Passive techniques for the enhancement of convective heat transfer in single phase duct flow

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Passive techniques for the enhancement of convective heat transfer in single phase duct flow

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Abstract. This review presents the main results of the experimental campaign on passive techniques for the enhancement of forced convective single phase heat transfer in ducts, performed in the last years at the Laboratory of the Industrial Engineering Department of the University of Parma by the Applied Physics research group. The research was mainly focused on two passive techniques, widely adopted for the thermal processing of medium and high viscosity fluids, based on wall corrugation and on wall curvature. The innovative compound heat transfer enhancement technique that couples together the effect of wall curvature and of wall corrugation has been investigated as well. The research has been mainly focused on understanding the causal relationship between the heat transfer surface modification and the convection enhancement phenomenon, by accounting the effect of the fluid Prandtl number. The pressure loss penalties were also evaluated. The principal results are presented and discussed.

1. Introduction

The enhancement of convective heat transfer is an everlasting issue within applied research, since it can ultimately provide an effective way to optimize industrial thermal processes by reducing, at the same time, the size of the equipment and then the energy and materials costs [1].

This issue becomes particularly important in industrial applications in which the thermal processing of medium and high viscosity 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 efficiency of the heat transfer apparatuses in which the fluids are conveyed is inevitably penalized.

In these industrial fields, engineers have then been striving for techniques producing enhanced heat transfer coefficients, accompanied by reduced pumping power requirements. This is witnessed also by the huge amount of papers available within the world scientific literature of the last decades regarding heat transfer augmentation and by the growing numbers of registered patents related to heat transfer enhancement technology or devices (in 2001 Bergles estimated a steady-state rate of literature generation of about 400 papers and reports per year [2]).

According to the classification suggested by Bergles (see for instance [3]), the techniques for the enhancement of forced convection encompass the optimal use of active devices, propelled by mechanical aids or electrostatic fields (see for instance [4]), and passive devices, mostly based on some suitable conformation of the interface surface (see for instance [2,5]), or else a combination of both active and passive techniques [1].

Passive techniques are more attractive because no power is required to facilitate the enhancement.

Among them, treated surfaces, rough surfaces, displaced enhancement devices, swirl-flow devices, surface-tension devices, coiled tubes, or flow additives [1–2] are found. Moreover the simultaneous use of two or more of these techniques has been investigated in literature in order to verify if it is able to produce an enhancement larger than that produced by only one technique. This simultaneous use is termed compound enhancement [1]. Regarding compound techniques, it must be noted that they offer a promising tool for further enhancing the performances of thermal apparatuses, since the heat transfer coefficients are expected to be increased above any of the several techniques acting alone [6].

Another possible classification of heat transfer enhancement techniques into different generations and based on the stage of development of the heat transfer technology was further suggested by Bergles [2]: according to this approach, the first generation heat transfer technology is represented by the plain surfaces, while the fourth is represented by compound techniques.

The effectiveness of all of the above referred passive techniques is often well assessed in literature and performance evaluation criteria under both first and second principle approach have also been developed in order to compare the thermal performance benefits achievable through their use with the inevitably correlated pressure drop penalties (see for instance [7,8]).

By focusing on passive heat transfer enhancement techniques, it is important to remind that the main mechanisms on which they rely are: periodic disruption of the development of the boundary layers, generation of swirling/rotating and/or secondary flows, promotion of boundary-layer separation and attachment/reattachment phenomena, promotion of an early departure from the laminar flow and increase of the turbulent intensity.

In the experimental investigations described in the following paragraphs, two passive techniques widely adopted for the thermal processing of medium and high viscosity fluids and based on wall corrugation and/or wall curvature will in particular be analyzed, by reviewing the main results of the experimental campaign performed in the last years by the Applied Physics research group at the Laboratory of the Industrial Engineering Department of the University of Parma.

2. The enhanced tubes tested, experimental conditions and data processing

The investigation was focused on straight corrugated wall tubes and on helically coiled tubes having both smooth and corrugated wall.

2.1 Wall corrugated tubes

The introduction of roughened wall surfaces has revealed a successful method for enhancing heat transfer in tubes and, therefore, for reducing the heat exchangers' size for a given heat duty requirement. Among this kind of techniques, the most successful was found in single and multistart spirally wall corrugated tubes [5,9]. They are characterized by an internal helical ridging corresponding to and external helical grooving, obtained by embossing a smooth tube, generally made of stainless steel.

The enhancement effect associated to the wall corrugation is generally ascribed to the periodic interruption of the boundary layers development, to the increase in heat transfer area, to the generation of swirling and/or secondary flows and, above all, to the promotion of the transition to the turbulent flow regime [1,5].

The enhanced sections tested in the experimental investigations, were generated by stainless steel type AISI 304 tubes having a wall thickness of 1 mm. Both spirally (with single and cross helical profile) and transversely wall corrugated tubes were considered. Some representative enhanced tubes are shown in figure 1.

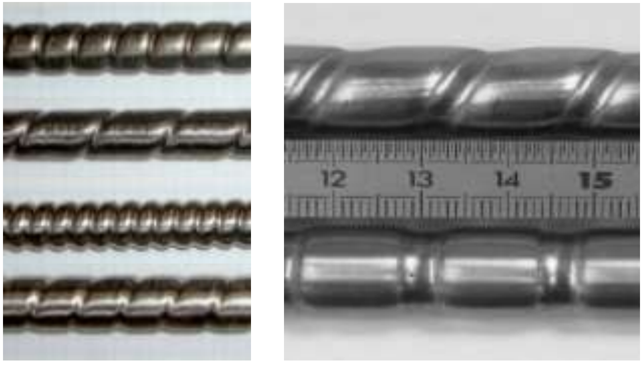

Figure 1. Representative spirally and transversely corrugated pipes.

2.2 Helically coiled tubes
In the experimental campaign hereby reviewed, both smooth and corrugated wall helically coiled tubes made of stainless steel type AISI 304, having a wall thickness of 1 mm were tested. They were characterized by coils following a helical profile along the tube's axis with different coil radius. The corresponding dimensionless curvature δ (the ratio of the tube diameter to the helix diameter) ranged from 0.031 to 0.056.

Two helically coiled pipes (one having smooth wall and the other having corrugated wall), representative of the ones tested in the experimental campaign, are shown in figure 2.

It is well known in literature that the enhancement of heat and mass transfer in coiled pipes with respect to the straight tube behaviour, is due to the distortion of the velocity and temperature profiles induced by the centrifugal force that acts with different intensity over the tube's cross section. The centrifugal force produces the onset of secondary flow structures, appearing as twin counter rotating vortices in the cross-sectional plane [10], that are responsible for a further enhancement of the forced convection mechanism. By neglecting all other effects related to geometry, except the centrifugal acceleration term, Dean derived a simplified momentum balance equation (known as ''Dean approximation'') in which the only dimensionless independent parameter is the Dean number, that couples together inertial with centrifugal effects [11], defined as $Dn = \delta^{1/2}Re$, where δ is the dimensionless curvature and *Re* is the Reynolds number.

Figure 2. Representative smooth and corrugated helically coiled pipes.

2.3 Experimental conditions and data processing
The enhanced sections were tested in heating condition obtained by Joulean dissipation directly within the tube wall. The experimental setup allowed investigating the heat transfer performance of the tubes under the thermal entrance problem.

As working fluids, both glycerol and ethylene glycol, that both show a Newtonian flow behaviour, and mixtures of these two fluids and water were used. Some tests were also performed by adopting real fluid foods, such as orange concentrate, that behaves has as non-Newtonian pseudoplastic fluid [9].

The directly measured quantities were: the electric power dissipated within the tube's wall, the fluid mass flow rate, the bulk fluid temperature at the inlet and outlet sections, the outer wall temperature (both by traditional contact sensors and the infrared thermographic technique) and the pressure drop along the test section. Further details about the experimental apparatus and the data acquisition system can be found in [5,12-16].

The derived quantities were: the average wall heat flux (obtained by the value of the heat power dissipated by Joule effect into the wall) the internal wall temperature (obtained by solving the steady state heat conduction problem with heat generation in the tube wall), the bulk temperature at any location in the heat transfer section (obtained by the energy balance of the system).

The heat transfer performance of the enhanced sections was then quantified, for different corrugation profiles, by evaluating the Nusselt number (both the local value along the axial coordinate and the average value over the heated section) and the friction factor (average value along the test section) by varying Reynolds, Dean and Prandtl numbers. The experimental condition enabled to test the enhanced sections in the Reynolds, Dean and Prandtl number ranges 4÷2000, 10-300 and 30-15000 respectively.

It must be noted that only if the axial and peripheral heat conduction effects within the wall can be considered negligible, the Joulean heating condition well approximates the thermal boundary condition of uniform wall heat flux at the fluid wall interface. If these assumptions cannot be made, a conjugate heat transfer approach is instead needed [17]. Regarding the experimental conditions hereby reviewed, being wall thermal conductivity relatively low and being the wall thin, the axial heat conduction effects were expected to be negligible. Regarding the peripheral heat conduction effects within the wall instead, they were expected to be negligible in straight tubes, while they couldn't be overlooked in coiled pipes, due to the secondary flow pattern.

In particular, in helically coiled tubes, being the fluid temperature profile strongly asymmetrical with respect to the tube's axis, the peripheral heat conduction in the wall caused a heat flux redistribution and the convective heat transfer coefficient was then expected to vary significantly along the periphery of the tube's cross section. In particular it was expected that the convective heat transfer coefficient presented higher values at the outer bend side of the wall surface than at the inner bend side due to the secondary flow pattern. It must be reminded that this irregular distribution may be critical in some industrial applications, such as in those that involve a thermal process. For instance, in food pasteurisation, the irregular temperature field induced by the wall curvature could reduce the bacteria heat-killing or could locally overheat the product.

From the value of the heat power dissipated by Joule effect into the wall, the circumferentially averaged values of the wall heat flux was derived, while for estimating its distribution along the wall periphery a different approach was needed.

At some specific cross section in the downstream region of the heated section, the local convective heat transfer coefficient and the local Nusselt number as a function of the angular coordinate along the wall periphery were then derived by processing, under the Inverse Heat Conduction Problem (IHCP) approach, the external wall temperature distribution acquired by means of an infrared camera [14,15]. This procedure was based on the inverse solution of the energy equation within the solid domain in which the external wall temperature distribution was experimentally measured and in which the heat flux distribution at the internal wall-fluid interface was assumed to be unknown.

To cope with the difficulty embedded in IHCPs, many techniques have been suggested in literature, and the most well-known are: function specification methods [18,19], iterative methods [20,21], methods based on filtering proprieties [22,23] and regularisation techniques [24]. In the processing of the data acquired in the experimental campaign hereby reviewed, the Tickhonov regularization methods and the filtering methods were adopted.

Among the regularisation techniques, Tikhonov regularization method is perhaps the most common: it promotes the construction of stable approximate solutions to the original IHCP by solving a well-posed problem via the minimisation of an objective function [24]. The objective function is generally expressed by the sum of the squared difference between the measured and the estimated temperature discrete data and of a regularisation parameter times a term that expresses the smoothness of the unknown quantity. The success of this approach relies on a proper choice of the regularization parameter, and this didn't reveal to be an easy task. In the processing of the data acquired in the experimental campaign hereby reviewed, the Tickhonov regularization methods was implemented with the aid of the most common approaches for choosing the regularization parameter, i.e. the classical L-curve method and the fixed-point iteration technique [14,15].

Another approach to solve the above described IHCP and that revealed very promising is the filtering technique [25]. This approach avoids the formulation of complex algorithms, since the wanted information (heat transfer coefficient or heat sources distribution) can be derived by directly solving the heat conduction equation that uses as input data a de-noised temperature field obtained by filtering the raw infrared map.

In order to estimate the local convective heat transfer coefficient along the wall periphery of the coiled section, the filtering technique approach, based on the Gaussian filter, was successfully applied to the infrared temperature maps acquired on the external wall surface [25].

2.4 Enhancement efficiency

The performances of the enhanced geometries were also suitably described by the following dimensionless quantities: the average friction factor enhancement, the heat transfer enhancement and the overall enhancement efficiency defined as follows:

$$\varepsilon_f = \frac{f}{f_0} \tag{1}$$

$$\varepsilon_H = \frac{Nu}{Nu_0} \tag{2}$$

$$\eta = \frac{\varepsilon_H}{\varepsilon_f} \tag{3}$$

where *Nu* and *f* are the Nusselt number and Darcy friction factor averaged along the test section respectively and where the subscript 0 refers to the plain geometry, i.e. the straight smooth wall tube behaviour.

In the data reduction the average bulk temperature between the inlet and outlet sections was used for evaluating all fluid properties, unless otherwise specified. Moreover, the heat transfer area was taken equal to the envelope cylinder surface area.

For the highest viscosity fluid (glycerol) in order to take into account the effect of the viscosity variation with temperature, which was expected to activate modifications within the boundary layers which alter the heat transfer mechanism in comparison to the constant property problem, the approach suggested by Sieder and Tate was adopted [26].

3. Results

3.1 Corrugated Wall Tubes

In the investigation both Straight tubes having Helical Corrugation (SHC) and Straight tubes having Transverse Corrugation (STC) were tested. The results reported in [5,27,28] clearly showed that the effect of the wall corrugation is negligible for Reynolds number values lower than a critical value (of the order of 500). In these conditions, the helical corrugation induced significant swirl components to which, however, an as much significant heat transfer enhancement was not associated [5].

Therefore for the spirally corrugated tubes as the ones exemplified in figure 1 the main augmentation mechanism was mainly ascribed to the turbulence promotion. Moreover the data pointed out a stronger ability of the transverse corrugation and of the cross helical corrugation with respect to the single helical corrugation in promoting the onset of instabilities within the flow [5,27]. A significant effect on the onset of instabilities within the flow was also proved to be related to the effect of the variation of the fluid thermophysical properties, mainly viscosity, with temperature. Moreover for the helical corrugation, the critical local Reynolds number was proved to be proportional to the dimensionless corrugation pitch. Some representative results are reported in figure 3 and in figure 4. The Nusselt number values averaged over the heated length, reported in figure 3 for SHC tubes with corrugation depth equal to 1 mm, confirmed that by increasing the Reynolds number the data depart from the smooth wall tube solution and approach a behaviour that was reported for the turbulent heat transfer in spirally fluted tubes [27]. The transition occurred at different Reynolds numbers according to the corrugation profile and, in any case, well below the critical Reynolds number value expected for smooth wall tubes. The experimental results for the Darcy friction factor, shown in figure 4 were instead by far below the turbulent flow data predicted for the spirally fluted tubes, even in the Reynolds number range where the flow should approach this flow regime, as suggested by the heat transfer results.

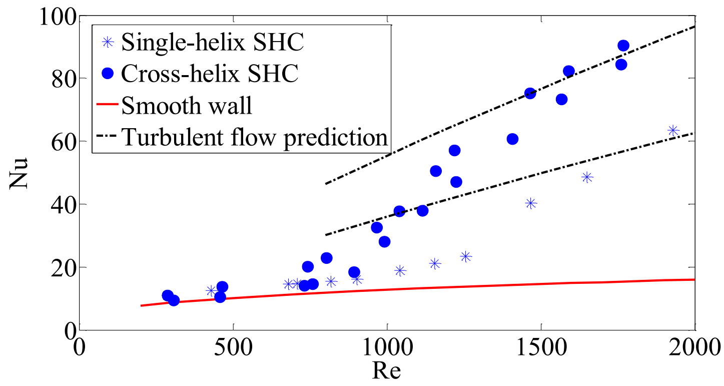

Figure 3. Averaged Nusselt number versus Reynolds number for the corrugated wall tubes and comparison with the smooth wall solution and with the turbulent flow data predicted for spirally fluted tubes [27] ($Pr \cong 30$).

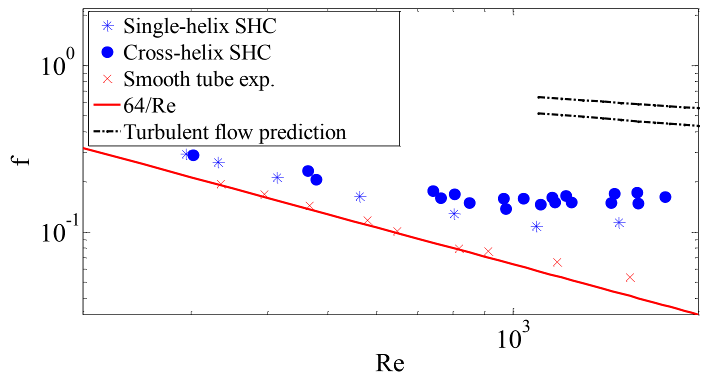

Figure 4. Averaged friction factor versus Reynolds number for the corrugated wall tubes and comparison with the smooth wall solution and with the turbulent flow data predicted for spirally fluted tubes [27].

3.2 Helically coiled Smooth Wall (HSW) and Helically coiled Corrugated Wall tubes (HCW)

Due to the secondary fluid pattern, the heat transfer enhancement registered for the HSW sections over the straight smooth wall behaviour, was significant, already for very low Reynolds number values [12]. In these conditions the HCW sections were instead less effective and the beneficial effect due to the pipe's curvature was even destroyed by the wall corrugation [12]. By increasing the Reynolds number to about 100 the effect of the wall curvature prevailed, while a further increase of the Reynolds number to values up to 1000 showed that in this flow regime the enhancement effect due to the wall corrugation superimposes to the one due to the curvature, by further augmenting the Nusselt number [5,27]. This behaviour was interpreted as a consequence of a departure from the steady laminar flow induced by the wall corrugation, as it was observed in wall corrugated straight tubes.

Some representative results are reported in figures 5 and 6. The data confirm that both wall curvature and wall corrugation are responsible for an additional convective transport mechanism that increases both the heat transfer and the pressure drop when compared to that in a straight tube. The same conclusion is pointed out by the synoptic plot of figure 7 in which the enhancement efficiency is reported vs. the Reynolds number for both the smooth and wall corrugated helically coiled tubes for a representative curvature ratio value. The data confirmed that, above the critical condition, the wall corrugation, as superimposed to the wall curvature, brings an additional heat transfer enhancement that is combined with a modest expense of the additional pressure-losses. In figure 7 the data regarding the SHC tubes behaviour obtained for a different Prandtl number fluid are reported, too. The results show that the simultaneous use of the two passive heat transfer enhancement techniques (wall curvature compounded with wall corrugation) is particularly effective if compared to the performance achievable through their separate use.

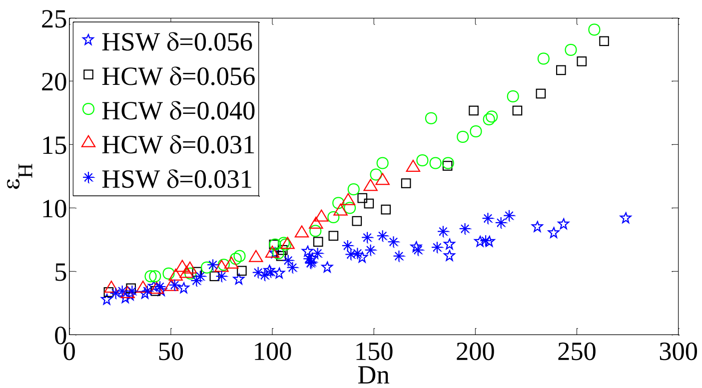

Figure 5. Heat transfer enhancement vs. Dean number for the helically coiled tubes (*Pr*≅200).

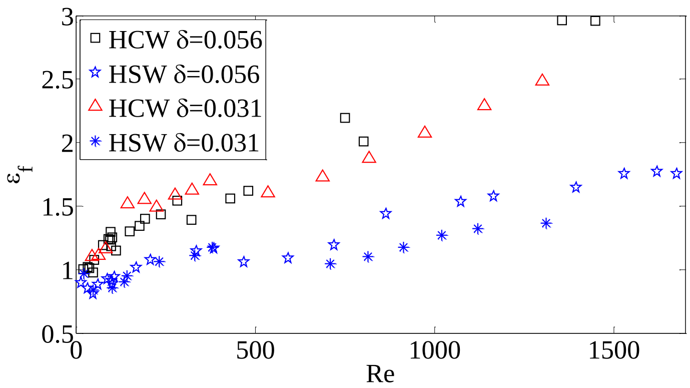

Figure 6. Friction factor enhancement vs. Reynolds number for the helically coiled tubes.

3.3 Convective heat transfer coefficient distribution

The data so far discussed referred to Nusselt number averaged along the wall circumference and these data are certainly important for the design of helically coiled pipes heat exchangers. Although this, the knowledge of the distribution of the convective heat transfer coefficient along the wall periphery is often a crucial and necessary information whose importance cannot be overlooked. For instance, in food pasteurization and sterilization a non-uniform heating could result in a failure to achieve a safe process. While for straight sections the heat transfer coefficient variations along the wall periphery are expected to be limited, in coiled tubes this phenomenon cannot be disregarded. In particular, the uneven distribution of the convective heat transfer coefficient along circumferential angular coordinate at the fluid-wall interface could negatively affect the residence time distribution thus impacting on the safety of the thermal process. The few data available in literature that face this problem mainly adopt the numerical approach and most of them consider low Prandtl number fluids [29].

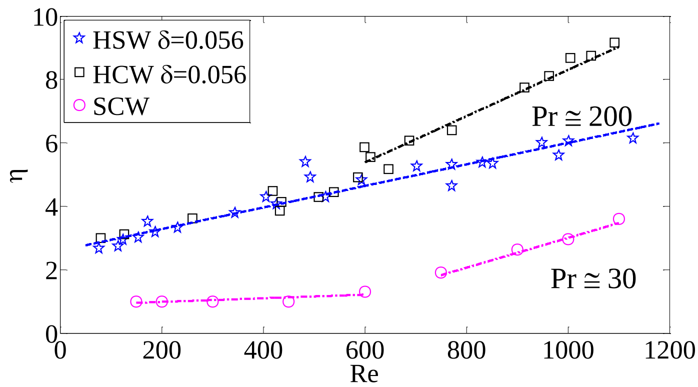

Figure 7. Enhancement efficiency vs. Reynolds number for the HSW and HCW tubes with δ=0.056 and comparison with SCW tube behaviour.

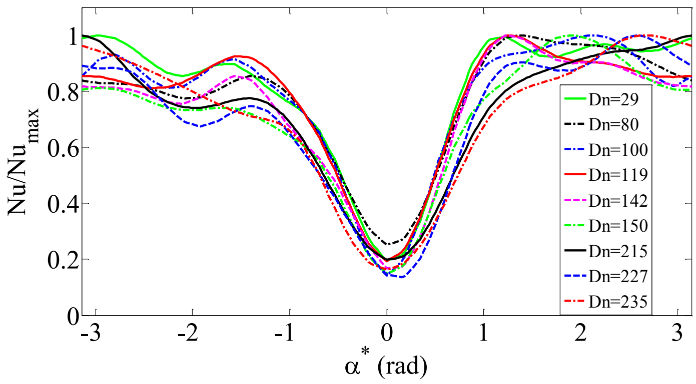

Figure 8. Normalized local Nusselt number for various Dean numbers along the wall perimeter (Pr≅200).

The research work performed in the last two years by the Applied Physics research group of the Industrial Engineering Department of the University of Parma tried to fill this literature gap, by optimizing an experimental procedure aimed to estimate the local convective heat transfer coefficient in coiled tubes. So far, the investigation was focused on the laminar flow regime in coiled tubes having smooth wall.

The temperature distribution maps on the external coil wall, acquired by means of the infrared thermographic technique at some representative pipe’s cross sections, were employed as input data of the linear IHCP in the wall under a solution approach based on both the Tikhonov regularization method [15] and the filtering technique [25].

Some representative results obtained with the Tickhonov regularization method with the support of the fixed-point iteration technique to determine the proper regularization parameter are reported in figure 8. The data confirm a strong uneven distribution of the convective heat transfer coefficient along the wall perimeter: at the outer bend side of the coil it is about five times that one at the inner bend side. Within the experimental uncertainty, the Nusselt number profile along the curvilinear coordinate, normalized with its maximum value, was almost independent on Dean number, by confirming the data of Jayakumar et al. [30] for turbulent heat transfer in helical pipes. Similar results were obtained by adopting the filtering approach that resulted also a powerful tool, alternative to the Tikhonov Regularization approach, to obtain the convective heat transfer coefficient distribution by starting from the same data, i.e. the infrared temperature maps acquired on the external tube wall [25].

4. Conclusions

The main results of the experimental campaign on passive techniques for the enhancement of forced convective heat transfer in ducts performed in the last years by the Applied Physics research group at the Laboratory of the Industrial Engineering Department of the University of Parma were presented.

Two passive techniques, widely adopted for the thermal processing of medium and high viscosity fluids and based on wall corrugation and/or wall curvature, were in particular discussed.

The main results can be summarized as follows:

−the solution based on the corrugations of the heat transfer surface, such as rough wall tubes, mainly relies on the promotion of the boundary layer transition to the turbulent flow regime. Therefore, this technique is very effective for medium viscosity fluids, i.e. fluids that are able to reach Reynolds numbers of the order of at least a few hundred while this solution is almost ineffective for highly viscous fluids.

-the solution based on modifications of the heat transfer surface that produce a swirl, or helical type flow appears very interesting instead also in the conditions in which the flow persists in the laminar regime, also in case of very low Reynolds numbers. In fact, although it is well known that the curvature of the wall stabilizes the flow by delaying the transition to the turbulent flow, the effect of the secondary flow due to the centrifugal force that acts on the fluid flowing along the curvilinear trajectory, has a positive effect on the convective heat transfer coefficient.

-the simultaneous use of the two passive heat transfer enhancement techniques (wall curvature compounded with wall corrugation) is particularly effective if compared to the performance achievable through their separate use for Dean numbers higher than about 120, while it produces a negative effect for highly viscous fluids which flow under very low Reynolds numbers (of the order of 10).

-the heat transfer augmentation due to curvature is accompanied by a side effect related to the unevenness of the convective heat transfer coefficient distribution along the wall perimeter: in the Dean and Prandtl number range hereby investigated, the convective heat transfer coefficient at the outer bend side of the coil is about five times that one at the inner bend side, and the normalized Nusselt number distribution shows a shape that is almost independent on Dean number.

The here discussed results highlight that, as expected, within passive heat transfer enhancement, the proper solution to obtain a significant benefit strongly depends on the fluid flow regime: below a critical condition a given augmentation technique could be ineffective or even counterproductive.

Further efforts are needed in order to establish a fruitful connection between the scientific research outcomes such as the ones here reviewed and the technological innovation process which the process industry nowadays requires.

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