Journal of Physics: Conference Series

PAPER • OPEN ACCESS

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To cite this article: F Bozzoli *et al* 2019 *J. Phys.: Conf. Ser.* **1224** 012046

IOP Conf. Series: Journal of Physics: Conf. Series **1224** (2019) 012046 doi:10.1088/1742-6596/1224/1/012046

Experimental investigation on the convective heat transfer enhancement in tubes with twisted-tape inserts

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Abstract. Passive heat transfer enhancement techniques are one of the most interesting approaches to increase the thermal efficiency of heat transfer apparatuses since they do not require external power. Between them, twisted-tape inserts are a very economical technique that can be easily installed in an already working plant in order to increase its thermal performance. The aim of the present work is to experimentally investigate the heat transfer augmentation mechanism induced by the insertion of twisted tapes. The heat transfer performance of the twisted-tape inserts is compared with that of coiled pipes, whose effect on heat transfer is based on the onset of swirl components into the flow as it happens for twisted tapes devices. The two heat transfer enhancement mechanisms have been deeply investigated by using different fluids in order to obtain a robust comparison both for the laminar and the turbulent flow regime.

1. Introduction

The heat transfer enhancement techniques constitute a very interesting research issue in the heat transfer field. In many industrial applications, such as those that involve food, chemical and pharmaceutical products, engineers have then been struggling for techniques generating enhanced heat transfer coefficients, accompanied by reduced pumping power requirements.

Today this research topic is attracting a renewed interest in the process industry due to the increase of energy and raw materials cost and 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 number of registered patents related to heat transfer enhancement technologies or devices. These techniques essentially reduce the overall thermal resistance by increasing the heat transfer coefficients with or without the increase of the heat transfer surface.

The benefits that can derive are, for instance, the reduction of the size of the heat exchanger that means costs reduction, the decrease of the temperature difference at which they work that means a reduction of the thermal stress for the product or the increase of the thermal power exchanged.

The techniques of increase of the convective heat transfer can be divided into active techniques, that require, for instance, mechanical aid or electrostatic fields, and passive techniques that do not require external power [1,2,3]. The passive techniques for the enhancement of convective heat transfer are based on changes induced on the fluid flow through a proper conformation of the surface, such as curvature of the walls or surfaces roughness or corrugation or through the insertion of devices in the main flow directly or by means of additives [1, 2].

In particular, the inserts are elements that are positioned in the flow passage with the aim of increasing the heat transfer rate. This enhancement technique results particularly attractive for the low cost, the rapid installation and the easy maintenance [4]. Within this category of passive heat transfer enhancement techniques, displaced devices, twisted tapes, wire coils are the most commonly adopted. Twisted tapes are metallic or non-metallic strips twisted with some suitable technique at desired shape and dimension and inserted in the flow with the aim of introducing swirl components into the flow and of disrupting the boundary layer at the pipe surface [5]. However, the pressure drop inside the tube will be increased by introducing the twisted-tape insert. For this reason, twisted-tape inserts have been the object of many research works that analysed the different configurations of these devices studying full-length and short-length twisted tape having a constant pitch or a variable one [6-8] in order to investigate the optimal design and achieve the best thermal performance with less friction loss.

Hong and Bergles [7] performed one of the first experimental work on twisted-tape inserts investigating both the pressure drops and the heat transfer enhancement in pipes by using water and ethylene glycol as working fluids. The insertion of twisted tapes produced an increase of the Nusselt number up to 9 times the empty tube value; in addition, the Authors proposed a correlation based on the Reynolds and Prandtl numbers.

Manglik and Bergles [8] proposed empirical correlations for the Nusselt number and friction factor, identifying a dimensionless swirl parameter. The Authors stressed the attention on this parameter, identified as a crucial factor to describe the behaviour of these devices and the interaction between viscous, inertia and centrifugal forces.

Sarma *et al*. [9] investigated the turbulent regime in pipes fitted with twisted-tape inserts; these Authors proposed a correlation for the Nusselt number as a function of the Reynolds and Prandtl numbers and the twisted tape pitch to tube diameter ratio.

Particularly interesting is the research provided by Ujhidy et *al*. [10] who investigated the laminar flow in coils and tubes containing twisted tapes and helical static elements. These Authors highlighted the similarity of the flow pattern characterised by swirl components present in coils and tubes with twisted tapes or helical static elements proposing also a modification of the Swirl number.

The present work is focused on the experimental investigation of the heat transfer augmentation mechanism induced by the insertion of twisted tapes both in the laminar and in the turbulent flow regimes. In particular, in connection with the work of [10], it has been investigated the role played by the swirl components introduced in the flow by also comparing the heat transfer performance of the twisted-tape inserts with that of coiled pipes.

2. Experimental setup

Three different stainless steel twisted tapes (figure 1) characterized by a thickness δ of 0.001 m and axial pitch p of 0.08 m, 0.15 m and 0.18 m, were considered in the present investigation.

The tape was fitted inside a straight stainless steel tube having wall thickness of 0.001 m and internal diameter d = 0.013 m, that is equal to the tape width.

The experimental setup, schematically shown in figure 2, enabled to investigate the twisted tape performances under the constant heat flux boundary condition at the fluid-wall interface.

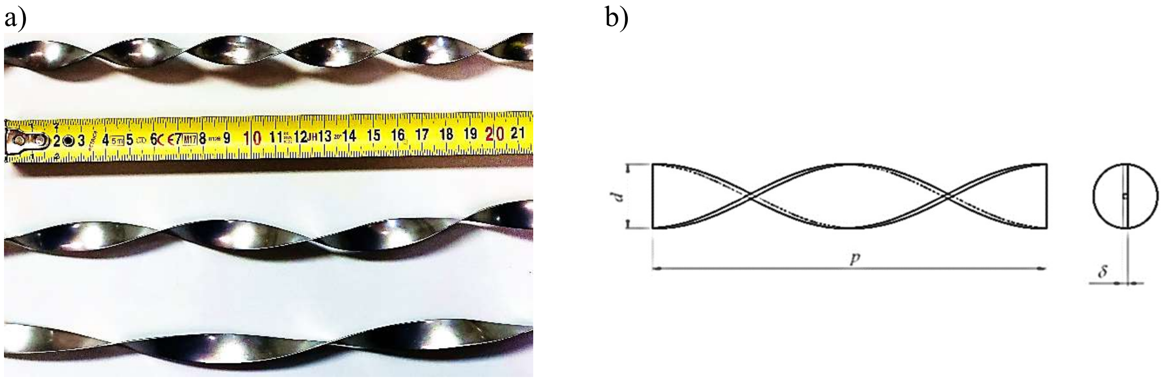


Figure 1. The twisted tapes tested (a) and a twisted tape sketch (b)

This condition was obtained by Joule dissipation in the tube wall. To this aim, the test section was fitted with steel fin electrodes which were connected to a power supply, type HP 6671A, working in the ranges 0–8 V and 0–220 A. The power supplied to the fluid was assumed uniform along the heat transfer section, while a secondary heat exchanger, fed with city water, provided a constant working fluid temperature at the inlet tube section. The outer wall of the tube was thermally insulated in order to minimize the heat exchange to the environment. The temperature of the wall and of the fluid at the inlet and outlet sections was measured by 40 type T thermocouples, previously calibrated and connected to a multichannel ice point reference, type KAYE K170-50C. The data acquisition system used to collect the measurements consists in a high precision multimeter, type HP 3458A, connected to a switch control unit, type HP 3488A, driven by a personal computer. Further details of the experimental setup are reported in [11] and, therefore, they are omitted here.

The tube wall temperature was measured by placing T-type thermocouples at different circumferential and axial locations on the external tube surface, along the heated section.

The fluid bulk temperature at the inlet and outlet sections was measured by using thermocouple sensors inside mixing chamber placed upstream and downstream the tested tube, respectively. It has to be pointed out that the mixing chamber were preceded by mixing inserts in order to reduce the fluid temperature stratification.

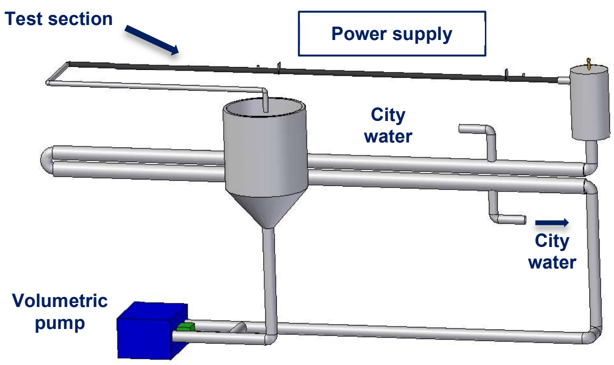


Figure 2. Sketch of the experimental setup.

The bulk temperature at any axial position was then estimated from the energy balance of the fluid, also accounting for the heat losses through the thermal insulation, whose thermal resistance was previously estimated during the calibration of the experimental apparatus. The mass flow rate was obtained by measuring the time needed to fill a volumetric flask placed at the outlet of the test rig. Both ethylene glycol and water were used as working fluid in the Reynolds number ranges 240-3500 and 1400-16100 for the two fluids, respectively. Moreover, the Prandtl number of the two working fluid, at the working mean temperature, was ranging from 40 to 55 for ethylene glycol and from 6 to 8 for water.

To assess the overall experimental procedure, the estimated and the measured fluid bulk temperature values at the outlet section were compared.

3. Results and discussion

The heat transfer performance was quantified by means of the Nusselt number, by adopting the internal tube diameter as the characteristic length in data reduction.

The local Nusselt number was computed as follows:

$$Nu_x = \frac{h_x \cdot D_{env}}{\lambda} \tag{1}$$

where h_x is the circumferentially averaged local convective heat transfer coefficient, based on the circumferentially averaged wall temperature and on the local fluid bulk temperature. In the data reduction the average bulk temperature between the inlet and outlet sections was, instead, used for evaluating all fluid properties.

The uncertainties related to the heat transfer measurements, are estimated by the classical error propagation theory [12]. According to the parameter uncertainties reported in table 1, the relative error on the measured Nusselt number is about 9%.

For the experimental configurations here considered, the fully developed conditions were always reached

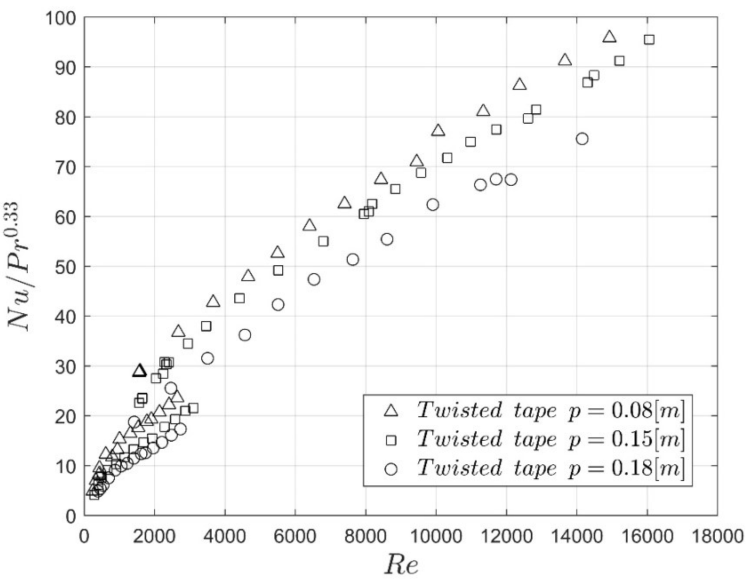


Figure 3. Asymptotic Nusselt number against the Reynolds number

Table 1. Parameter uncertainties

Parameter	$V\,[V]$	$I\,[A]$	$D[mm]$	$\Delta T\ [K]$	$\lambda\ \left[\frac{W}{m\,K}\right]$
Uncertainties	0.001	0.01	0.05	0.1	0.02

Therefore, the asymptotic Nusselt number was used in the discussion of the results to permit a more meaningful comparison with the coiled pipes thermal behaviour that will be discussed later. The tests were performed by varying the mass flow rate and, as a consequence, the Reynolds number.

For all the runs the asymptotic Nusselt number divided by Pr^{033} is reported in figure 3 as a function of the Reynolds number. Figure 3 points out a significant effect of the tape axial pitch. Moreover, for this particular geometry, it is also possible to identify two different flow regimes, that according to Manglik and Bergels [8], appear around $Re<2500$ and $Re>11500$ for the laminar and turbulent flow regimes respectively. Indeed, Manglik and Bergles [8] suggest that the swirl flow induced by the twisted-tape insert is characterised by the Swirl number, a dimensionless parameter which represents a balance between the centrifugal force due the helical flow, the convective inertia of the bulk axial flow and the viscous force:

$$Sw^2 \propto \frac{Centrifugal\ force \cdot Convective\ inertia\ forces}{Viscous\ forces^2} \tag{2}$$

Thus, the Swirl number can be defined as follows:

$$Sw = \sqrt{\frac{2\frac{\rho W^2}{p}\cdot\frac{\rho W^2}{d}}{\left(\frac{\mu W}{d^2}\right)^2}} = Re\sqrt{2\frac{d}{p}} \tag{3}$$

where μ and ρ are the dynamic viscosity and the density of the fluid, respectively.

The asymptotic Nusselt number divided by $Pr^{0.33}$ as a function of the above defined swirl parameter is then reported in figure 4, for both the laminar and the turbulent flow regimes.

In the same figure the correlation proposed by Hong and Bergles [7] for laminar flow regime:

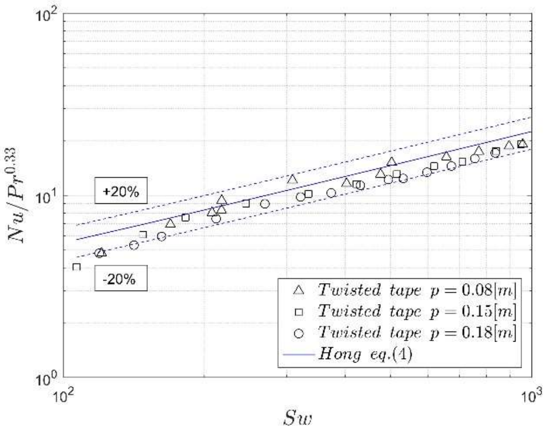
$$Nu = 5.172\left(1 + 0.005484\left(\frac{Re}{p/2d}\right)^{1.25} Pr^{0.7}\right)^{0.5} \tag{4}$$

and the correlation provided by Sarma *et al*. [9] for turbulent flow regime:

$$Nu = 0.023\,Re^{0.8}Pr^{0.4}\left(1 + \frac{2\pi}{p/d}\right)^{0.87} \tag{5}$$

are reported, too.

(a)



(b)

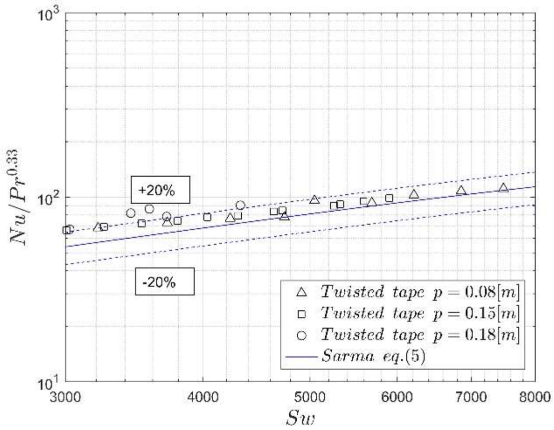


Figure 4. Asymptotic Nusselt number against the Swirl number
(a) laminar flow regime; (b) turbulent flow regime

The present experimental results confirm the prediction of the above correlations in all the considered *Sw* number range.

The convective heat transfer enhancement mechanism of the twisted-tape insert is based on the onset of a secondary swirl flow, which superimposes to the axial flow, thus reducing the thermal boundary layer thickness at the tube wall.

A similar enhancement mechanism is produced by the coiled tube [11], geometry which is schematically shown in figure 5.

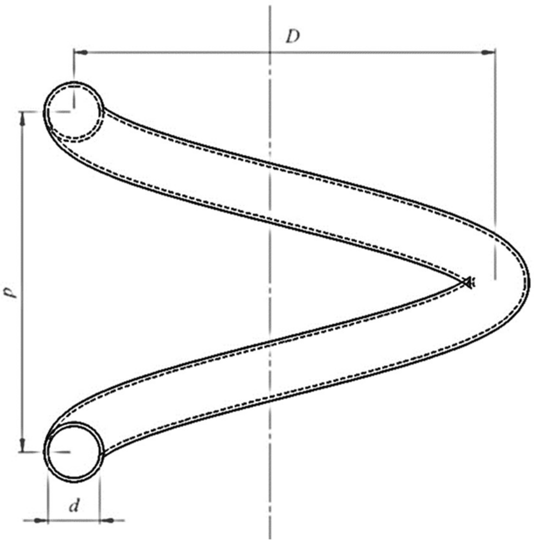


Figure 5. Sketch of the coiled pipe

For coiled tubes, Naphon and Wongwises [13] proposed the below reported correlations. For the laminar flow regime:

$$Nu = \left(2.153 + 0.31\left(Re\sqrt{\frac{d}{D}}\right)^{0.643}\right)Pr^{0.177} \tag{6}$$

which is valid for 20 < *De* < 2000, 0.7 < Pr < 175 and 0.0267 < *d/D* < 0.0884 and where *d* is the tube diameter, *D* is the coil diameter and *De* is the Dean number defined as:

$$De = Re\sqrt{\frac{d}{D}} \tag{7}$$

For the turbulent flow regime, the authors suggested the following correlation:

$$Nu = 0.328Re^{0.58}Pr^{0.4} \tag{8}$$

which is valid for 6000<*Re*<180000.

Following again the suggestion of Manglik and Bergles [8], that underlined the importance of the swirl factor, and the attempt of Ujhidy et *al*. [10] to define a dimensionless parameter that joins the swirl effects generated by different devices, it was computed an analogous Swirl number for the coiled tubes:

$$Sw = \sqrt{\frac{\frac{2\,\rho W^2}{D}\cdot\frac{\rho W^2}{d}}{\left(\frac{\mu W}{d^2}\right)^2}} = Re\sqrt{2\frac{d}{D}} \tag{9}$$

It differs slightly from the case of twisted-tape insert, since for the twisted-tape insert the centrifugal force is proportional to 1/*p*, while being proportional to 1/*D* for the coiled tube.

The correlations of Naphon and Wongwises [13] for the coiled tube are compared in figure 6 with the present experimental results obtained with the considered twisted tapes.

In the laminar flow regime correlation (6) underestimates the twisted tape performance; in the turbulent flow regime, instead, correlation (8) greatly overestimates it. This may be related to the different velocity distribution in the different flow regimes for the two considered devices.

However, the graphs point out that the Swirl number previously defined contains the information related to the swirl phenomena since both the twisted tape and the coiled tube performance has the same trend if they are graphed as a function of it. This suggest a modification of the multiplying coefficient that appear in the correlations (6) and (8) as follows:

$$Nu = 1.41\cdot\left(2.153 + 0.31\left(Re\sqrt{\frac{d}{D}}\right)^{0.643}\right)Pr^{0.177} \tag{10}$$

for the laminar flow regime, while for the turbulent flow regime:

$$Nu = 0.6\cdot 0.328Re^{0.58}Pr^{0.4} \tag{11}$$

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(a) (b)

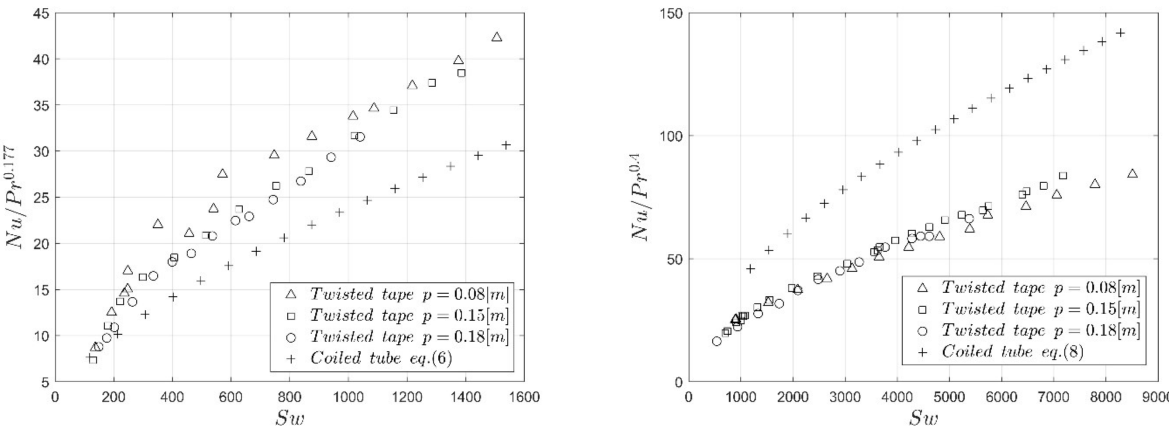

Figure 6: Nusselt number as a function of the Swirl number:
(a) laminar flow regime; (b) turbulent flow regime

The multiplicative coefficients introduced in the Naphon and Wongwises correlations were estimated by using the least square approach within the Matlab ® environment.

The modified Naphon and Wongwises correlations, expressed by equations (10) and (11), are graphed in figure 7, which confirm their ability to model the convective heat transfer mechanism of the twisted-tape insert. It has to be highlighted that the proposed correlations are valid as far as the geometrical similarity is respected.

(a) (b)

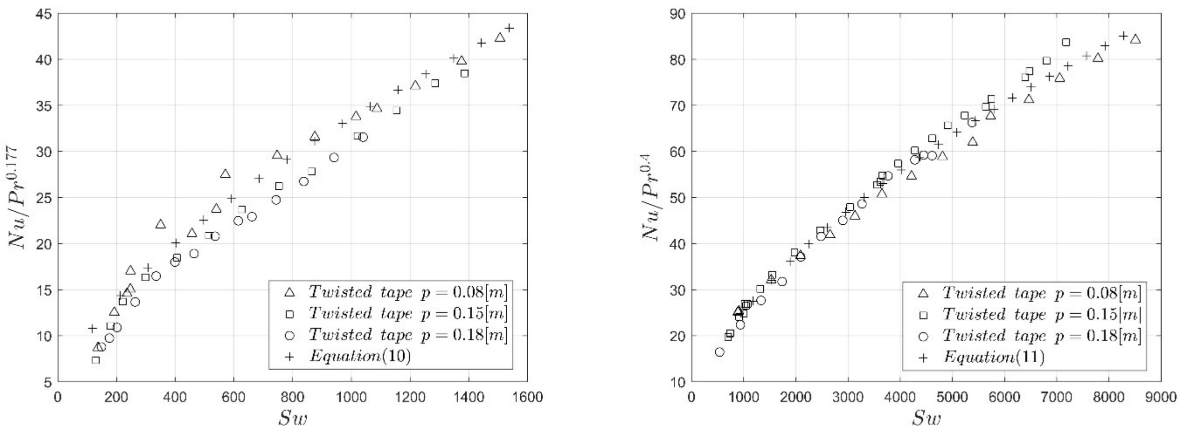

Figure 7: Nusselt number as a function of the Swirl number:
(a) laminar flow regime and (b) turbulent flow regime

4. Conclusion

In the present paper, the convective heat transfer performance of three different stainless steel twisted tapes, characterized by a thickness δ of 0.001 m, a width of 0.013 m and axial pitch *p* of 0.08 m, 0.15 m and 0.18 m, has been experimentally investigated. The experiments were conducted in both the laminar and the turbulent flow regimes, using ethylene glycol and water as working fluid in the Reynolds number ranges 240-3500 and 1400-16100, respectively.

The results were compared with the correlations already available in literature. Particular attention was given to the role played by the swirling flow components due to the twisted-tape inserts, by comparing the performance of this heat transfer enhancement device with that of coiled pipe, whose effect on heat transfer is based, as well, on the onset of swirl components into the flow. The results point out that a correlation obtained by modifying the correlation of Naphon and Wongwises for coiled tubes, can successfully model the convective heat transfer mechanism of the twisted-tape inserts.

Acknowledgments

This work was partially supported by the Emilia-Romagna Region (POR-FESR 2014-2020: “Applicazione delle micro/nanotecnologie per lo sviluppo di componenti innovativi per gli impianti di condizionamento " and “Progetto Nuovi paradigmi per la progettazione, costruzione ed il funzionamento di macchine e impianti per l'industria alimentare").

5. References

[1] Bergles A E 1998 *Handbook of heat transfer. Chapter 11: techniques to enhance heat transfer*, (New York McGraw-Hill)
[2] Webb R L 1994 *Principles of enhanced heat transfer* (New York John Wiley & Sons Inc)
[3] Rainieri S, Bozzoli F Cattani L and Vocale P 2014 Parameter estimation applied to the heat transfer characterisation of Scraped Surface Heat Exchangers for food applications *J Food Eng* **125** 147–156
[4] Tu W, Tang Y, Hu J, Wang Q and Lu L 2015 Heat transfer and friction characteristics of laminar flow through a circular tube with small pipe inserts *Int. J. Therm. Sci.* **96** 94-101
[5] Liu S and Sakr M 2013 A comprehensive review on passive heat transfer enhancements in pipe exchangers *Renew. Sust. Energ. Rev.* **19** 64-81
[6] Date A W and Singham J R 1972 Numerical prediction of friction and heat transfer characteristics of fully developed laminar flow in tubes containing twisted tapes *J. Heat Transfer* **94** 54
[7] Hong S W and Bergles A E 1976 Augmentation of laminar flow heat transfer in tubes by means of twisted-tape inserts *J. Heat Transfer* **98** 251-256
[8] Manglik R M and Bergles A E 1993 Heat transfer and pressure drop correlations for twisted tape inserts in isothermal tubes *J. Heat Transfer* **115** 890-896
[9] Sarma P K Subramanyam T, Kishorea PS, Rao D V and Kakac S 2002 A new method to predict convective heat transfer in a tube with twisted tape inserts for turbulent flow *Int. J. Therm. Sci.* **41** 955–960
[10] Ujhidy A, Nemeth J and Szepvolgyi J 2003 Fluid flow in tubes with helical elements *Chem. Engng and Processing* **42** 1–7
[11] Rainieri S, Bozzoli F and Pagliarini G 2012 Experimental investigation on the convective heat transfer in straight and coiled corrugated tubes for highly viscous fluids: Preliminary results *Int J Heat Mass Tran* **55** 498–504
[12] Kline S and McClintock F A 1953 Describing uncertainties in single-sample experiments *Mech Eng* **75** 3-8
[13] Naphon P and Wongwises S 2006 A review of flow and heat transfer characteristic in curved tubes *Renew. Sust. Energ. Rev.* **10** 463-490