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Convective heat transfer to temperature dependent property fluids in the entry region of corrugated tubes

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Abstract

The present experimental investigation is aimed to the analysis of the thermal performances of corrugated wall tubes employed in a broad variety of industrial applications in order to intensify the convective heat transfer. Both axial symmetrical and helical corrugations with different pitch values, have been considered in the present analysis. In particular, the phenomena arising due both to the wall roughness and to fluid property variation are investigated in the thermal entrance region in the Reynolds number range 90–800. The comparative analysis of the behaviour of the two types of corrugation enables to draw interesting conclusions about the effects of the corrugation pattern on the heat transfer enhancement mechanism. The data show that the helical corrugation induces significant swirl components to which, however, an as much significant heat transfer enhancement is not associated. The variation of the fluid physical properties with temperature, promotes the transition to an unstable flow. Regarding to this phenomenon a critical local Reynolds number, proportional to the dimensionless corrugation pitch has been identified. The dependence of the local Nusselt number on the corrugation pitch has been investigated, too.

Keywords: Convection; Enhancement; Experimental

1. Introduction

In a wide variety of engineering applications the working liquid undergoes thermal processes under the laminar flow regime, where only low convective heat transfer coefficients are reachable. For instance, this situation is encountered for highly viscous Newtonian fluids, like oils and many fluid food products. In order to reduce heat exchanger size and operational cost, in these cases it may be necessary to apply some techniques for increasing the rate of heat transfer. Among the many different methods which have been considered for enhancing the heat transfer in forced convection [1], the techniques which promote secondary recirculation flows, by inducing non-axial velocity components, appear very interesting for practical applications. This goal can be obtained by using different methods: spiral fins applied to the heat transfer surface, metallic twisted tape inserted into the tube, spirally roughened walls.

Watkinson et al. [2] have experimentally investigated the forced convection of oils in integral straight and spiral fin tubes. They observe that for low Reynolds number values, the helical components in the fluid flow cause significant augmentation in the heat transfer coefficient. For the laminar flow they suggest a correlation which predicts a dependence of the average Nusselt number on the fin pitch raised to the power −0.5. They conclude that the spiral components produce a stronger effect in laminar than in turbulent flow. For higher Reynolds number values, the secondary swirl flow due to the fins causes instability by inducing an early transition to the turbulent flow, to which a strong heat transfer enhancement is associated. For small helix angles, almost the same dependence on the helix pitch has been predicted by Ravigururajan and Bergles [6] for the fully turbulent flow. In fact they suggest a correlation where the Nusselt number is expressed as a function of the rib pitch raised to the −0.21 power and of the helix

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Nomenclature

c_p	specific heat at constant pressure (kJ kg^{-1} K^{-1})
D_{env}	maximum internal diameter (m)
e	corrugation depth (m)
h	convective heat transfer coefficient (W m^{-2} K^{-1})
K	viscosity ratio ($K = \mu_b/\mu_w$)
L_D	hydrodynamic entrance length (m)
Nu	Nusselt number (hD_{env}/λ)
p	helix pitch (m)
p^*	dimensionless helix pitch (p/D_{env})
Pr	Prandtl number ($\mu c_p/\lambda$)
q	heat flow rate (W/m^2)
Re	Reynolds number (wD_{env}/ν)
T	temperature (°C)
ΔT	dimensionless temperature difference ($(T_{up} - T_{down})/(T_o - T_i)$)
w	mean axial velocity (m/s)
x	axial coordinate (m)
x^*	dimensionless axial coordinate ($x^* = x/D_{env}RePr$).

Greek symbols

μ	dynamic viscosity (Pa s)
ν	kinematic viscosity (m^2/s)

Subscripts and superscripts

b	evaluated at the bulk temperature
down	evaluated along the lower cylinder generatrix
cp	constant property
i	inlet
o	outlet
up	evaluated along the upper cylinder generatrix
w	evaluated at the wall temperature
x	local value
—	averaged along the heated length

angle raised to the power 0.29, that is a dependence on p close again to the -0.5 power.

The numerical results reported by Date and Singham [3] show that for highly viscous fluids heat transfer enhancement to up about 70% can be obtained by using swirl flow inserts. Hong and Bergles [4], by providing experimental results, observe that the fully developed laminar flow Nusselt number decreases with the helix pitch raised to the power 0.622. Manglik and Bergles [5] outline two different flow regimes: for low Reynolds number values, the flow occurs in a spiral pattern determined by the tape's surface; by increasing the Reynolds number the centrifugal force due to the rotating components produces a secondary flow which remixes the fluid. A significant heat transfer enhancement is associated to both these flow regimes. These Authors suggest a correlation where the laminar flow mean Nusselt number is expressed as a function of the Swirl number to the 0.767 power, that is they suggest a dependence on the helix pitch nearly to the -0.5 power.

Regarding spirally enhanced tubes, the analysis of the bibliography recently reported by Garimella and Christensen [7] shows that most of the published experimental works are focused on the turbulent flow, also because this type of corrugation produces an early transition to this flow regime. Regarding to this aspect, Rainieri et al. [8], by reporting experimental results, have confirmed the observation made by Watkinson et al. [2] on integral inner-fin tubes in the laminar and transition regime: for Reynolds number values lower than the critical one usually taken for the smooth wall, a significant enhancement of both the Nusselt number and the friction factor, which seems associated to an early departure from the laminar regime, occurs. For the turbulent flow, Garimella et al. [9] report heat transfer enhancement up to about 500% due to the augmentation of the heat transfer associated with the mechanical surface working, which causes the onset of swirl components into the main flow and the periodic destruction of the boundary layer. Regarding to these phenomena Nakayama et al. [10] formulate the hypothesis that, according to the angle of the corrugation to the tube axis, different flow regimes can occur: for low helix angles the fluid flows following the corrugation pattern close to the wall, while for high angles the fluid tends to cross the corrugation by inducing a periodic separation of the boundary layer which induces important augmentation of the heat transfer coefficient. Also the results of Li et al. [11], derived by visualization tests by means of hydrogen bubbles technique, confirm this behaviour. Garimella et al. [9] present visualization results, for annuli with spirally fluted, indented and ribbed inner tubes. For low Reynolds number values, when the laminar flow persists, they observe that for the fluted and the ribbed corrugation types, close to the inner wall the fluid follows the helical wall profile, while for the indented corrugation type the axial components dominate. The Authors associate this behaviour to the angle of the indented wall considered, which is close to 90°. More recently Ravigururajan and Bergles [12] reported the results of an experimental investigation based on the flow visualization in tubes with wire coils inserts, used to simulate the behaviour of a spirally enhanced wall. They observe that, by decreasing the Reynolds number, the thickness of the rotational boundary layer increases, while the angle of rotation tends toward the rib helix. Furthermore they observe that the angle of rotation increases with increasing roughness height. These experimental results underline also that the hydrodynamic developing length seems to be insignificant even at very low Reynolds number.

The present paper is focused on forced convection in the thermal entry region of both transverse and helical corrugated tubes for a highly viscous Newtonian fluid. The study is intended to gain an insight on the mechanism which is responsible of the heat transfer enhancement in the low Reynolds number range. Moreover the interaction between the helical corrugation and the property variation effects is investigated. In fact, the physical phenomena governing the heat transfer enhancement for spirally enhanced tubes have not been wholly understood, despite the fact that many empirical correlations have been suggested for predicting their overall thermal performance. The data reported in literature, in fact, regard the mean Nusselt number and friction factor over the whole tube length. Furthermore the flow visualization studies regard isothermal conditions, while the local Nusselt number distributions reported by Rainieri and Pagliarini [13] for non-Newtonian fluids, seem to point out a significant property variation effect.

2. Experimental apparatus

The experimental facility consists essentially of an hydraulic loop, where the test section is placed horizontally, and of a data acquisition system. The components of the apparatus are schematically represented in Fig. 1. The test section is equipped to determine the heat transfer coefficient distribution in the thermal entrance region, while the measure of the local friction factor would require a different instrumentation.

The tests have been performed on six stainless steel tubes. Four of them have a one start helical wall corrugation while two of them have a transverse wall corrugation. All the tubes have a maximum internal diameter D_{env} of 14 mm, a wall thickness of 1 mm, a corrugation depth e of 1.5 mm, while different corrugation periodicity has been considered: namely $p = 16$, 32, 48 and 64 mm. The meaning of the parameters used to describe the geometry under investigation are reported in a schematic drawing in Fig. 2, while two of the tubes tested are shown in Fig. 3. The geometrical

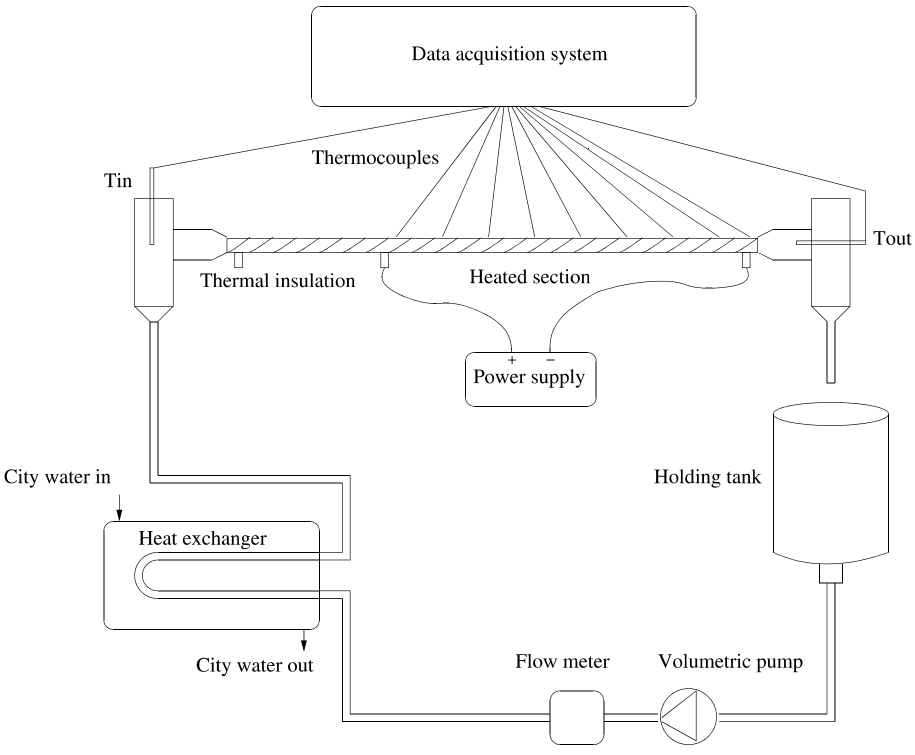

Fig. 1. Schematic drawing of the experimental apparatus.

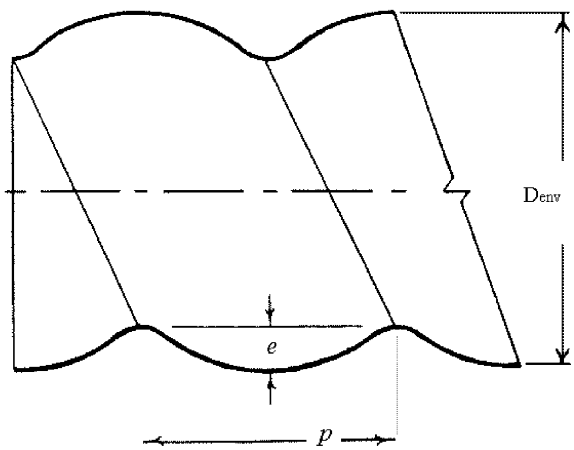

Fig. 2. Schematic drawing of the geometry tested.

Fig. 3. Helical and transverse corrugated tubes tested.

Table 1
Test section parameters

Tube label	Corrugation type	Corrugation pitch p (mm)
Hp16	Helical	16
Hp32	Helical	32
Hp48	Helical	48
Hp64	Helical	64
Tp16	Transverse	16
Tp32	Transverse	32

parameters of the tubes under test are specified in Table 1, along with the label used to identify them in the discussion of the results.

The tube wall is equipped with copper fin electrodes which are connected to a power supply, type HP 6671A, working in the ranges 0–8 V and 0–220 A. Because of the low thermal conductivity of the wall material, the Joulean dissipation in the tube wall enables to approximate the condition of uniform heat flux at the fluid boundary. The heated section is about 2 m long and it is preceded by an unheated development approach section of about 1 m. The whole section is thermally insulated by a 35 mm thick rubber sponge layer, to minimize the heat exchange to the environment.

The wall temperature has been measured through 42 copper–constantan thermocouples, previously calibrated and connected to a multichannel ice point reference, type KAYE K170-50C. The temperature probes have been placed externally on the crest of the corrugation along the upper and lower generatrix of the envelope cylindrical surface. The internal wall temperature has been then obtained by solving the steady state heat conduction problem with heat generation in the tube wall. The inlet temperature has been measured by a thermocouple probe directly immersed in the fluid. The bulk temperature at any location in the heat transfer section has been calculated from the power supplied to the tube and heat losses through the insulation, whose thermal resistance had been estimated in a preliminary calibration of the apparatus, while the outlet temperature has been checked also by placing a thermocouple in the mixing tank at the end of the heated section. Flow rates were obtained by measuring the time needed to fill a volumetric flask placed at the outlet of the test section, while a magnetic flowmeter incorporated in the loop was simply used as a flow indicator. The data acquisition system consists essentially of a high precision multimeter (type HP 3458A) connected to a switch control unit (type HP 3488A) driven by a Personal Computer.

The working fluid used is ethylene glycol, whose properties exhibit a strong temperature dependence. The most significant variation is shown by the coefficient of viscosity, as it appears from the data reported in Table 2, derived from [14], regarding the temperature range of interest in the present investigation. In the data reduction the maximum internal diameter (the envelope diameter D_{env}) has been used as the characteristic length to evaluate both the Reynolds and the Nusselt number and the heat transfer area, taken equal to the envelope cylinder surface. The inlet temperature has been used for evaluation of all fluid properties, unless otherwise specified.

The uncertainties associated to the directly measured quantities have been assumed to be 0.2 °C for temper-

Table 2
Properties of the working fluid

Ethylene glycol				
T (°C)	ρ (kg/m^3)	μ (Pa s)	λ (W/m K)	c_p (J/kg K)
10	1120	38.4E−3	0.257	2336
70	1078	3.98E−3	0.265	2606

ature, and <0.1 °C for temperature difference. For what concerns mass–flow rate and wall heat flux a percent uncertainty less than 1% has been estimated. Regarding the relevant dimensionless parameters, the uncertainties have been calculated by using the usual propagation of errors procedure [15]. By considering the bias error associated to the measurement procedure, the maximum uncertainty for the Nusselt number and the Reynolds number has been estimated to be respectively ±4% and ±3%.

3. Results

Different Reynolds numbers, in the range $90 < Re < 800$, have been considered, as well as different wall heat flux values, in order to point out the effect of property variation. The test conditions of the runs are reported in Tables 3 and 4. It is expected that density variation causes natural convection effects, while modifications in the boundary layer are expected due to viscosity variation. The intensity of the former effect has

Table 3
Test conditions of the runs with thermal developing flow with developed velocity profile ($L_D = 1$ m) for the helical corrugation tubes

Helical corrugation							
Tube Hp16				Tube Hp32			
Re	q (W/m^2)	$\overline{K}$	$\overline{\Delta T}$	Re	q (W/m^2)	$\overline{K}$	$\overline{\Delta T}$
110	344	1.1	≃0	115	358	1.1	≃0
90	1.1E4	3.2	0.5	95	1.09E4	3.1	0.4
200	1245	1.2	≃0	220	1240	1.2	≃0
180	1.63E4	5.4	−0.2	180	1.27E4	4.1	0.1
380	3600	1.5	≃0	450	3640	1.5	≃0
420	1.3E4	2.7	−0.1	405	1.3E4	3.1	0.1
780	1860	1.2	−0.1	785	4820	1.5	0.1
770	1.31E4	2.1	≃0	700	1.65E4	2.6	0.1
Tube Hp48				Tube Hp64			
Re	q (W/m^2)	$\overline{K}$	$\overline{\Delta T}$	Re	q (W/m^2)	$\overline{K}$	$\overline{\Delta T}$
115	354	1.1	≃0	115	630	1.2	≃0
90	1.09E4	3.1	0.5	90	1.09E4	3.1	0.5
230	1010	1.2	≃0	235	1200	1.3	≃0
180	1.63E4	5.1	0.1	185	1.6E4	4.7	0.1
495	3670	1.5	≃0	480	2045	1.2	−0.4
390	1.63E4	4.6	0.1	390	1.64E4	4.5	≃0
685	5325	1.7	≃0	700	5235	1.8	≃0
690	1.64E4	3.4	0.1	690	1.65E4	3.4	≃0

Table 4
Test conditions of the runs with thermal developing flow with developed velocity profile ($L_D = 1$ m) for the transverse corrugation tubes

Transverse corrugation							
Tube Tp16				Tube Tp32			
Re	q (W/m^2)	$\overline{K}$	$\overline{\Delta T}$	Re	q (W/m^2)	$\overline{K}$	$\overline{\Delta T}$
120	550	1.2	0.1	110	530	1.2	0.1
100	1.1E4	2.7	0.5	90	1.05E4	2.8	0.6
245	1285	1.2	≃0	220	1210	1.2	≃0
195	1.3E4	3.9	0.9	185	1.24E4	3.1	1.0
460	3525	1.4	0.4	440	3365	1.4	0.8
400	1.3E4	2.2	1.5	415	1.21E4	2.4	1.9
770	1400	1.1	0.5	770	3415	1.3	0.3
780	1.43E4	2.9	1.0	740	1.3E4	1.9	2.1

been monitored by evaluating the parameter $\Delta T = (T_{\mathrm{up}} - T_{\mathrm{down}})/(T_{\mathrm{o}} - T_{\mathrm{i}})$, that is the dimensionless difference between the upper and the lower temperature on the tube section. The latter effect has been quantified by evaluating the parameter $K = \mu_{\mathrm{b}}/\mu_{\mathrm{w}}$, that is the ratio between the viscosity evaluated at the bulk and at the wall temperature. The values of these parameters, averaged along the heated length, are reported in Tables 3 and 4 for each run.

The analysis of the data reported in Tables 3 and 4 points out a substantial difference between the behaviour of the axial symmetrical corrugation and the helical one at high wall heat flux. More precisely, the transverse corrugation shows quite high $\overline{\Delta T}$ values, while, for the second type of corrugation, the values of this parameter are close to zero in all of the tests, with a few exceptions corresponding to the lowest Reynolds numbers ($Re \simeq 100$). The systematic nature of the phenomenon is here assumed as an indirect evidence that the helical corrugation induces swirl components into the flow which prevent a fluid stratification. This hypothesis is confirmed by direct observation made by several authors (Li et al. [11], Garimella et al. [9], Ravigururajan and Bergles [12]) for similar wall corrugations. Regarding the modification in the boundary layer induced by viscosity variation, the Nusselt number correction usually follows the property ratio scheme:

$$\frac{Nu}{Nu_{\mathrm{cp}}} = \left(\frac{\mu_{\mathrm{b}}}{\mu_{\mathrm{w}}}\right)^{n}. \tag{1}$$

For the exponent n, the value $+0.14$ has been extensively used to correlate experimental data for both the laminar and the turbulent flow of moderate and high Prandtl number liquids.

In order to understand the heat transfer enhancement mechanism nature, the local Nusselt number in the thermal entrance region has been compared for the helical and the transverse corrugation wall tubes in Figs. 4–7, for different Reynolds number as well as wall heat flux values versus the dimensionless abscissa $x^* = x/D_{\mathrm{env}} Re Pr$. In these figures (Figs. 4–7) and in the following ones, the solid line refers to the analytical solution holding for the smooth wall with uniform heat flux boundary condition and constant property fluid [16]. For each Re value, two sets of curves are considered. The first set refers to wall heat flux values low enough that the fluid property variation may be neglected, while for the second one, strong fluid property variation effects are to be expected, as the values of $\overline{K}$ and $\overline{\Delta T}$ reported in Table 1 confirm.

When considering low heat flux values, in the case of the transverse corrugation, the local Nusselt number approaches the smooth wall behaviour at low Reynolds numbers, while for $Re \geqslant 400$ it increases and it seems to approach a constant value which is higher for lower corrugation pitch values. The behaviour shown by the transverse corrugation for relatively high Re values seems to be ascribed to the periodic disruption of the hydrodynamic boundary layer, due to the transverse

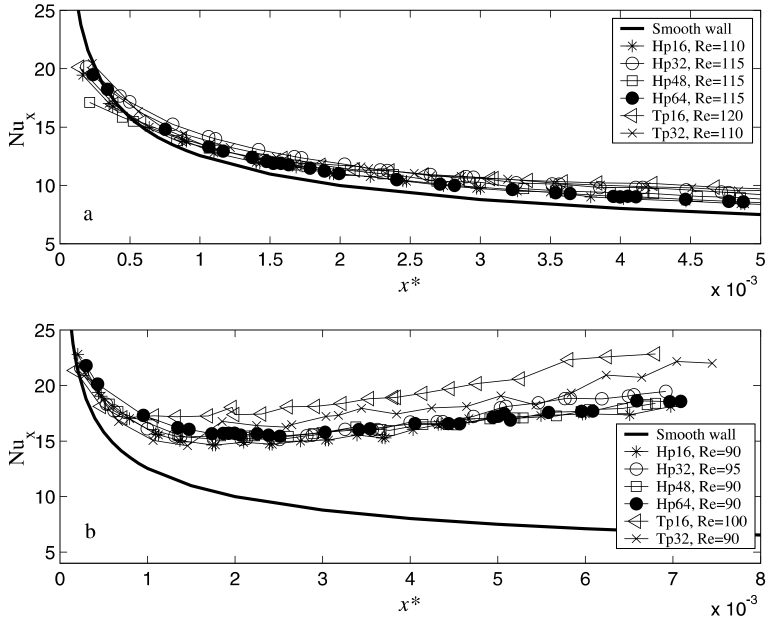

Fig. 4. Local Nusselt number for tubes with helical and transverse corrugation for the runs with $Re \simeq 100$: (a) low wall heat flux, (b) high wall heat flux.

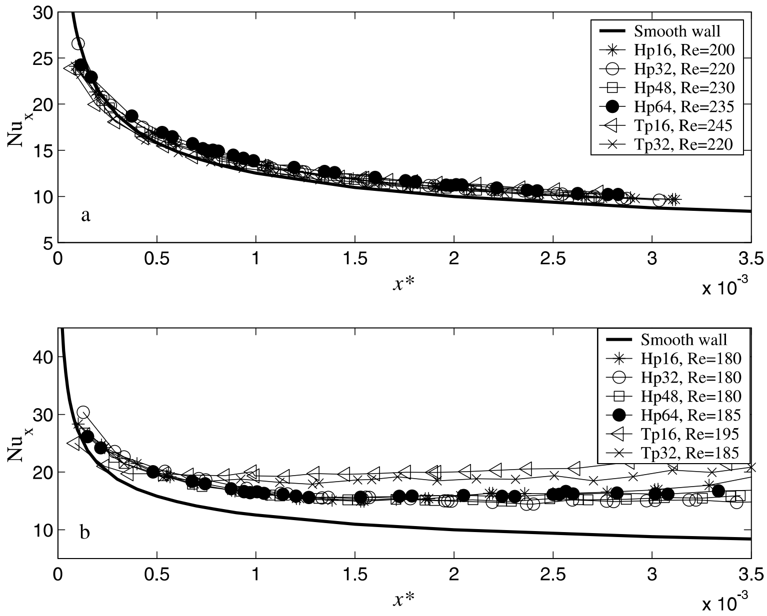

Fig. 5. Local Nusselt number for tubes with helical and transverse corrugation for the runs with $Re \simeq 200$: (a) low wall heat flux, (b) high wall heat flux.

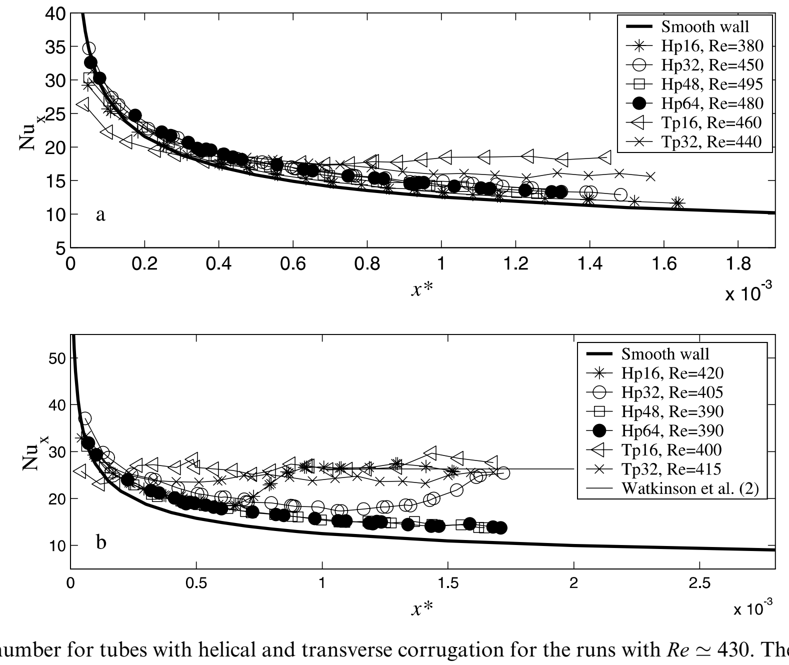

Fig. 6. Local Nusselt number for tubes with helical and transverse corrugation for the runs with $Re \simeq 430$. The thin solid line refers to the data inferred from the results reported by Watkinson et al. [2]: (a) low wall heat flux, (b) high wall heat flux.

corrugation, which promotes instability in the flow, thus leading to an early transition towards turbulent flow. Instead, in the case of the helical corrugation, the local Nusselt number approaches the smooth wall behaviour in the whole considered Reynolds number range. If compared with the transverse corrugation, therefore, the spirally shaped corrugation seems to favor the persistence of laminar flow, thus preventing the heat transfer enhancement associated with unstable or turbulent flow.

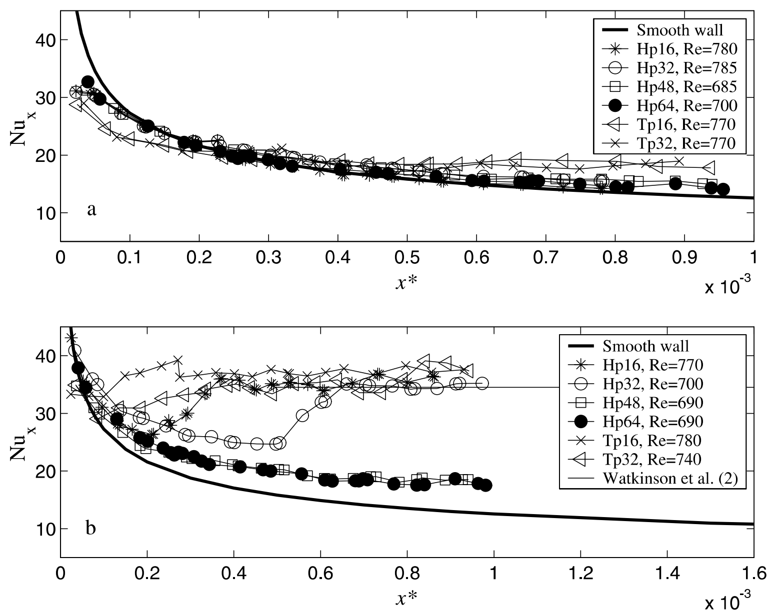

Fig. 7. Local Nusselt number for tubes with helical and transverse corrugation for the runs with $Re \simeq 740$. The thin solid line refers to the data inferred from the results reported by Watkinson et al. [2]: (a) low wall heat flux, (b) high wall heat flux.

When the wall heat flux increases, important changes take place in the Nusselt number distributions, depending on the Reynolds number value. For the helical corrugation wall, the local Nusselt number increase near the thermal inlet seems chiefly to be ascribed to the fluid viscosity variation. This interpretation is corroborated by the comparison between the measured Nusselt number values and the values given by Eq. (1), for the K values specified in Tables 3 and 4. Furthermore, tests performed on a smooth wall tube with the same working fluid and for comparable Reynolds number and wall heat flux values, showed Nusselt number increases within the range 20–30%, due to fluid property variation [17]. However the highest enhancement is achieved systematically by the wall with $p = 32$ mm. This suggest that the helical corrugation might have some effect in enhancing the convective heat transfer, however masked by the property variation effect. Towards the outlet, two different behaviours are shown by the results. For $Re \leqslant 200$, the Nusselt number increases gradually due again to fluid property variation, chiefly density. At higher Re values ($Re \geqslant 400$), for the tubes with $p = 16$ and 32 mm, the local Nusselt number shows a sudden increase and then reaches a nearly constant value near the end of the heated section. This behaviour seems to be ascribed to a transition from laminar to unstable and then turbulent flow. The property variation effects seem then to play a determinant role on the onset of the transition. This can be explained by associating the instability to modifications in the hydrodynamic boundary layer due to viscosity variation. An analogous observation has been made by Marner and Bergles [18], who investigated the heat transfer augmentation for highly viscous fluids in internally finned tubes and tubes with twisted tape inserts. They conclude that, in case of heating, the spiral fins magnify the viscosity ratio effect and produce swirling into the flow at higher Graetz number values, that is at lower distance from the thermal inlet. Inspection of Figs. 6 and 7 points out that, for both the transversely and the spirally corrugated walls, the Nusselt number reaches a nearly common value, dependent on the Reynolds number. This result confirms the experimental results of Watkinson et al. [2] who, by analyzing several tubes with internal helical or longitudinal fins for uniform wall temperature boundary condition, concluded that, when transition to turbulent flow occurs, all the walls behaves nearly in the same way. In the same figures (Figs. 6 and 7) the Nusselt number values inferred from the correlation suggested by Watkinson et al. [2] for high spiral integral inner-fin tubes, are also shown. The substantial agreement with the asymptotic values of the present investigation seems to confirm that, for the considered cases, at the end of the heated section a turbulent flow condition is really reached. A further confirmation of this assumption comes from the results of Rainieri et al. [8] who, by analyzing the same type of spirally enhanced tubes with the same boundary condition and test section length, noted that the mean Nusselt number along the heated section, for $Re > 800$, agrees with the correlation suggested by Richards et al. [19] for fully turbulent flow. For higher pitch values ($p = 48$ and 64 mm), the transition is not reached within the length of the test section, even if the oscillations in the Nusselt number close to the exit seem to show that some instability is setting up into the flow. To get further insight about the limit of the laminar flow regime, new tests were performed, by considering a longer heated section. In these tests the heating was started at the tube inlet, thus producing a condition of simultaneous development of both the hydrodynamic and the thermal boundary layers. The test conditions are reported in Table 5. In Fig. 8 the local Nusselt number which refers to the runs specified in Table 5 is compared with the curves considered above, for the same tubes and for almost the same Reynolds number value, which include the effect of a region of dynamic development upstream of the heated section. The data show that the transition occurs also for these tubes, but it happens at greater distance from the starting of the heating. Furthermore, the presence of a section of dynamic development produces negligible effects for this kind of wall corrugation, as already observed by Ravigururajan and Bergles [12].

Also the results obtained for high wall heat flux confirm the greater ability of the transverse corrugation wall in enhancing convective heat transfer. For this kind of corrugation, at Reynolds number values greater than 400, the Nusselt number assumes a nearly constant value all over the heated length, denoting that, because of the strong property variation, a turbulent flow condition is reached few diameters downstream of the thermal inlet.

In order to put in evidence the effect of the helix pitch in the region where the laminar flow persists with the presence of swirl components, all the data are reported versus the usual dimensionless abscissa x^* in Fig. 9. The same results are reported in Fig. 10, versus a modified dimensionless abscissa $x^* p^{*-0.5}$, where $p^* = p/D_{\mathrm{env}}$ is the dimensionless pitch, in which the dependence of the parameter p to the -0.5 power, as suggested by Watkinson et al. [2] and Manglik and Bergles [5] when a swirl flow takes place in laminar flow, has been considered. From the analysis of the data, it comes out that, by

Table 5
Test conditions of the runs with simultaneously developing flow ($L_D = 0$ m) for the $p = 48$ and 64 mm helical corrugation tubes

Helical corrugation							
Tube Hp48				Tube Hp64			
Re	q (W/m^2)	$\overline{K}$	$\overline{\Delta T}$	*Re*	q (W/m^2)	$\overline{K}$	$\overline{\Delta T}$
720	1.1E4	2.4	0.1	730	1.1E4	2.6	−0.1

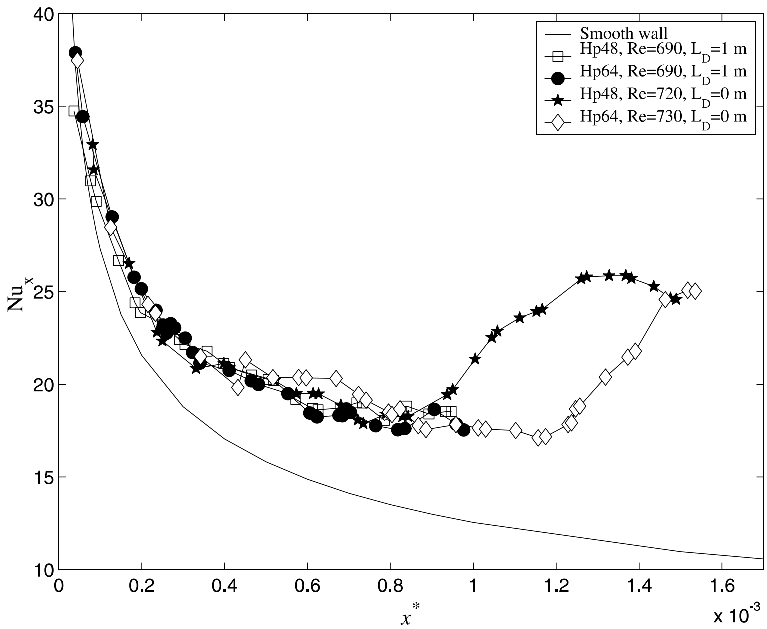

Fig. 8. Local Nusselt number regarding the tubes with the $p = 48$ and 64 mm helical corrugation for the runs with simultaneously developing flow and thermally developing flow with developed velocity profile.

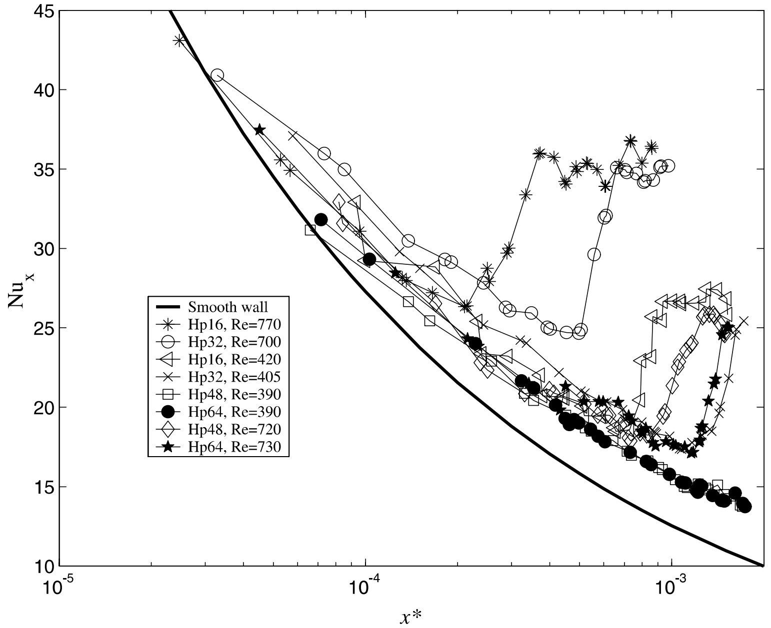

Fig. 9. Local Nusselt number versus x^* for the helically corrugated tubes regarding the runs with high heat flux.

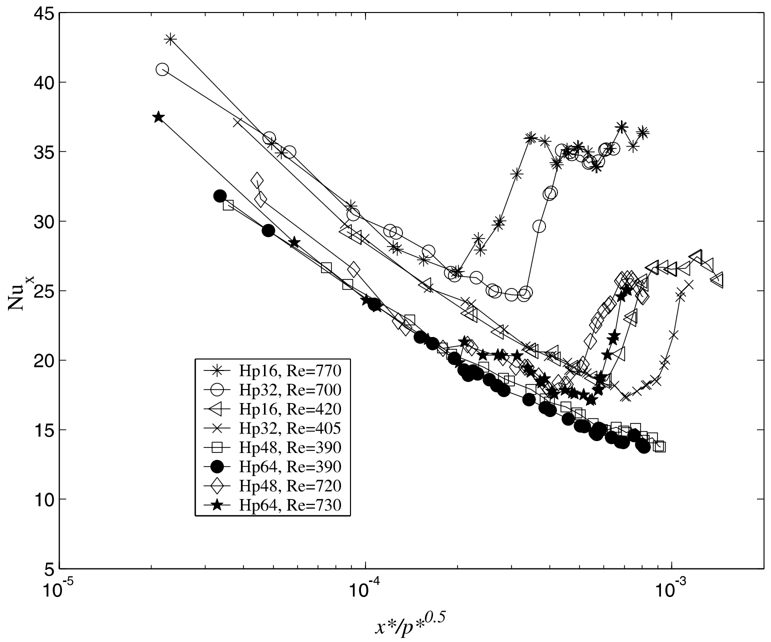

Fig. 10. Local Nusselt number versus the modified dimensionless abscissa for the helically corrugated tubes regarding the runs with high heat flux.

using the modified abscissa, the data corresponding to the laminar region seem to gather into two groups, one for $p \leqslant 32$ and one for $p \geqslant 48$. Certainly, the corrugation pitch values tested in the present investigation, are few to allow a definitive conclusion to be drawn. However, the observed different behaviour for low and high pitch values seems to confirm the presence of two swirl flow regimes, depending on the corrugation angle, as shown

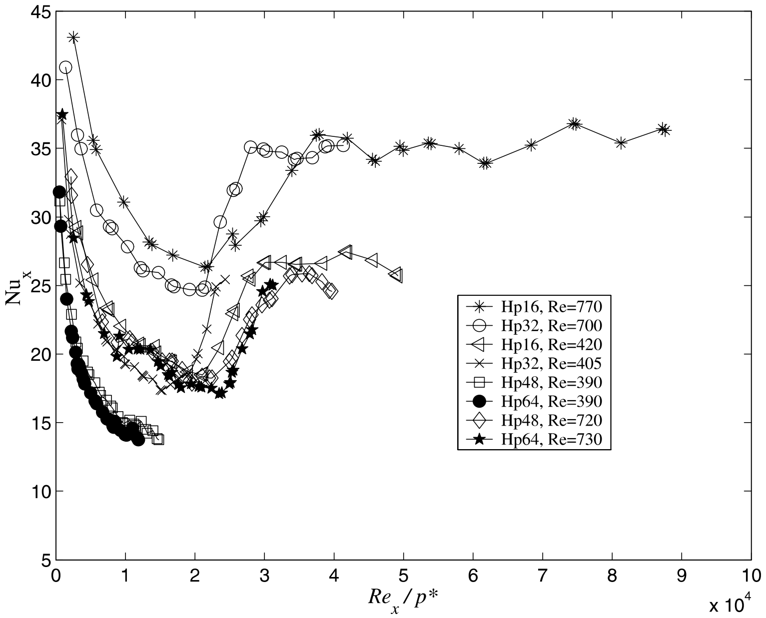

Fig. 11. Local Nusselt number versus the local Reynolds number for the helically corrugated tubes regarding the runs with high heat flux.

by the flow visualization results of Li et al. [11] and Ravigururajan and Bergles [12].

In order to put in evidence the effect of the corrugation pitch on the onset of the transition the data corresponding to the high heat flux are reported also versus a local Reynolds number $Re_x = wx/\nu$ divided by a dimensionless pitch p^* in Fig. 11. The data seem to identify a critical local Reynolds number, depending on the dimensionless pitch ($Re_{x,c} \simeq 2 \times 10^4 p^*$), beyond which the laminar swirl flow turns into an unstable flow due to the phenomena discussed in the argumentation above reported. The critical local Reynolds number results to be proportional to the dimensionless pitch, that is to the number of perturbation encountered by the fluid within a given length. Similar consideration are reported by Garimella and Christensen [7] for annuli with spirally fluted inner tubes, who underline the great significance of the number of perturbations per unit length encountered by the fluid in justifying the increase of both the friction factor and the Nusselt number at low pitch values. This agrees also with the results of Ravigururajan and Bergles [12] who underlined that the turbulence decreases by decreasing the helix angle, that is by increasing the helix pitch.

4. Conclusions

Convective heat transfer in the thermal entrance region of horizontal tubes having spirally enhanced wall has been experimentally investigated in the Reynolds number range $90 < Re < 800$. Four different corrugation pitch values have been tested. For comparative purposes, the investigation has been extended to tubes having an axial symmetrical wall corrugation. Ethylene glycol has been employed as working fluid. It is a highly viscous fluid and its viscosity coefficient shows a strong temperature dependence. The critical analysis of the data obtained in the present investigation allows the following conclusions to be drawn, holding at least for highly viscous fluids, like the one used in the present study:

– for not too low Reynolds numbers, ($Re \geqslant 200$), the helical corrugation induces significant swirl components to which, however, an as much significant heat transfer enhancement is not associated;
– the variation of the fluid physical properties with temperature, besides producing important heat transfer augmentation, although comparable with the behaviour reported for the smooth wall, promotes the transition to an unstable flow;
– for the helical corrugation, the critical local Reynolds number, defined by considering the axial coordinate starting at the thermal inlet section, is proportional to the dimensionless corrugation pitch;
– in the turbulent flow the Nusselt number is practically independent on the corrugation shape;
– the presence of an hydrodynamical developing section has not significant effects on the convective heat transfer.

The results seem confirm that, in the laminar flow with swirl components, the local Nusselt number depends on the corrugation pitch raised to the −0.5 power, although showing a different behaviour for low and high pitch values. However, the moderate number of pitch values tested does not allow to draw a definitive conclusion regarding to this phenomenon. Eventually, the comparison between the tubes having the helical and the axial symmetrical corrugation pointed out a stronger ability of the transverse corrugation in enhancing the rate of heat transfer, by promoting a transition to the turbulent flow. Though, the spiral corrugation may be preferable in comparison with the transverse one, since it can be easily fabricated on a large scale. Besides, the smooth profile shown by this type of wall corrugation is expected to prevent the formation of deposits close to the wall, by making these tubes very attractive in the food processing industry.

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