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Local heat-transfer coefficient estimation in cross-helix corrugated tubes under turbulent regime

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Abstract. In a wide variety of engineering applications, convection heat transfer enhancement plays a major role in reducing the heat exchanger's size and costs and increasing their efficiency. The passive methods for increasing convective heat transfer are the most interesting since they don't require any external power to achieve the desired enhancement. For this reason, this approach is preferred in the industrial sector of heat exchangers. For the enhancement of the performances of these apparatuses, in particular, the tubular ones used in the pharmaceutical, food, and chemical, the most adopted passive solution is the corrugation of their tube walls. Two pipes characterized by cross-helix corrugation have been experimentally analyzed both investigating the global heat transfer performance and studying the local heat transfer coefficient distribution to profoundly examine the thermal performance enhancement mechanisms of this passive technique. It has been observed that at $Re = 16,000$, the heat transfer rate in the smaller pitch tube is 18.18% higher than that in the larger pitch tube. The inverse heat conduction problem in the tube wall is addressed to achieve the local coefficient distribution method by using temperature readings collected from an infrared camera on the outer surface of the tube as the initial data.

1. Introduction

Convective heat transfer enhancement is a challenge for engineers and researchers to minimize the cost, reduce the size, enhance the efficiency, and effectiveness of heat exchangers. In this respect, for improving the heat transfer performances of heat exchangers it is generally possible to identify two different categories of techniques: passive ones and active ones. If there is no external power required, the approach is identified as passive otherwise active [1]. Normally passive techniques are preferred since they do not require external power, they are cheaper and easier to implement in many devices. Several passive approaches are used to enhance the heat transfer, such as treated and rough surfaces [2], swirl-flow and surface-tension devices [3], coiled tubes [4], or flow additives [5]. In many industrial sectors, the mostly adopted technique is represented by the wall pipe corrugation and among the different corrugation profiles adopted the cross-helical one demonstrated promising thermal performance [6].

This technique showed to be cost-effective and easy to manufacture, making it appealing for food industry applications. It is possible to manufacture this type of profiles easily and continuously by creating two helical corrugations in opposite directions, while also meeting the standards of hygienic design. Wall corrugation on a pipe's inner surface has several effects that increase heat transfer. It interrupts the thermal boundary layer, it enhances the heat transfer surface area by increasing the contact area between the fluid and the pipe wall, and it promotes the early onset of the transition to turbulent flow [7,8]. Bozzoli et al. [9] conducted an experimental study to investigate the performance of cross-helix corrugated geometries in terms of heat transfer enhancement and pressure drop for food thermal process. In this study, six different types of corrugated pipes, characterized by different corrugation pitch and depth values, were tested. The result revealed that the capacity of heat transfer enhancement of cross helix geometries was significantly higher than single helix and the performance of smaller pitch size pipes was higher than larger pitch size. Another research carried out the application of corrugated cross helix geometries in gas flow in turbulent regime [10]. To match a common range of gas-liquid heat exchangers, they analysed Reynolds number range was 5000–23000. They found the highest heat transfer with cross-helix tube having corrugation depth 1.86mm, but the drawbacks were higher friction factor.

Based on the literature review, it is concluded that previous studies on cross-helix corrugated pipes have only examined their performance in terms of average heat transfer. The investigation of the local



convective heat transfer coefficient has significance for many different industrial procedures, such as food sterilization and pasteurization treatments. These processes use corrugated walls, and any non-uniform temperature distribution on the inside may compromise their overall performance. As a result, understanding and assessing the convective heat transfer coefficient at the local level is critical to ensuring the efficiency and effectiveness of these industrial applications. To the best of our knowledge, no research has been conducted on the local thermal performance of cross-helix corrugated pipes. In current research this gap starts to be filled with experimental investigation of cross helix local thermal performances by solving the Inverse Heat Conduction Problem (IHCP) [11] in the pipe wall employed by adopting as starting data the temperature measurements acquired by an infrared camera on the outer tube wall. The procedure approach is widely acknowledged to have some complications. This is mainly due to the ill-posed nature of the IHCP problem, which makes it extremely sensitive to even minor changes in the input data.

2. Pipe geometry and Experimental setup

In the present work, two cross-helix corrugated stainless-steel type AISI 304 pipes were tested. In figure 1 it is presented a sketch of the corrugation profile, while in figure 2 there are shown the pictures of the two analysed tubes and in (table 1) the dimensions of the corrugation profile are reported.

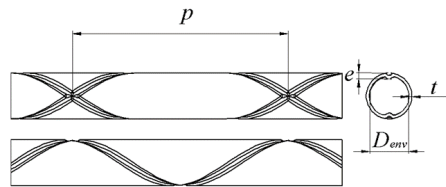


Figure 1. Visual representation of the wall corrugation pipes

Table 1. Dimensions of corrugated profile

Pipes	p (mm)	e (mm)	D_{env} (mm)	t (mm)
T1	16	1	14	1
T2	32	1	14	1

(a)



(b)



Figure 2. Details of the corrugation of pipes (a) T1 (b) T2

The pipes were connected to a test rig comprising of a volumetric pump that delivers the water from the storage tank to the pipe. Water was used as working fluid: before reaching the corrugated pipe, it flows inside a heat exchanger fed with tap water in order to keep constant the temperature at the inlet of the tested pipe. When the working fluid leaves the corrugated pipe, it is returned to the storage tank to restart the cycle.

The entire length of both pipes has three sections such as inlet, heating, and outlet section. The inlet section has a one-meter length, that allows the working fluid to achieve a developed hydrodynamic flow while in the heating section (1.84m) two steel fin electrodes are welded and connected to a power supply (HP 6671A) that operates in the range 0-220V and 0-8A. Joule heating effect was used to obtain the condition of uniform heat generated along the heat transfer section. The pipes were insulated with expanded polyurethane to minimize heat losses. At the inlet and outlet sections, T-type thermocouples were used to measure the temperature of the tube wall as well as the fluid. These thermocouples were

precisely calibrated and linked to a multichannel ice point reference, especially the KAYE K170-50°C, ensuring exact and accurate temperature measurements throughout the experiment.

The graphic illustration of the experimental setup is shown in figure 3. A small portion of insulation was removed from corrugated pipes to be accessible to a thermal imaging camera. Then, that portion was uniformly painted with a thin layer of black paint with known emissivity (0.95). The effective emissivity of paints was estimated at different known temperatures. To measure the wall temperature six images are taken with the help of the system reported in figure 3. The FLIR SC7000 camera was characterized by a 640 x 512 pixels sensor with a sensitivity of 20 mK at 303 K and an accuracy of ± 1 K. The complexity of the image processing procedure stemmed from the non-flat nature of the observed target surface. The image processing technique employed in [12] enabled the correction of optical distortions in the captured images caused by surface curvature. The obtained images were subsequently processed to create a continuous temperature map on the tube wall, due to suitable position references established on the tube wall.

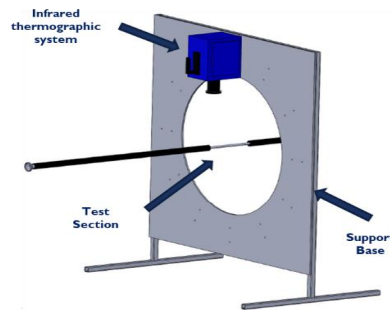


Figure 3. Graphic illustration of local experimental setup

3. Image processing and estimation procedure:

The inverse heat conduction problem (IHCP) for the wall surface was solved by calculating the local convective heat transfer coefficient. This was made possible with the help of the temperature distribution determined by infrared camera measurements. The given physical problem deals with the heat transfer through the wall of a cylindrical pipe (as shown in figure 4a). Moreover, the steady-state energy balance equation governing heat conduction within the domain of the cylindrical pipe's wall is expressed as follows:

$$k\nabla^2 T + q_g = 0 \quad (1)$$

The following two equations defined the boundary conditions.

$$-k \frac{\partial T}{\partial r} = \frac{(T - T_{amb})}{R_{env}}, \quad r = r_{ext} \quad (2)$$

$$-k \frac{\partial T}{\partial r} = h_{int}(T - T_b), \quad r = r_{int} \quad (3)$$

Where T_b is the bulk temperature of fluid flowing inside the tube, T_{amb} is the ambient temperature and R_{env} is the overall heat transfer resistance between the ambient and pipe wall.

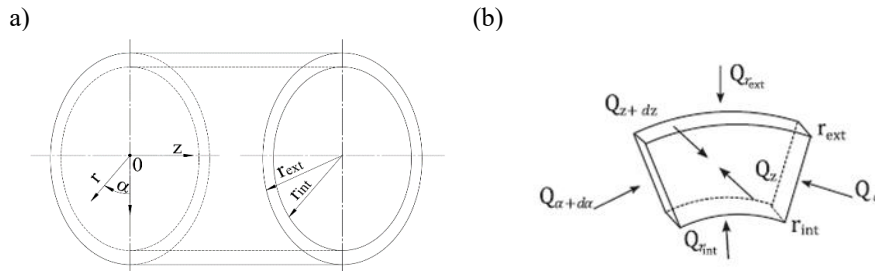


Figure 4. (a) Geometrical domain and b) test section representation.

The temperature on the external surface was assumed to be equal to the temperature on the internal surface under the thin-wall approximation.

$$T(\alpha, r, z) \cong T(\alpha, r_{int}, z) \cong T(\alpha, r_{ext}, z) \quad (4)$$

The Biot number is a dimensionless number that defines as the convective heat transfer coefficient multiplied by tube thickness and tube wall thermal conductivity. When the Biot number is less than 0.1, the thin-wall approximation is valid, suggesting that the conditions are adequate for simplified analysis. [13]. The thin-wall approximation simplifies the analysis in this context by assuming insignificant temperature gradients within the wall, which is generally true for systems with low Biot numbers. Figure 4b depicts the steady-state local energy balance equation with reference to the infinitesimal cylindrical section.

$$Q_{\alpha+dx} + Q_{\alpha} + Q_{r_{ext}} + Q_{r_{int}} + Q_{z+dz} + Q_z + Q_g = 0 \quad (5)$$

By expressing all the terms in Eq. (5) and after some simplifications Eq. (6), one can arrive at the expression for the determination of the local convection coefficient at the internal wall-fluid interface:

$$h_{int} = \frac{k \cdot \ln\left(\frac{r_{ext}}{r_{int}}\right) \cdot \frac{\partial^2 T}{\partial \alpha^2} + \frac{k}{2} (r_{ext}^2 - r_{int}^2) \cdot \frac{\partial^2 T}{\partial z^2} - \frac{r_{ext}}{R_{env}} (T - T_{env}) + \frac{q_g}{2} (r_{ext}^2 - r_{int}^2)}{r_{int} \cdot (T - T_b)} \quad (6)$$

When dealing with discrete noisy data, Eq. (6) produces incorrect results due to the second-order derivative operator's extraordinary sensitivity to even small noise disturbances [11]. The accuracy and stability of the results may be compromised in such instances due to noise amplification during the differentiation process. A suitable method to solve this problem which was found in Gaussian filtering that allows to remove the noise in raw temperature data. Various Researchers [14] evaluated the performance of the Gaussian kernel using this particular method. In a 2-D frequency domain, the transfer function of this sort of filter can be written as follows.

$$H(u, v) = e^{-\left(\frac{u^2 + v^2}{2u_c^2}\right)} \quad (7)$$

It provides valuable insights into the filter's frequency response, enabling the analysis of its behavior and performance in processing signals or images. Where u_c is the cut-off frequency, assuming that equally along the u and v coordinates. The cut-off frequency for this type of filter is unknown in the real application, so in order to successfully apply this method, a benchmark must be chosen. In the present work, this benchmark was given by the discrepancy principle as formulated by Morozov [15].

4. Result and Discussion

In (figure. 5a) the Darcy friction factor, defined by following equation (8) for the two corrugated tubes under test is reported against the Reynolds number together with the results obtained for a smooth tube.

$$f = \frac{\Delta p D}{p L} \frac{2}{w^2} \quad (8)$$

It can be observed that in (figure 5a) the T2 corrugated pipe, when compared to the T1 corrugated pipe, experiences a lower pressure drop in the range of $Re = 4 \times 10^3 - 16 \times 10^3$. Moreover, values of f for both corrugated pipes were higher than those for the smooth tube. To evaluate the global efficiency (η) of both pipes by considering the thermal enhancement effect and the pressure drops caused by the corrugation is calculated by the following equation.

$$\eta = \frac{Nu/0.023 Pr^{0.4} Re^{0.8}}{(f/f_0)^{1/3}} \quad (9)$$

where the subscript 0 refers to the smooth pipe. In (figure 5b) it can be seen that in both corrugated pipes, efficiency is decreases with the increase of Reynolds number. Moreover, it is possible to observe that in terms of efficiency η , both T1 and T2 showing same efficiency at lower $Re = 10^4$, but with the increase of Reynolds number T1 pipe performs better than the T2.

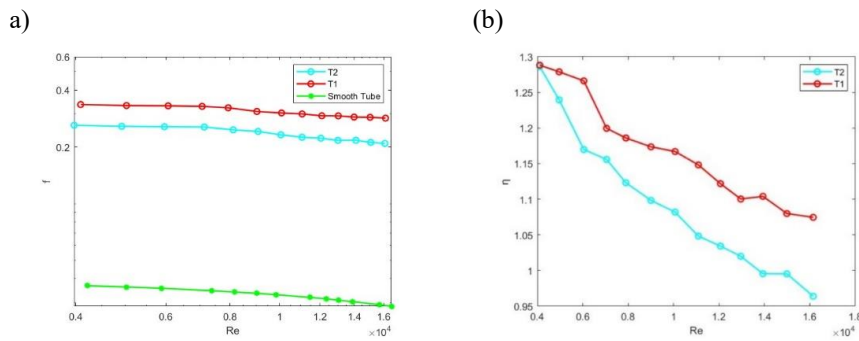


Figure 5. (a) Darcy friction factor vs. Reynolds number b) Efficiency (η) vs. Reynolds number

In this research, the performance of two cross-helix corrugated pipes is experimentally analysed in turbulent regime flow (Re values in the range $4 \times 10^3 - 16 \times 10^3$). For a better understanding of the reader, schematic visualization of the corrugation position and flow direction is shown in figure 6.

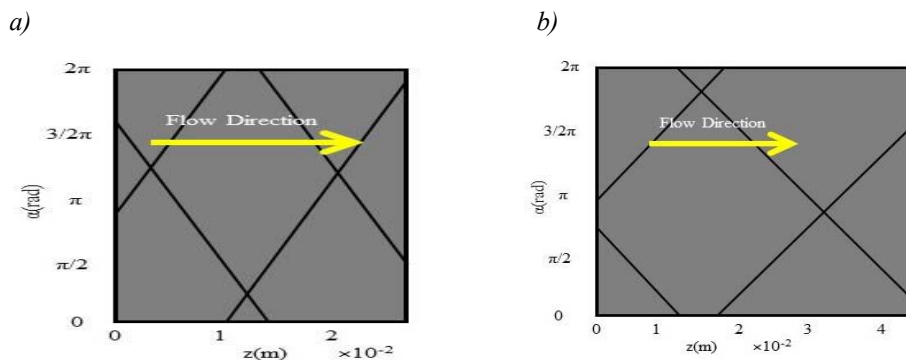


Figure 6. Flow direction and corrugation position in both cross-helix pipes (a) T1 (b) T2

The local heat transfer coefficient distribution at the internal wall to the liquid interface can be restored by starting from the temperature distribution on the outer pipe surface and using the IHCP approach explained in section 3. The filtered temperature maps, obtained by applying the Gaussian filter equation 7 on rough temperature distributions at $Re = 4 \times 10^3$ of both tested pipes is reported in (figure 7), it is possible to observe that immediately after the corrugations, the wall temperature increases in both pipes, reaching a local maximum.

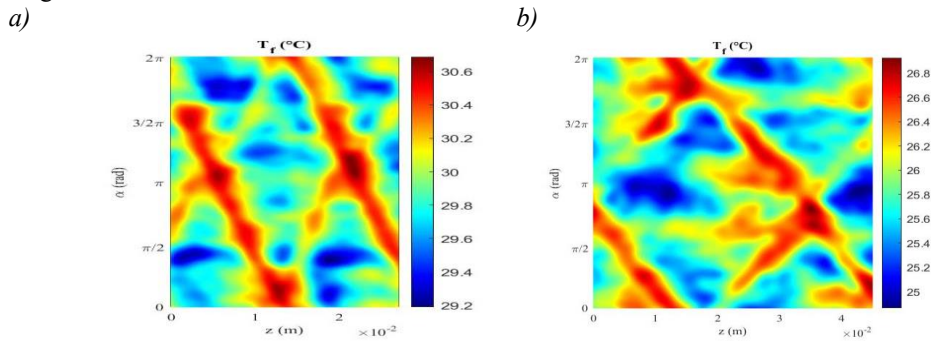


Figure 7. Filtered temperature distributions at $Re=4000$ a) T1 b) T2

The performance of both cross-helix corrugated pipes was investigated by analysing the internal convective heat transfer coefficient distributions for three different Reynolds numbers $Re = 4 \times 10^3$, 10^4 , and 1.6×10^3 were presented in figure 8,9 and 10. The convective heat transfer coefficient (h) confirmed the behavior of the temperature distribution, which had a minimum value after the corrugation and then reached a maximum as the fluid stream followed. The local heat transfer coefficient distribution at $Re=4 \times 10^3$ is more uniform in the case of T1 pipe, promoting a higher heat transfer enhancement. Considering the T1 pipes having smaller pitch size, when compared to the T2 pipe, there is a considerable decrease in the h value between corrugations. The described trend holds true for situations with $Re = 10^4$ and $Re = 16 \times 10^3$.

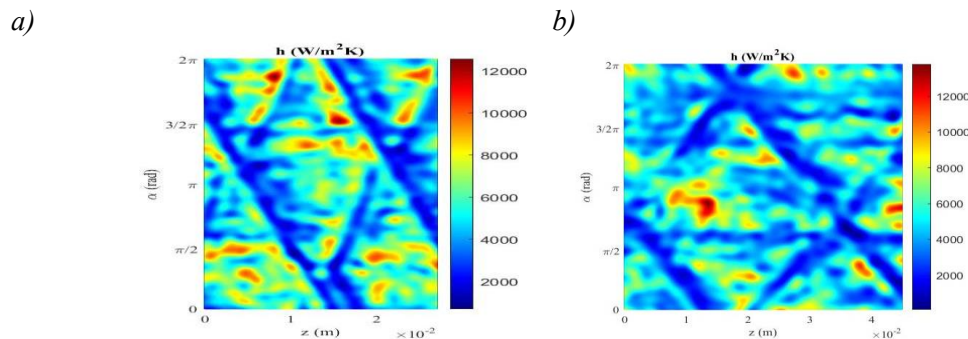


Figure 8. Convective heat transfer distribution (h) at $Re= 4000$ a) T1 b) T2

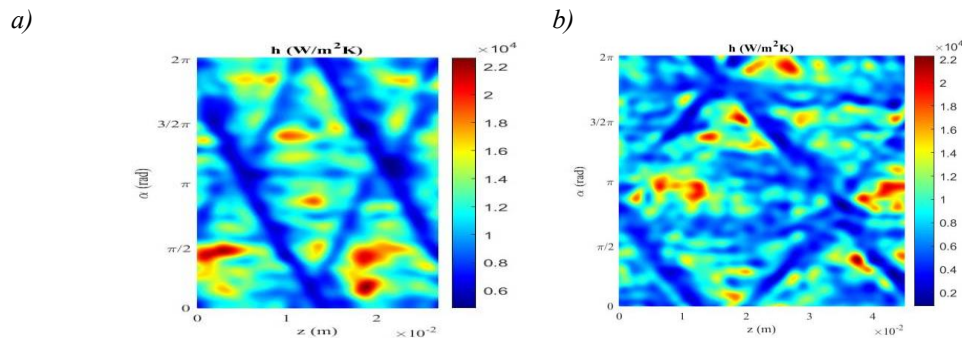


Figure 9. Convective heat transfer distribution (h) at $Re= 10000$ a) T1 b) T2

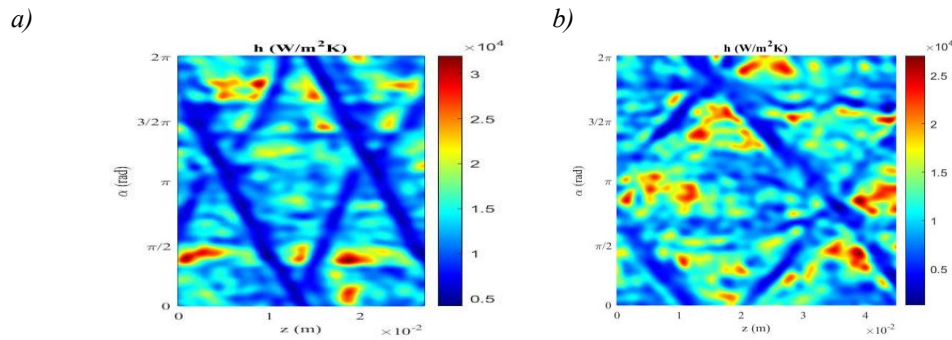


Figure 10. Convective heat transfer distribution (h) at $Re= 16000$ a) T1 b) T2

Moreover, it is feasible to declare that, in those regions, the heat exchange between the fluid and wall decreases. This type of behaviour could be found in the boundary layers disruption effect caused by the fluid confronting surface roughness along the direction of flow. Furthermore, after leaving the corrugation temperature gradually decreases until the next corrugation. This process is repeated when the fluid passes through from each corrugation.

Concluding, in figure 11 the Nusselt distribution for the radial coordinate $\alpha=\pi$ is represented. Also in this case, the behaviour of the internal convective heat transfer coefficient is confirmed: when the fluid flows after the corrugation, Nusselt number immediately decreases and then starts to increase until the next corrugation. This behaviour can be found also in the local convective heat transfer coefficient (h) as shown in previous figures 9,10 and 11. This phenomenon is repeated at each corrugation of the pipes. Moreover, it can be observed that the performance in terms of Nusselt number of T1 pipe is higher when compared to T2 pipe.

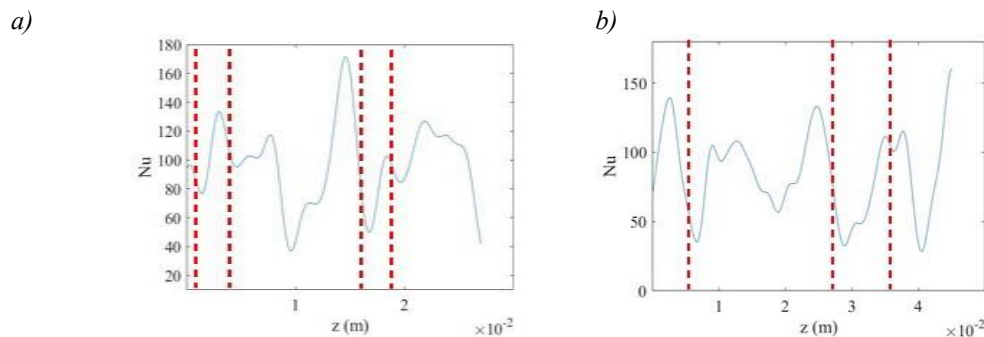


Figure 11. Local Nusselt number at $\alpha=\pi$ at $Re=4000$ a) T1 f) T2

5. Conclusions

In the present research work, the thermal performance of two cross-helix corrugated pipes are investigated from local experimental points of view in a turbulent flow regime. Both pipes have two different corrugation pitch values, i.e., 16 and 32 mm. The analysis of the global performance highlighted for both pipe and T1 pipe obtained higher efficiency as compared T2 pipe. With the increase of Reynold number efficiency decreases of both corrugated cross helix corrugated pipes. However, the drawback of this enhancement in both tested pipes is found that higher pressure drops as compared to reference geometry. Moreover, T1 pipe is the best choice when the aim is to obtain the highest value of convective heat transfer coefficient (h). The results from the measurements conducted locally indicate that using corrugation in both T1 and T2 geometries can effectively enhance the heat transfer capability of heat exchangers in turbulent conditions. Furthermore, another important result of the present research

is the execution and testing of an effective experimental method that could be successfully applied for the optimization of wall corrugation in critical thermal processes.

References:

- [1] R. L. Webb, "Principles of enhanced heat transfer," 1994.
- [2] L. Ventola *et al.*, "Rough surfaces with enhanced heat transfer for electronics cooling by direct metal laser sintering," *Int. J. Heat Mass Transf.*, vol. 75, pp. 58–74, 2014, doi: 10.1016/j.ijheatmasstransfer.2014.03.037.
- [3] M. Sheikholeslami, M. Gorji-Bandpy, and D. D. Ganji, "Review of heat transfer enhancement methods: Focus on passive methods using swirl flow devices," *Renew. Sustain. Energy Rev.*, vol. 49, pp. 444–469, 2015, doi: 10.1016/j.rser.2015.04.113.
- [4] F. Bozzoli, L. Cattani, and S. Rainieri, "Effect of wall corrugation on local convective heat transfer in coiled tubes," *Int. J. Heat Mass Transf.*, vol. 101, pp. 76–90, 2016, doi: 10.1016/j.ijheatmasstransfer.2016.04.106.
- [5] P. Freymuth, "Flow visualization in fluid mechanics," *Rev. Sci. Instrum.*, vol. 64, no. 1, pp. 1–18, 1993, doi: 10.1063/1.1144433.
- [6] Z. S. Kareem, M. N. Mohd Jaafar, T. M. Lazim, S. Abdullah, and A. F. Abdulwahid, "Passive heat transfer enhancement review in corrugation," *Exp. Therm. Fluid Sci.*, vol. 68, pp. 22–38, 2015, doi: 10.1016/j.expthermflusci.2015.04.012.
- [7] N. Kurtulmuş and B. Sahin, "A review of hydrodynamics and heat transfer through corrugated channels," *Int. Commun. Heat Mass Transf.*, vol. 108, p. 104307, 2019, doi: 10.1016/j.icheatmasstransfer.2019.104307.
- [8] W. T. Ji, A. M. Jacobi, Y. L. He, and W. Q. Tao, "Summary and evaluation on single-phase heat transfer enhancement techniques of liquid laminar and turbulent pipe flow," *Int. J. Heat Mass Transf.*, vol. 88, pp. 735–754, 2015, doi: 10.1016/j.ijheatmasstransfer.2015.04.008.
- [9] F. Bozzoli, L. Cattani, and S. Rainieri, "Cross-helix corrugation: The optimal geometry for effective food thermal processing," *Int. J. Heat Mass Transf.*, vol. 147, p. 118874, 2020, doi: 10.1016/j.ijheatmasstransfer.2019.118874.
- [10] A. Harleß, E. Franz, and M. Breuer, "Heat transfer and friction characteristics of fully developed gas flow in cross-corrugated tubes," *Int. J. Heat Mass Transf.*, vol. 107, pp. 1076–1084, 2017, doi: 10.1016/j.ijheatmasstransfer.2016.10.129.
- [11] J. Beck, "Inverse Heat Conduction Ill-posed problems," pp. 137–157, 179–180, 198–211, 2011.
- [12] F. Bozzoli, L. Cattani, and S. Rainieri, "Effect of wall corrugation on local convective heat transfer in coiled tubes," *Int. J. Heat Mass Transf.*, vol. 101, pp. 76–90, 2016, doi: 10.1016/j.ijheatmasstransfer.2016.04.106.
- [13] D. F.P. and De Witt, *Fundamentals of Heat and Mass Transfer*, vol. 4, no. 1. 2002.
- [14] F. Bozzoli, L. Cattani, G. Pagliarini, and S. Rainieri, "Infrared image filtering applied to the restoration of the convective heat transfer coefficient distribution in coiled tubes," *Opto-Electronics Rev.*, vol. 23, no. 1, pp. 107–115, 2015, doi: 10.1515/oere-2015-0004.
- [15] V. A. Morozov, *Methods for Solving Incorrectly Posed Problems*. New york. [Online]. Available: <https://www.ptonline.com/articles/how-to-get-better-mfi-results>

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