanka [9] numerically investigated the heat transfer for flow through a periodic array of wavy passages: the Authors observed that above a critical value of Reynolds number of about 180 a transition appeared, and that it was characterised by a destabilization of thermal boundary layers replacing the fluid close to the wall with that of the core region thus producing a mechanism of heat and mass transfer enhancement. Niceno and Nobile [7] studying different types of wavy channels observed the appearance of the unsteady regime for a range of Reynolds number values between 80 and 200.

Garcia et al. [10] analysed the thermal behaviour of three types of enhancement technique based on artificial roughness: corrugated tubes, dimpled tubes and wire coil inserts. The Authors identified the onset of the transition to the unsteady regime in the range $100 < Re < 200$ for the wire coils inserts and $900 < Re < 1000$ for the other two types of roughness. The type of transition (e.g., smooth or sudden) and the Reynolds number value at which it occurs were found to be strongly depending on the roughness profile and geometrical characteristics.

Regarding wall corrugation, the helical profile is the most widely adopted in industrial applications, since these tubes can be quite easily manufactured by a continuous cold rolling process for both inward or outward corrugation. In this case, the main geometrical parameters that characterize the corrugation profile are: the tube diameter, the corrugation pitch, the corrugation depth, the number of helix starts and the corrugation profile [2]. For spirally inward corrugations, the corrugation depth was proved to be the most significant parameter that affects the critical Reynolds number: the critical Reynolds number decreased as the corrugation depth increased [4]. Moreover, for a given corrugation depth also the sharpness of the corrugation profile had a non-negligible effect [5]. This phenomenon was even amplified by a cross-helix type corrugation, for the first time investigated in literature in [11]. This kind of corrugation, that was obtained by rolling twice the same tube with two helical corrugations evolving along opposite directions, showed a performance that for a given corrugation depth exceeded the single helix-type corrugation behaviour and presented an earlier transition to unstable regime [11].

In the present investigation the effect of the cross-helix type corrugation profile on the forced convection heat transfer mechanism is experimentally further investigated in the Reynolds and Prandtl number range $25 \div 1000$ and $115 \div 150$ respectively. In particular, the effect of both the corrugation depth and of the corrugation pitch is analysed, by pointing out the effect of heat transfer enhancement and of promotion of the transition to unstable regime.

2. Tubes Geometry, Experimental Setup and Data Processing

The tubes under test show a wall corrugation with a cross-helix profile and they belong to the general category usually referred to as spirally enhanced tubes. In the present investigation nine tubes, having different corrugation pitch l and depth e (as specified in table 1) were tested.

Table 1. Corrugated tubes characteristics.

	l (mm)	e (mm)	l/D	e/D	Ω
Tube No. 1	13	0.6	0.928	4.29×10^{-2}	1.98×10^{-3}
Tube No. 2	13	0.8	0.928	5.71×10^{-2}	3.52×10^{-3}
Tube No. 3	13	1.0	0.928	7.14×10^{-2}	5.49×10^{-3}
Tube No. 4	18	0.6	1.29	4.29×10^{-2}	1.43×10^{-3}
Tube No. 5	18	0.8	1.29	5.71×10^{-2}	2.54×10^{-3}
Tube No. 6	18	1.0	1.29	7.14×10^{-2}	3.97×10^{-3}
Tube No. 7	29	0.6	2.07	4.29×10^{-2}	0.89×10^{-3}
Tube No. 8	29	0.8	2.07	5.71×10^{-2}	1.58×10^{-3}
Tube No. 9	29	1.0	2.07	7.14×10^{-2}	2.46×10^{-3}

They are obtained by embossing a smooth tube made of stainless steel having an internal diameter D of 14 mm and a wall thickness s of 1 mm. In Table 1 also the dimensionless pitch and corrugation depth are specified together with the corresponding severity index Ω , defined as $e^2/l \cdot D$ according to Withers and Habdas [12]. Two representative tubes are shown in figure 1.



Figure 1a: Tube No.1.



Figure 1b: Tube No.9.

The tubes were tested by using the experimental set-up and data acquisition system sketched in figure 2. The working fluid was conveyed by a volumetric pump to a holding tank, and it entered the test section equipped with stainless-steel fin electrodes, which were connected to a power supply, type HP 6671A. 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. 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 tested 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 test section inlet. The heated section was preceded by an unheated development section of about 1 m. To minimise the heat exchange with the environment, the heated section was thermally insulated. 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 assumed distributed uniformly per unit length over the heat transfer surface area, decreased by the heat losses through the insulation. Volumetric flow rates were obtained by measuring the time needed to fill a volumetric flask placed at the outlet of the test section.

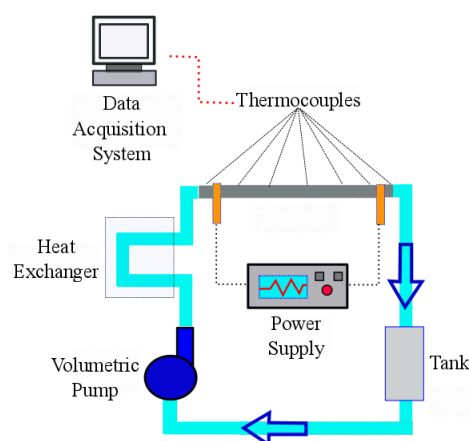


Figure 2. Sketch of the experimental setup.

The thermal entrance problem under the uniform wall heat flux boundary condition was then studied by using mixtures of water and ethylene glycol. In the temperature range characterizing the experimental

conditions, the fluid Prandtl number range was $115 \div 150$ while the effect within the boundary layer of the variation of the fluid properties with temperature was considered to be negligible.

The experimental set-up enabled to measure the local Nusselt number along the axial coordinate and its average values along the heated section as follows:

$$Nu_x = \frac{h_x \cdot D}{\lambda} \quad (1)$$

$$Nu = \frac{1}{L_H} \int_0^{L_H} Nu_x dx \quad (2)$$

where λ is the fluid thermal conductivity, x the axial coordinate, L_H the length of the heated section while the local convective heat transfer coefficient h_x is defined as follows:

$$h_x = \frac{q}{(\bar{T}_w - T_b)} \quad (3)$$

where \bar{T}_w and T_b are the circumferentially averaged wall temperature and local bulk fluid temperature respectively. The circumferentially averaged wall temperature was obtained by averaging the values measured by the thermocouples placed on the external surface of the pipe wall (i.e., two sensors at each axial position), under the insulating material. In evaluating the heat exchanged per unit surface area, q , the heat transfer area was assumed equal to the envelope cylinder surface area. In the data reduction the local bulk temperature was used for evaluating fluid properties. The Nusselt number value has been evaluated by varying the Reynolds number, which is defined as follows:

$$Re = \frac{w \cdot D}{\nu} \quad (4)$$

being ν and w the fluid kinematic viscosity and mean fluid velocity over the tube's cross section, respectively. Regarding the relevant dimensionless parameters, the uncertainties have been calculated by using the usual propagation of errors procedure [13] and the maximum uncertainty for Nusselt number and the Reynolds number was estimated to be respectively $\pm 10\%$ and $\pm 3\%$.

3. Results

The local Nusselt number along the dimensionless abscissa, defined in equation 5, is reported in figure 3 for tubes No. 1, 2, 3, 7, 8, 9 for two representative Reynolds number values. In the figure, it is reported also the analytical solution holding for the smooth wall pipe with uniform heat flux boundary condition [14] which was previously proved to be representative for the present experimental set-up [6,11,15].

$$x^* = \frac{x}{Re \cdot Pr \cdot D} \quad (5)$$

The effect of the corrugation is evident: for all tubes here reported, by increasing the Reynolds number the heat transfer characteristics of the enhanced tube depart from the smooth wall behavior by providing a significant heat transfer augmentation.

For the lowest investigated Reynolds number values, the thermal behavior of all the pipes is similar to each other and close to smooth wall behavior while for the highest Reynolds number values their performance changes depend on the geometrical parameters. For the two tubes with the lowest corrugation depth (tubes No.1 and 7, $e = 0.6$ mm) the enhancement is not as much significant. For the tubes with the corrugation depths $e = 0.8$ mm (tubes No.2 and 8) and $e = 1$ mm (tubes No.3 and 9), the enhancement over the smooth wall behavior becomes very important. This behavior could be interpreted as a consequence of a departure from the steady laminar flow induced by the cross-helix wall corrugation by pointing out that the corrugation depth is a fundamental parameter in promoting this transition.

In figure 3 it is also possible to observe that, in agreement with the findings of Rainieri et al. [11], in corrugated pipes in the fully developed conditions there is not a flat distribution of the local Nusselt number but it presents peaks and descents due to the continuous interruption of the boundary layer promoted by the wall corrugation.

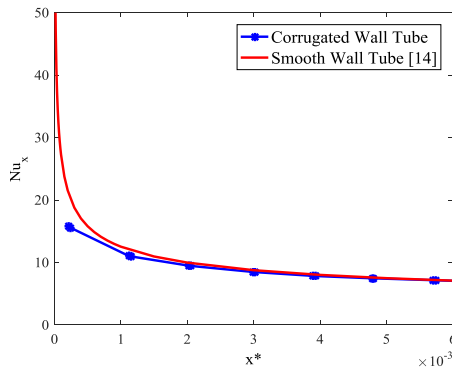


Figure 3a: Local Nusselt number versus the dimensionless abscissa for $Re = 89$ - Tube No.1.

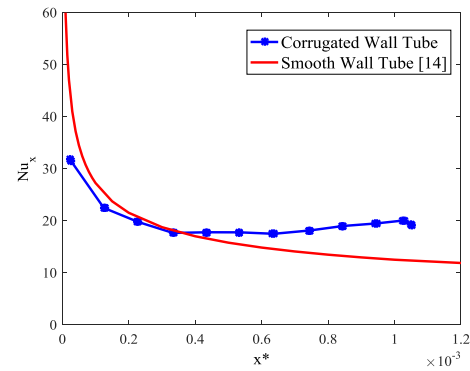


Figure 3b: Local Nusselt number versus the dimensionless abscissa for $Re = 978$ - Tube No.1.

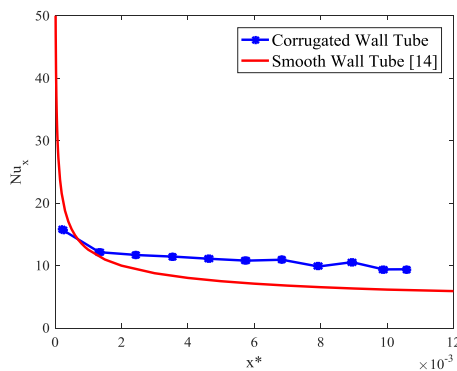


Figure 3c: Local Nusselt number versus the dimensionless abscissa for $Re = 88$ - Tube No.2.

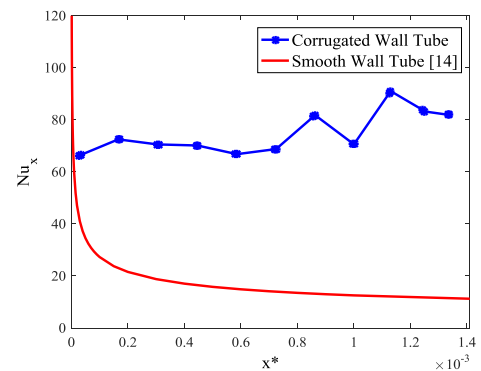


Figure 3d: Local Nusselt number versus the dimensionless abscissa for $Re = 791$ - Tube No.2.

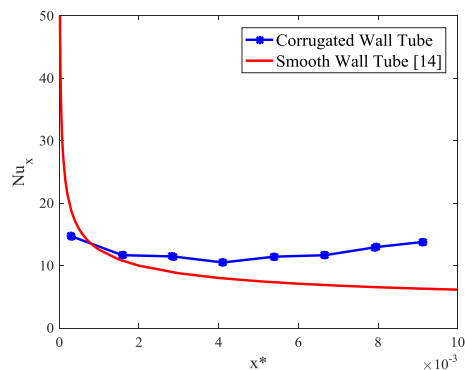


Figure 3e: Local Nusselt number versus the dimensionless abscissa for $Re = 89$ - Tube No.3.

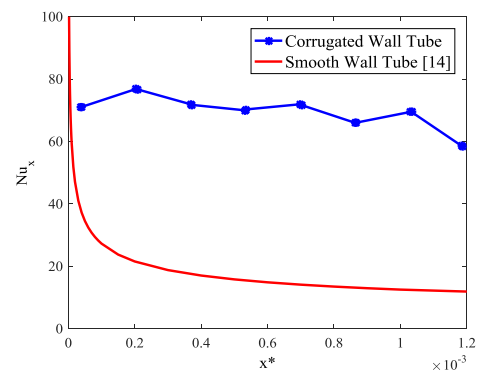


Figure 3f: Local Nusselt number versus the dimensionless abscissa for $Re = 763$ - Tube No.3.

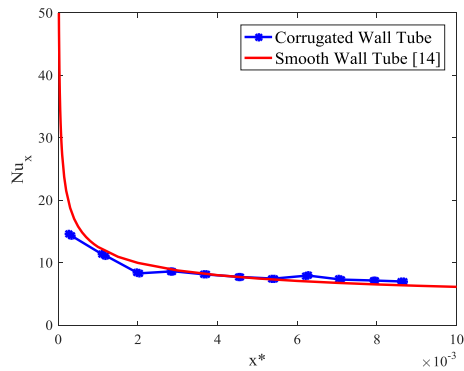


Figure 3g: Local Nusselt number versus the dimensionless abscissa for $Re = 100$ - Tube No.7.

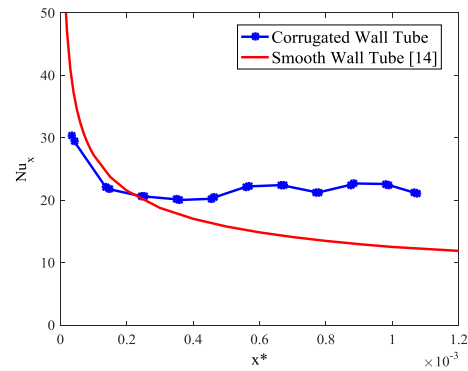


Figure 3h: Local Nusselt number versus the dimensionless abscissa for $Re = 909$ - Tube No.7.

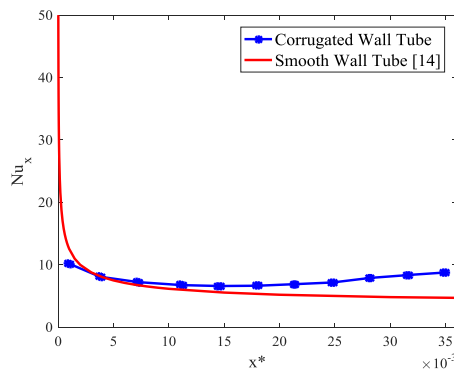


Figure 3i: Local Nusselt number versus the dimensionless abscissa for $Re = 26$ - Tube No.8.

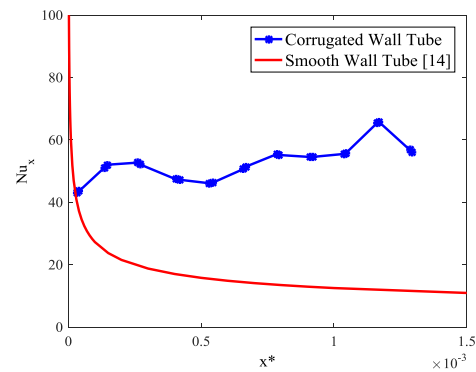


Figure 3l: Local Nusselt number versus the dimensionless abscissa for $Re = 735$ - Tube No.8.

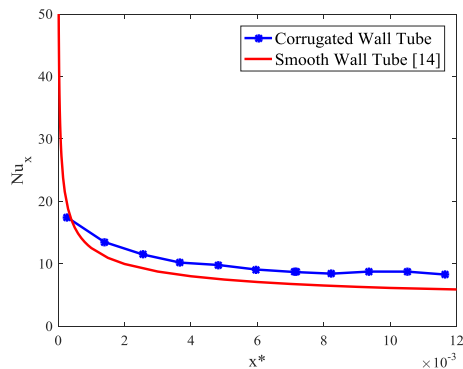


Figure 3m: Local Nusselt number versus the dimensionless abscissa for $Re = 74$ - Tube No.9.

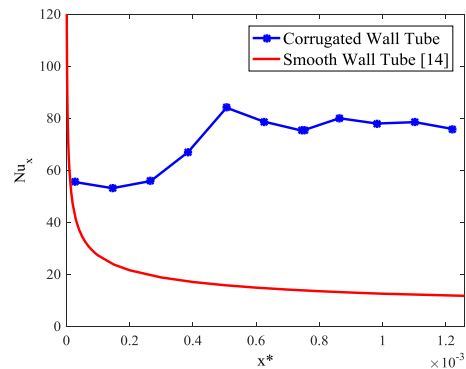


Figure 3n: Local Nusselt number versus the dimensionless abscissa for $Re = 762$ - Tube No.9.

In order to highlight the effect of the corrugation depth, it is meaningful to compare the heat transfer performance among pipes having the same corrugation pitch and different corrugation depths. In figure 4 the Nusselt number averaged over the heated length versus Reynolds number is reported for the tubes having the same corrugation pitch ($l=13$ mm (a), $l=18$ mm (b) and $l=29$ mm (c)) and different corrugation depths ($e = 0.6, 0.8$ and 1 mm). These data confirm that the corrugation depth is a crucial parameter in increasing the heat transfer performance for the pipes under test. The results also highlight

that for a given corrugation pitch, the critical condition can be described by a precise value of the corrugation depth. Under a specific value of the corrugation depth e , the wall corrugation effects are almost negligible, while above that value the enhancement does not depend on the severity of the wall corrugation. Moreover, it is possible to notice that for the pipes with corrugation depth equal to 0.8 and 1 mm a much anticipated onset of transition from the laminar to an unstable flow regime occurs. In particular, a clearly departure from the smooth wall behavior due to the onset of an unstable regime occurs in the range $100 < Re < 250$; in addition, it is also possible to observe that the transition from laminar to unstable regime probably occurs through a sequence of intermediate state as found in [8]. In fact, a possible second transition it is present in the range $600 < Re < 800$.

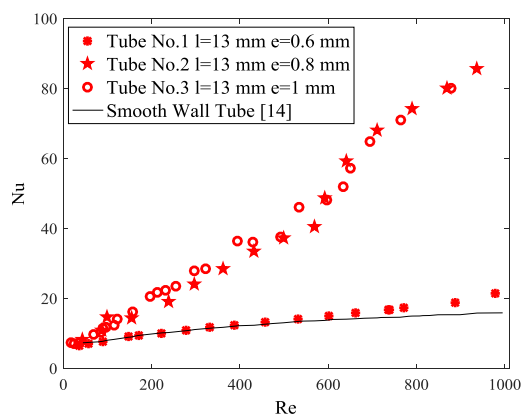


Figure 4a: Average Nusselt number for tubes No.1, 2 and 3 ($l=13$ mm) versus Reynolds number and comparison with the smooth wall tube behavior [14].

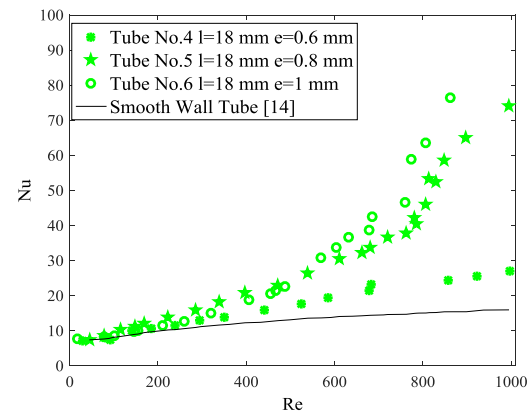


Figure 4b: Average Nusselt number for tubes No.4, 5 and 6 ($l=18$ mm) versus Reynolds number and comparison with the smooth wall tube behavior [14].

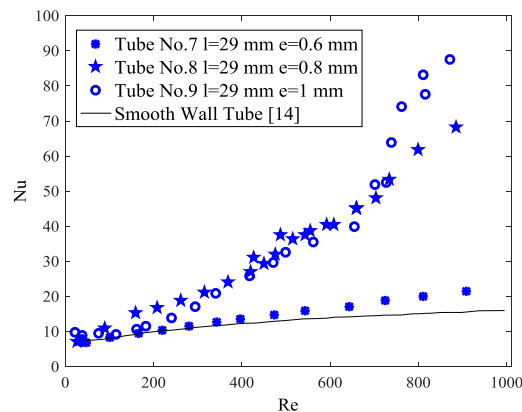


Figure 4c: Average Nusselt number for tubes No.7, 8 and 9 ($l=29$ mm) versus Reynolds number and comparison with the smooth wall tube behavior [14].

Regarding the effect of the corrugation pitch instead, in figure 5 the average Nusselt number versus the Reynolds number is reported for groups of tubes characterized by the same corrugation depth ($e=0.6$ mm (a), $e=0.8$ mm (b) and $e=1$ mm (c)) and by different corrugation pitches ($l=13$, 18 and 29 mm). The effect of the corrugation pitch l on the heat transfer performance of the pipes under test is less defined. For the lowest corrugation depth value hereby considered ($e=0.6$ mm) the best performance are associated to intermediate pitch value ($l=18$) while for the highest corrugation depth values ($e=0.8$ mm and $e=1$ mm) the pipe with the intermediate corrugation pitch presents the worst performance while the best performance is associated to the lowest pitch value ($l=13$ mm). Thus, from these data, it is not possible to establish a clear connection between the corrugation pitch effect and the heat transfer performance.

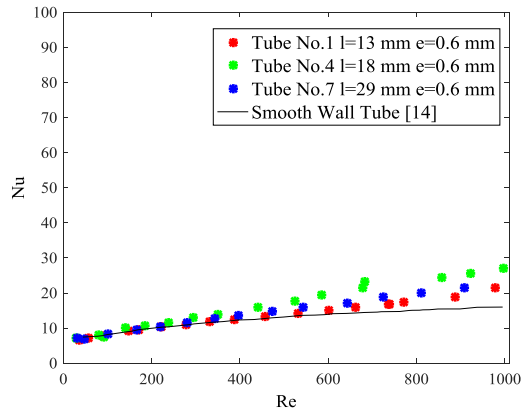


Figure 5a: Average Nusselt number for Tubes No.1, 4 and 7 ($e=0.6$ mm) versus Reynolds number and comparison with the smooth wall tube behaviour [14].

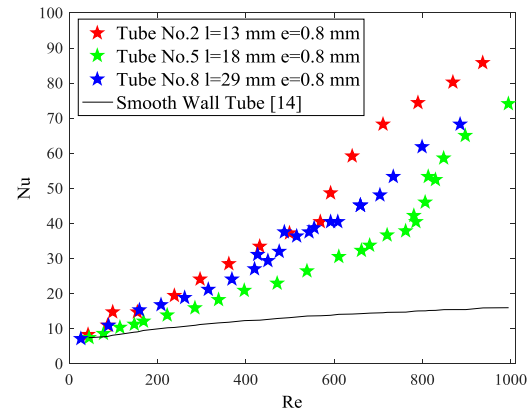


Figure 5b: Average Nusselt number for Tubes No.2, 5 and 8 ($e=0.8$ mm) versus Reynolds number and comparison with the smooth wall tube behaviour [14].

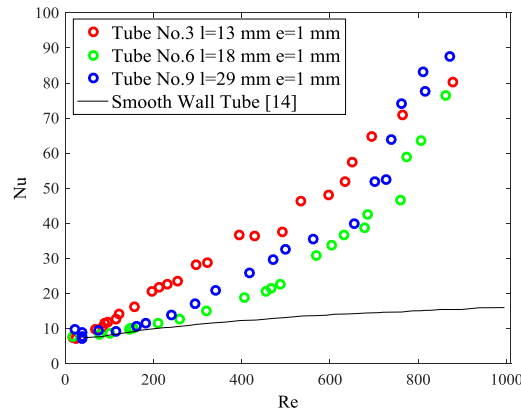


Figure 5c: Average Nusselt number for Tubes No.3, 6 and 9 ($e=1$ mm) versus Reynolds number and comparison with the smooth wall tube behaviour [14].

The limit of considering the average Nusselt number over the heated length is that the results obtained are valid only for the specific length under test: in order to make the results more general the asymptotic Nusselt number Nu_a was considered in the data processing. In fact, for the experimental configurations here studied, the thermally fully developed conditions were expected to be reached in the downstream region of the heated section. Regarding the measurements showing spatial oscillations in the local Nusselt number distribution, an average value in the fully developed region was considered.

The asymptotic thermal performances of all the nine tubes tested are compared in figure 6. The data confirm the predominance of the effect of the corrugation depth in promoting the heat transfer enhancement. Moreover, the data highlight the significant improvement of the convective heat transfer coefficient achievable with the cross-helix corrugation profile

The performance of the enhanced geometry was appropriately defined by the asymptotic heat transfer enhancement defined as follows:

$$\varepsilon_{h,\infty} = \frac{Nu_a}{Nu_{a,0}} \quad (6)$$

where the subscript 0 refers to reference problem, i.e. the analytical solution available for the thermally and hydrodynamically fully developed laminar flow problem [14] in straight smooth wall tubes under the uniform wall heat flux boundary condition.

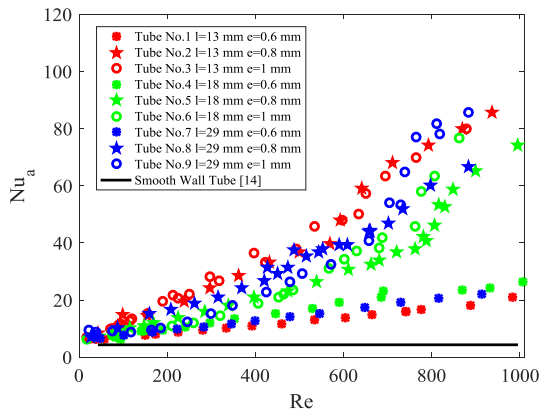


Figure 6: Asymptotic Nusselt number for the tubes under test and comparison with the smooth wall tube behaviour.

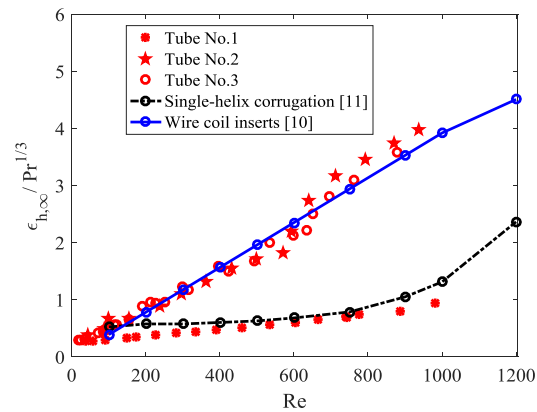


Figure 7: Heat transfer enhancement for the tubes No.1, 2 and 3 and comparison with the results obtained by Rainieri et al. [11] and by Garcia et al. [10].

In figure 7 the heat transfer enhancement performances are reported for the three tubes with corrugation pitch l equal to 13mm: in the same figure the results obtained by Rainieri et al. [11] for similar roughness (i.e. single-helix). It is interesting to compare these behaviors with the effects promoted by wire coil inserts since they have many points in common with severe wall corrugation (Garcia et al. [10]). Table 2 reports the geometric characteristics of these pipes. The results point out that the heat transfer enhancement obtained for the two cross-helix corrugated tubes with the two highest corrugation depth here tested exceeds the ones obtained for the single-helix corrugation by Rainieri et al. [11]. These data confirm the more effective capability of the cross-helix corrugation profile with respect to the single-helix one in anticipating the transition from the laminar flow regime. Moreover, there is a good matching between the results obtained for the cross-helix corrugation and the data achieved in [10] for the wire coils.

Table 2: pipes characteristics analysed in [11] and [10].

	l (mm)	e (mm)	l/D	e/D	Ω
Single-helix [11]	13	1.0	1.08	8.33×10^{-2}	6.41×10^{-3}
Wire coil inserts [10]	18	0.6	1.29	4.29×10^{-2}	1.43×10^{-3}

4. Conclusions

The forced convective heat transfer in cross-helix corrugated wall tubes was studied in the Reynolds and Prandtl number ranges $25 \div 1000$ and $115 \div 150$ respectively, by adopting ethylene glycol and water-ethylene glycol mixtures as working fluids. The aim of the research was to deepen the heat transfer enhancement achievable by using this passive technique. In particular, the effect of both the corrugation depth and of the corrugation pitch was analysed, by pointing out the correlated heat transfer enhancement effect: nine tubes, having different corrugation pitches and depths were tested. The results are expressed in terms of local, average and asymptotic Nusselt number and of heat transfer enhancement over the smooth wall tube behavior. The data show that the corrugation depth is a crucial parameter in increasing the heat transfer performance of the pipes under test. Under a specific value of the corrugation depth e the wall corrugation effects are almost negligible, while above that value the enhancement doesn't depend on the severity of the wall corrugation. For the pipes with corrugation depth equal to 0.8 and 1 mm a very early onset of transition from the laminar to an unstable flow regime occurs. In particular, a clearly departure from the smooth wall behavior occurs in the range $100 < Re < 250$ and a probable second transition is present in the range $600 < Re < 800$. These data confirm that the

transition from laminar to unstable regime can occur through a sequence of intermediate state as observed by Guzman and Amon [8]. On the other hand, the effect on heat transfer of the corrugation pitch, resulted instead less clear for the nine pipes under test. The heat transfer enhancement performances were also compared to the results obtained by Rainieri et al. [11] and by Garcia et al. [10] for pipes with single-helix corrugation and wire coil inserts. The results confirm the more effective capability of the cross-helix corrugation profile with respect to the single-helix one to enhancing the transition from the laminar flow regime by significantly pushing the heat transfer tube's performance over the smooth wall tube behavior. In addition, they show a good matching with the data achieved in [10] with the wire coil inserts. In order to complete the analysis on this passive heat transfer enhancement technique, a further study that accounts for the pressure drop augmentation will be addressed.

Acknowledgments

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