EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER AND PRESSURE DROP AUGMENTATION FOR LAMINAR FLOW IN SPIRALLY ENHANCED TUBES

Sara Rainieri, Angelo Farina and Giorgio Pagliarini
Università di Parma , Dipartimento di Ingegneria Industriale
V.le delle Scienze, 43100 Parma

ABSTRACT

The effect of the internal helical ridging on the heat transfer coefficient and friction factor is experimentally investigated for laminar internal flow of a Newtonian fluid in the case of uniform wall heat flux. Five different geometries are considered. The results are compared with the results of previous investigations and with the predictions for the smooth tube in order to point out the heat transfer enhancement and the friction factor increase in the laminar flow field.

1. INTRODUCTION

The introduction of roughened surfaces obtained with internal helical ridging or transverse ribbing, has revealed a successful method in enhancing heat transfer in tubes and, therefore, in reducing the heat exchanger size. Although most prior studies have focused on the turbulent flow, the laminar range is of particular interest in a wide variety of engineering situations, also including heat exchangers for viscous liquids in chemical process and food industry. When the flow is laminar because of small dimensions, low flow rates, or highly viscous fluids, like the ones processed in food industry, the use of smooth tubes gives poor performances and new configurations are needed to enhance the rate of heat transfer and, therefore, to reduce heat exchanger size and cost. The helical corrugation has revealed a type of roughness very suitable for dairy products, since fouling may be caused by sharper ridges.

Tubes having a transverse repeated-rib roughness with rectangular cross section have been analysed first by Webb et al. [1]. By using the law of the “wall similarity” and heat-momentum analogy he proposed some correlations where the friction factor is expressed as a function of the dimensionless geometric parameters and the Reynolds number, while the Stanton number is described as a function of the same variables and the Prandtl number as well. Gee and Webb [2] extended this study to the effect of the rib helix angle. Withers [3,4] developed correlating equations, based on the heat-momentum transfer analogy, for the heat transfer and pressure drop in tubes having simple and multiple helical internal ridging. These correlations were derived by experimental data obtained with water in the Reynolds number range 10000-120000 and Prandtl number range 4-10. Heat transfer enhancement of up to 2.5 to 3 were reported. Heat transfer augmentation in a spirally fluted tube was investigated by Yampolski [5]. He showed that the swirl in the flow due to the helical flutes enhances turbulent exchange on both the inside and the outside of the tube without causing a considerable increase in the friction factor. Other experimental data on pressure drop and heat transfer coefficient for turbulent flow inside doubly-fluted tubes were obtained by Richards et al [6]. Twelve different geometries have been analysed and the outcome shows that only some of them yield an improved performance, expressed by the Authors in terms of heat exchanger volume reduction. Garimella et al. [7] expressed the performance of spirally enhanced tubes in terms of a single non dimensional geometric parameter, the severity. Again turbulent flow is considered and it is shown that for severity values between 0.001 and 0.01 the heat transfer augmentation is accompanied by relatively low friction factor increase, thus confirming the efficiency of this kind of geometries. Garimella and Christensen [8] have investigated the fluid flow in annuli formed by placing spirally fluted, indented and ribbed tubes inside a smooth outer tube. Detailed temperature profile measurements and flow visualisation tests were performed for the laminar, transitional and turbulent flow in order to better understand the development of the swirl in the bulk flow. It was found that the fluted inner tubes are the most efficient in promoting the secondary flow and hence in enhancing the convective heat transfer. The same Authors [9,10] suggested also correlating equations for the friction factor and the Nusselt number for the laminar, transitional and turbulent flow in spirally fluted annuli.

The laminar flow has also been considered by Rowley and Patankar [11]. They obtained numerical results for pressure drop and heat transfer in circular tubes with circumferential internal fins with the boundary condition of linearly varying wall temperature. Despite the increased area provided by the fins, they showed that the presence of this kind of roughness often decreases the heat transfer coefficient rather than augmenting it. The recirculation of the fluid induced by the fins enhances the heat transfer only for Prandtl number greater than 5. A numerical technique for predicting the friction factor and the heat transfer in spirally fluted tubes has been proposed by Srinivasan et al. [12]. According to this method the flute region is modelled as a porous substrate, with direction dependent permeability, that affects the flow field in much the same manner as the spiral flutes. The Nusselt number values obtained with the numerical simulation are in good agreement with their experimental data for Reynolds number below 500.

Preliminary tests [13] confirmed that the presence of the helical ridging yields a significant heat transfer augmentation also in the laminar field and suggested the opportunity of further investigation.

2. TUBE GEOMETRY

All the tubes tested fall into the general category usually referred to as spirally enhanced tubes. They are characterised by an internal helical ridging corresponding to an external helical grooving, obtained by embossing a smooth tube made of stainless steel. Three of the five geometries tested in the present study show a single helix ridging and they have been obtained from a tube having an external diameter of 16 mm and a wall thickness of 1 mm. The others have two helical ridging that spiral along the tube in opposite direction and they have been obtained from a tube having an external diameter of 18 mm and a wall thickness of 1 mm. The tubes are shown if fig.1. Tubes 3 and 4 show a very irregular wall profile; in particular the two helix do not cross along the same generatrix of the cylindrical envelope surface. This is due to the particular manufacturing technique used which provide a double process to create consecutively the two helical corrugations. The geometric parameters usually used to describe spirally enhanced tubes are: the bore diameter D_b , the envelope diameter D_{env}, the ridge depth e, the pitch p and the number of starts N, (see for instance [7]) . A relevant non dimensional parameter introduced by Withers and Habdas [14] for this kind of geometries is the severity, defined as follows: $s=e^2/p\cdot D_n$, where D_n is the nominal tube diameter. The generic section of a cross-helix tube is shown in fig.2.

The local heat transfer coefficient, and therefore the local Nusselt number, have been obtained by considering a heat transfer surface equal to the surface of a cylinder having a diameter D_{env}. D_{env} has also been used as the characteristic length of the problem, in accordance with other investigators [3,6]. Richards et al. [6] used both the bore diameter and the envelope diameter to reduce their experimental data, and they drew opposite conclusions according to the characteristic length chosen. Srinivasan et al. [12] used instead the volume based diameter as the characteristic length for both the experimental and the numerical analysis. The envelope diameter can in practice be easily determined, while on the contrary the other dimensions, like the bore diameter, the nominal diameter D_n, or the volume based diameter result more difficult to be measured, as already pointed out by Richards et al. [6].

Table 1 shows the geometric data for the tubes discussed in the present study.

The geometric variables mentioned above are often not exhaustive to describe the various geometries since different manufacturing techniques are currently in use to produce them. For example tube 2 and 5 show the same values of ridge depth, pitch and diameter, and hence of severity, but the profile of the corrugation is different (see fig.1). The mechanical external pressure used to obtain the helical ridging has been applied in different ways, thus producing a sharper ridge in tube 5 than in tube 2.

Table 1- Geometric data for tubes tested

Tube Number	Bore Diameter D_b (mm)	Envelope Diameter D_{env} (mm)	Groove Depth e (mm)	Helix pitch p (mm)	Type of corrugation	Severity s
1	12	14	1	13	single helix	0.006
2	11	14	1.5	13	single helix	0.014
3	14	16	1	18	cross-helix	0.004
4	14	16	1	10	cross-helix	0.007
5	11	14	1.5	13	single helix	0.014

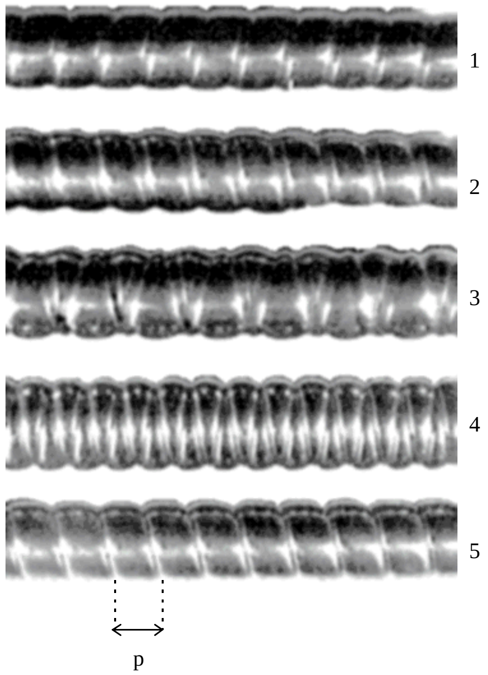
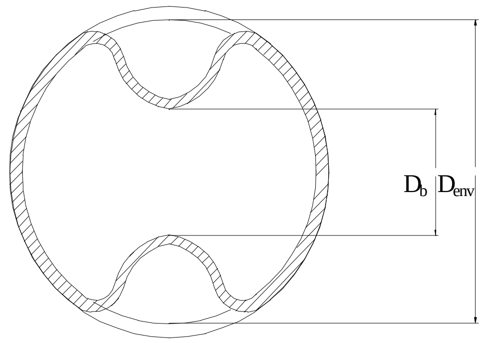


Figure 1- Tubes tested



$$e = \frac{D_{env} - D_b}{2}$$

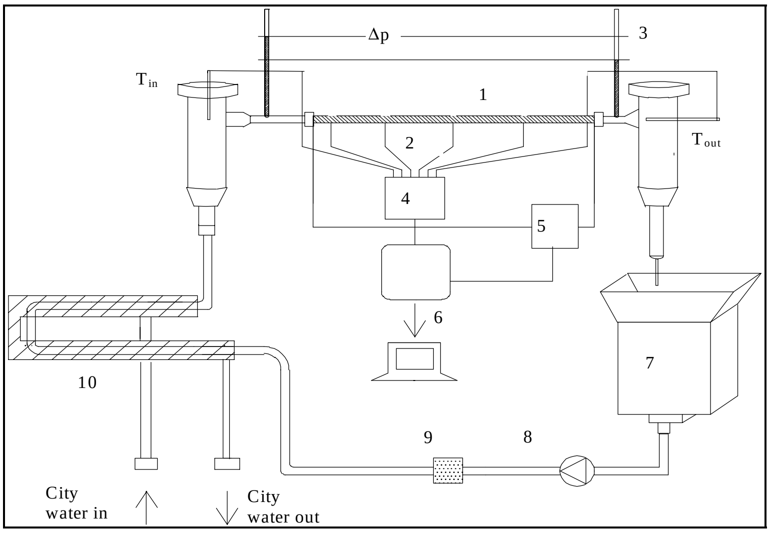
$$D_n = \frac{D_{env} + D_b}{2}$$

Figure 2- Geometry of the cross-corrugated tube

3. EXPERIMENTAL APPARATUS

The test section together with a schematic drawing of the flow loop are shown in fig.3.

The prescribed condition of uniform heat flux has been obtained by Joulean dissipation in the tube wall. The heated section is 1.84 m long and it is preceded by an unheated development approach section of about 1 m. In the case of the smooth tube this allows the fluid to reach the complete hydrodynamic development before entering the heated section for flow rates corresponding approximately to Reynolds numbers of up to 1500. The whole length of the heat transfer section is thermally insulated to minimise the heat exchange to the environment. The wall temperature has been measured trough 45 type T copper-constantan thermocouples, previously calibrated. They were attached circumferentially to the external surface of the tube, fixed with a layer of self bonding tape and electrically insulated from the tube itself. The inlet and outlet temperature has been measured by two thermocouple probes 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 trough the insulation, whose thermal resistance had been measured in a previous calibration of the apparatus. The data acquisition software provided to update the data coming from all the channels and to plot them as a function of time on a screen. The program provided besides to hold the heat flux constant by varying the output of the power supply according to the effective resistive load of the tube wall. The power supplied to the test section was selected for the various flow rates to limit the difference between the inlet and outlet bulk temperature to about 2.5 °C. In these conditions the effect of the variation of the fluid properties with temperature was assumed to be negligible. As working fluid a mixture containing approximately 50 per cent by weight of ethylene glycol in water has been used. The dynamic viscosity of this Newtonian fluid has been measured with a capillary-tube viscometer, type Ubbelohde. The Reynolds number range investigated is $200 < Re < 2000$. Flow rates were obtained by measuring the time needed to fill a volumetric flask placed at the outlet of the test section, while the magnetic flow-meter incorporated in the loop was simply used as a flow indicator. Pressure drop along the whole length of the tube was obtained by measuring the level reached by the fluid in two piezometric tubes in isothermal condition. Further details about the experimental apparatus and the data acquisition system can be found in [13].



1- Test section
2- Thermocouples
3- Piezometric tubes
4- Ice point reference
5- Power supply
6- Multimeter and personal computer
7- Holding tank
8- Volumetric pump
9- Flow-meter
10- Heat exchanger

Figure 3- Experimental apparatus

4. RESULTS

Heat transfer

Approximately 35 thermocouples probes have been distributed along the whole length of the heated section externally on the upper and lower crest of the corrugation. In order to measure also the local wall temperature variations, at some axial positions a thermocouple has been attached to the crest of the corrugation and other two probes have been placed in the preceding and in the following trough within a short distance. The internal wall temperature has been obtained by assuming a uniform heat generation in the tube wall and then calculating the temperature drop through the wall thickness. The temperature measured by the thermocouples placed in the external trough results lower than the one measured by the thermocouple placed in the adjacent external crest. In this region the local heat transfer coefficient results lower, probably due to some fluid stagnation. Figure 4 shows the mean temperature difference between the crest and the two adjacent troughs averaged along the heated section. Especially for tube 3, having a cross helical ridging and a long pitch, the local temperature variation becomes relevant in comparison to the bulk temperature difference between inlet and outlet (maximum value of about 2.5 °C). The fluid properties have been evaluated at the average bulk temperature, the arithmetic mean of the inlet and outlet fluid temperatures. Figures 5 and 6 show the local Nusselt number for tubes 1 and tube 3 versus the dimensional axial distance. The data show that for the lowest Reynolds numbers considered in both the tubes the local heat transfer coefficient approaches the value typical of a smooth tube; increasing the Reynolds number they show instead a considerable heat transfer enhancement. Tubes 2, 4, and 5 show a similar behaviour. Besides it can be observed that for high Reynolds number the local Nusselt number is almost constant along the heated section. This suggest that already at short axial distance from the point where the heating sets up the flow reaches a fully developed condition. The heat transfer augmentation in corrugated tubes is due to the distortion of the velocity and temperature fields. The presence of the repeated roughness elements causes in fact a disruption of the boundary layer and induces a secondary swirl flow, which both enhance the convective heat transfer. The Nusselt number enhancement shown by the results is in part also due to the fact that the heat transfer coefficient is based on the nominal area of a corresponding smooth tube having a diameter D_{env}, rather than the actual surface area of the corrugated tube. In this way higher heat flow rate at the wall, and therefore higher Nusselt numbers are predicted. The augmentation reported for the Nusselt number reflects then these combined effects. Figure 7 show the Nusselt number for the tubes tested averaged over the same dimensionless length versus the Reynolds number. Increasing the Reynolds number the data depart from the smooth tube prediction and approach a behaviour typical of the turbulent heat transfer. The transition occurs at different Reynolds number for the five tubes: for tubes 1, 2 and 5 having a single helical corrugation the transition value is between 700 and 800, while for tubes 3 and 4, having a cross-helical corrugation it is between 500 and 600. These data confirm what already pointed out by Garimella and Christensen [8,9] for annuli with spirally fluted and indented inner tubes for which they found a Reynolds transition value varying respectively in the range 310 < Re < 730 and 525 < Re < 1675. The high Reynolds number behaviour has been compared with the correlation suggested by Richards et al. [6] which is of the form:

$$Nu = C_i \cdot Re^{4/5} \cdot Pr^{1/3} \quad (1)$$

where the constant C_i depends on the wall geometry. They tested doubly fluted tubes, different from the ones tested in the present research, at Reynolds number greater than 10000 with constant wall temperature boundary condition. However, as shown in fig.7, eq.1 with the two extreme value of the constant C_i reported by Richards et al. [6] includes the experimental data obtained in the present investigation already for Reynolds number greater than about 1200. The same working fluid was used in all the runs and therefore the Prandtl number has been held constant for the various data set reported in fig.7. Figure 8 shows the Nusselt number averaged along the heated section for all the tubes tested divided by the corresponding value for the smooth tube. We will refer to this quantity as the heat transfer enhancement.

$$\eta_H = \int_0^L Nu_{cx} dx \,/\, \int_0^L Nu_{sx} dx \quad (2)$$

The comparison between the heat transfer enhancement of the tubes tested shows that tube 1 gives the poorest results, while tube 3 and 4, having a cross-corrugated surface, show the highest heat transfer enhancement. Garimella et al. [7] correlated heat transfer enhancement in spirally corrugated tubes by using a non-dimensional parameter, the severity. This non-dimensional parameter does not seem to be particularly significant for the data collected in the present study; in particular it does not well correlate the laminar heat transfer enhancement for the cross-corrugated tubes. In fact tube 1, having a single helix, and tube 4, having a double helix, have a similar value of severity, but they show a quite different value of the heat transfer enhancement.

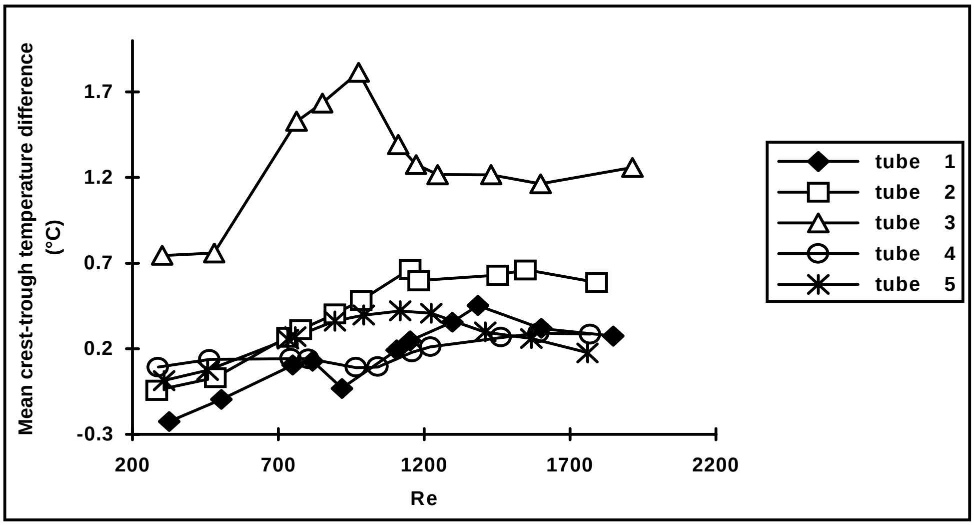


Figure 4- Mean crest-trough temperature difference averaged along the heated section for the tubes tested

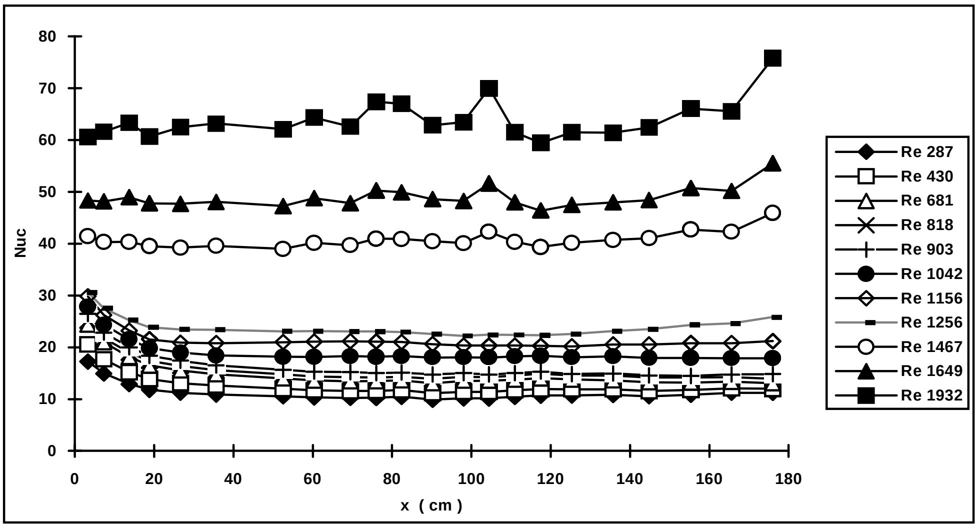


Figure 5- Local Nusselt number for tube 1 at various Reynolds numbers

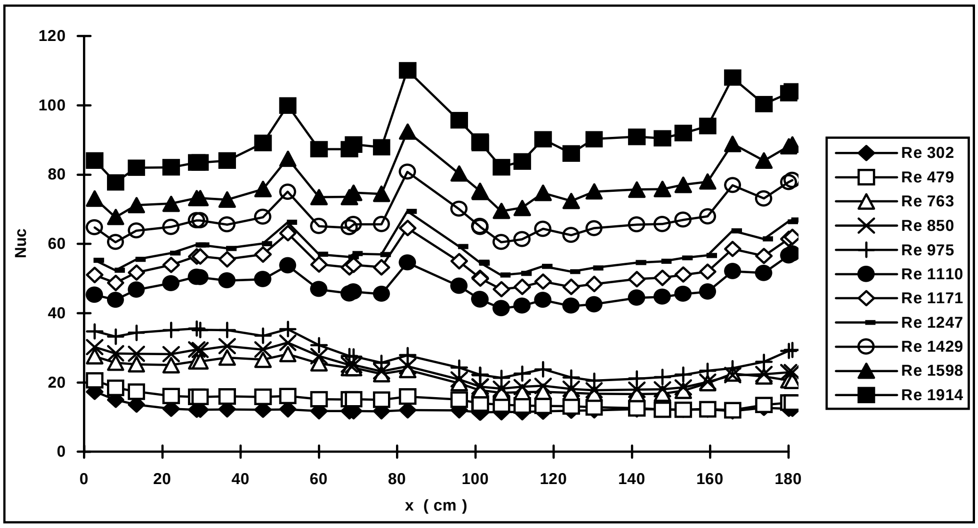


Figure 6- Local Nusselt number for tube 3 at various Reynolds numbers

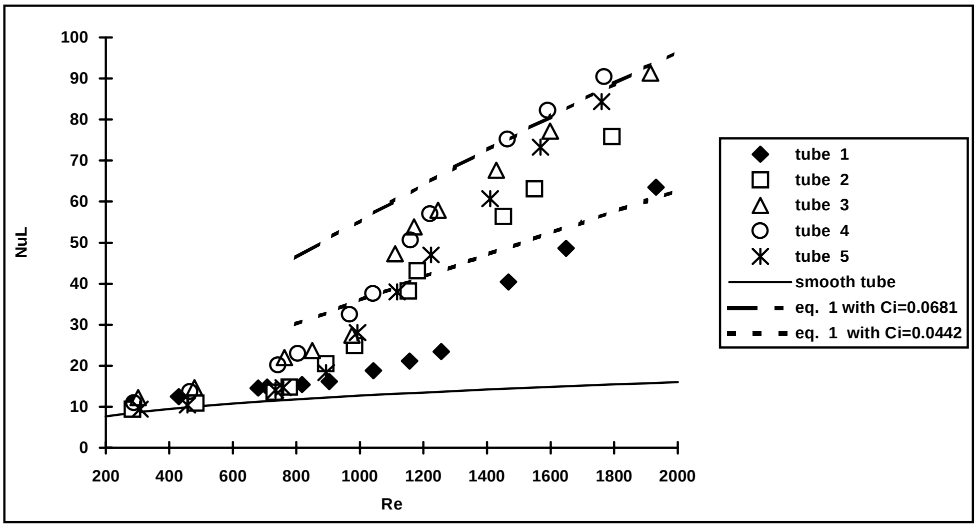


Figure 7- Mean Nusselt number for the tubes tested versus Reynolds number

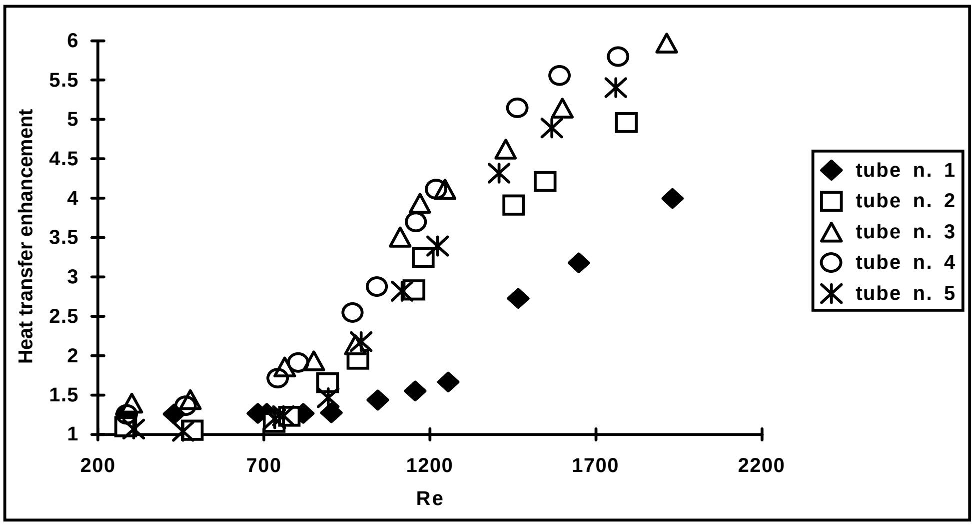


Figure 8- Heat transfer enhancement

Pressure drop

Pressure drops along the whole length of the tube were obtained by measuring the level reached by the fluid in two piezometric tubes. Richards et al. [6] suggested a correlation for the turbulent friction factor in doubly fluted tubes which is of the form:

$$f = B \cdot Re^{-n} \qquad (3)$$

The friction factor for the tubes having a single and a cross helical ridging together with both the theoretical and the experimental curves corresponding to the smooth tube are shown in fig.9. In the same figure also eq.3, corresponding to the same Richards et al. [6] tubes considered in fig.7 is shown. The experimental data for the friction factor are by far below eq.3 even in the Reynolds number range where the flow should approach a turbulent behaviour, as suggested by the heat transfer results. The friction factor increase $\eta_f = f_c / f_s$ is shown in fig.10 for the various Reynolds number considered. The highest values of η_f correspond to tubes 3 and 4 which both have a sharp corrugated wall, (se fig.1). The experimental values seem in qualitative good agreement with the data obtained for the spirally fluted annuli by Garimella and Christensen [9] who reported friction factor increases between 1.1 and 2.0 for the laminar flow and up to 10 for the turbulent flow.

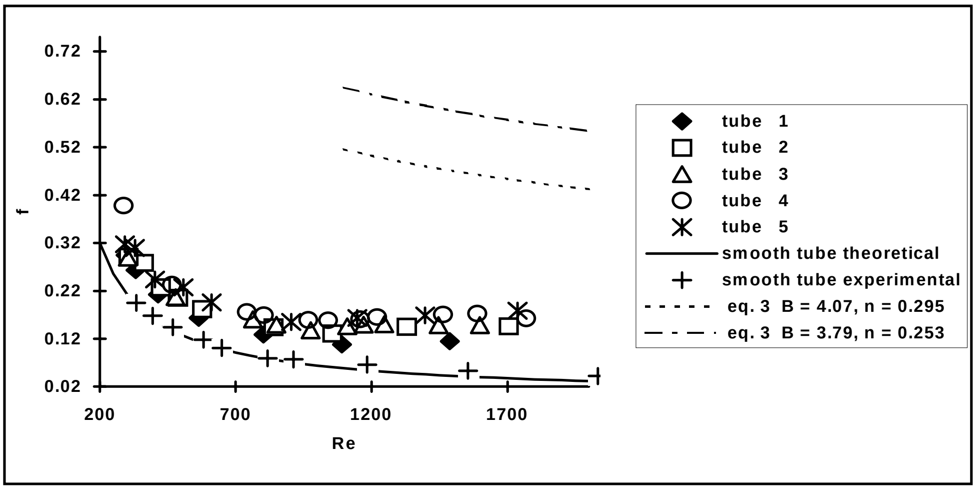


Figure 9- Friction factor versus Reynolds number

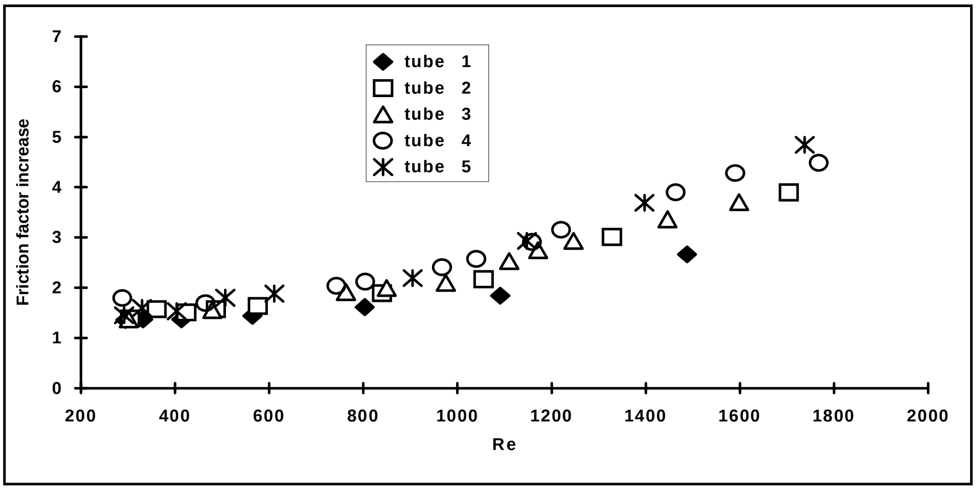


Figure 10- Friction factor increase versus Reynolds number

5. CONCLUSIONS

Laminar flow forced convection to a Newtonian fluid in spirally enhanced tubes has been experimentally investigated. The experimental data show that in the spirally enhanced geometries the transition to the turbulent flow can occur at Reynolds number values much lower than 2000. This is in good agreement also with the results obtained for annuli formed with spirally fluted inner tubes [7,8,9]. This early transition is accompanied by a significant heat transfer enhancement which assumes values between 1.1 and 6 in the Reynolds number range 300-1800. Despite the limited number of geometry tested, it can be concluded that the cross-corrugated tubes exhibit the highest heat transfer increase over the smooth tube values.

6. ACKNOWLEDGEMENTS

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7. NOMENCLATURE

D_{env}	envelope diameter	Pr	Prandtl number
D_b	bore diameter	Re	Reynolds number
D_n	nominal tube diameter = $(D_{env}+D_b)/2$	w	mean flow velocity
e	corrugation depth = $(D_{env}-D_b)/2$	x	axial distance
f	mean friction factor = $(2\cdot\Delta p\cdot D_{env}/L\cdot\rho\cdot w^2)$	Δp	pressure drop
L	length of the heated section	η_H	heat transfer enhancement
Nu	local Nusselt number	η_f	friction factor increase
Nu_L	mean Nusselt number	ρ	fluid density
p	helix pitch	**Subscripts**	
s	severity = $(e^2/p\cdot D_n)$	c	corrugated tube
		s	smooth tube

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