Strojnícky časopis – *Journal of* MECHANICAL ENGINEERING,
VOL 75 (2025), NO 2, 21 – 30

COMPUTATIONAL ANALYSIS OF CROSS-FLOW HEAT EXCHANGERS: EFFECT OF FIN GEOMETRY ON THERMAL AND HYDRAULIC PERFORMANCE

BOUHABEL Bourhane[1*], ABDELFATAH Marni sandid[1], ZOUHAIR Aouissi[1], SALIM Eutamene[1], HAMZA Benhacine[1], YAZID Ait Ferhat[1]

[1] *Center of Research in Mechanics (CRM), BP N73B, Freres Ferrad, Ain El Bey, Constantine,25021, Algeria*
Corresponding Author e – mail: bourhane18@gmail.com

Abstract: A numerical study using ANSYS Fluent 18.1 software was conducted to examine the fluid flow and heat transfer characteristics in a cross-flow heat exchanger equipped with rectangular, hexagonal, and semi-elliptical finned tubes. Simulations were performed for Reynolds numbers ranging from 3422 to 11980. The results show that rectangular fins produce the highest heat transfer values with a high pressure drop. However, semi-elliptical fins offer superior thermal and hydraulic efficiency. At Re = 5990, the semi-elliptical configuration improves overall efficiency by 5.17% and reduces pressure drop by 47.86% compared to the rectangular fin design.

KEYWORDS: Numerical study, ANSYS Fluent, Heat transfer, Reynolds numbers, pressure drop, efficiency.

1 Introduction

Heat exchangers are essential components in fast reactors [1] and also play a crucial role in industrial installations, featuring various flow configurations [2]. Among them, crossflow heat exchangers are widely used due to their compactness and efficiency. In these systems, the separation of flow around the tubes creates recirculation zones, whose influence intensifies with increasing Reynolds number [3]. Optimizing heat transfer in such exchangers has been the focus of extensive research. For instance, Karali [4] demonstrated that adding baffle plates and surface roughness enhances heat transfer with minimal pressure drop penalty, while Choudhary et al. [5] showed that perforated plates can improve both heat transfer and pressure drop simultaneously. Numerical studies have further emphasized the significance of tube geometry and arrangement. Elliptical tube exchangers, particularly those with optimized longitudinal and transverse pitch ratios and an aspect ratio of 0.53, have been shown to improve compactness and energy efficiency [6,7]. Additional investigations highlight the effectiveness of passive enhancements such as tapered longitudinal fins [8] and separation plates [9,10], which can significantly increase the Nusselt number while reducing the friction factor. Altogether, these findings underline the critical role of design geometry and passive devices in enhancing the thermohydraulic performance of crossflow heat exchangers.

2 Model description and numerical methods

2.1 Physical model

The geometric configuration investigated in this two-dimensional simulation concerns a crossflow heat exchanger (figure 1), designed to analyze the impact of different fin shapes on thermal and hydrodynamic performance. The flow domain measures 600 mm in length and 150 mm in height, within which cylindrical tubes of 25 mm diameter are arranged in a staggered (triangular) configuration. Four geometric variants are considered. The first, used as

DOI: 10.2478/scjme-2025-0020, Print ISSN 0039-2472, On-line ISSN 2450-5471

a reference configuration, includes no fins, allowing for a baseline comparison of thermal enhancement provided by the other designs. The second configuration features rectangular fins measuring 10 mm in length and 5 mm in width, symmetrically placed radially around each tube to increase heat exchange surface and induce flow disturbances. The third configuration employs hexagonal fins mounted around the tubes, aiming to enhance heat transfer while generating moderate turbulence. The fourth configuration introduces streamlined semi-elliptical fins aligned with the flow direction, with a major axis of 20 mm and a minor axis of 10 mm, designed to improve thermal performance while minimizing pressure losses. These different geometries are compared in order to evaluate their respective thermal efficiency and pressure drop characteristics, with the objectif of optimizing the design of crossflow heat exchangers.

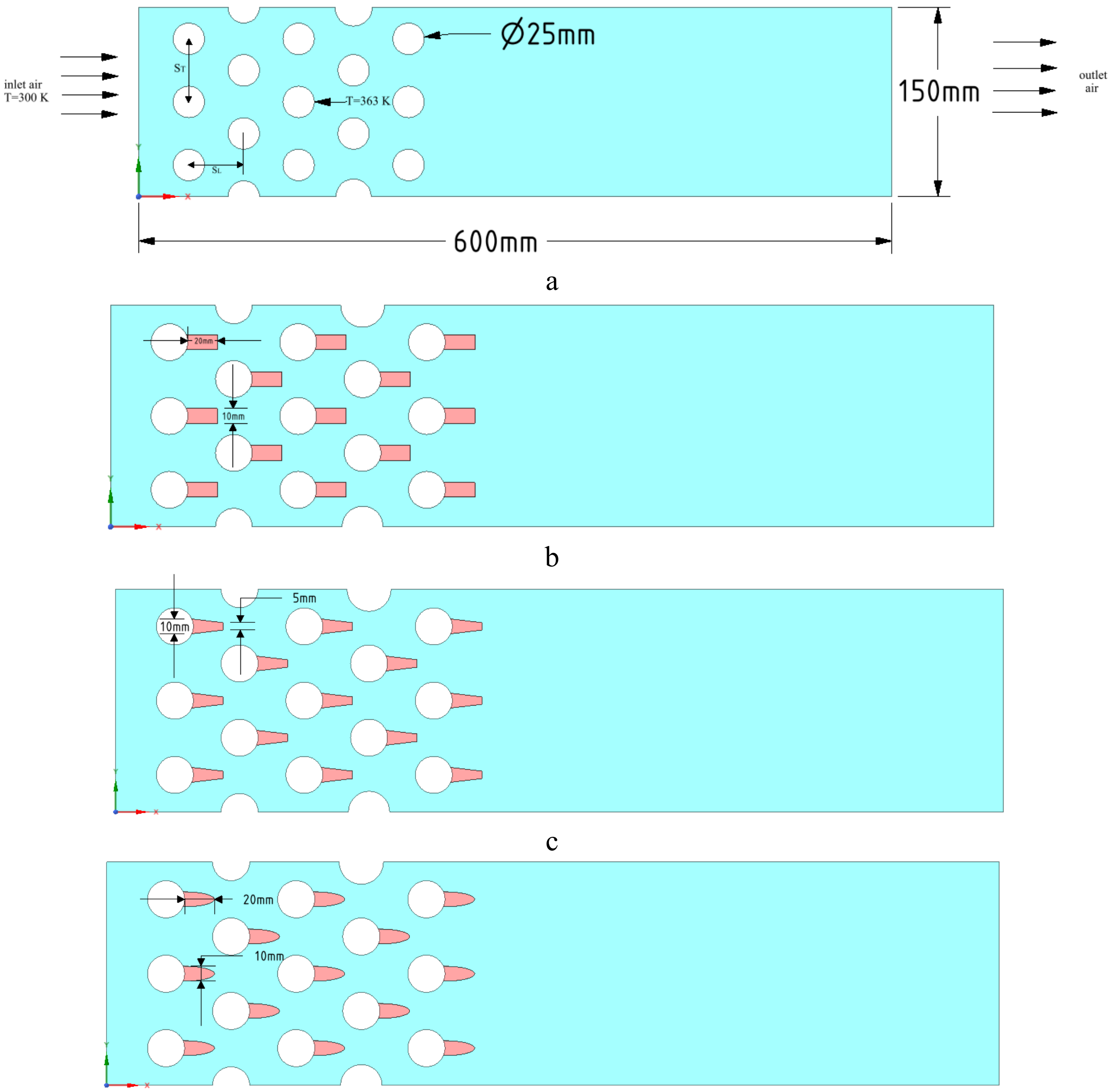

Fig. 1 Different geometric configurations of the heat exchanger: a) circular tube heat exchanger b) circular tube heat exchanger with rectangular fins c) circular tube heat exchanger with hexagonal fins d) circular tube heat exchanger with semi-elliptical fins.

2.2 Governing equation and boundary conditions

The governing equations for the fluid domain consist of the fundamental principles of conservation of mass, momentum, and energy. The fluid flow is assumed to be steady, incompressible, and Newtonian, with constant thermophysical properties. Therefore, the governing equations for the fluid domain in Cartesian tensor form are given as follows:

Continuity equation:

$$\frac{\partial_{u_i}}{\partial_{u_x}} = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial}{\partial_{x_i}}\left(\rho u_j u_j\right) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u_i u_j}\right) \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial_{x_i}}\left(\rho u_j T\right) = \frac{\partial}{\partial_{x_j}}\left((\tau + \tau_t)\frac{\partial T}{\partial_{x_j}}\right) \tag{3}$$

where τ and τ_t represents the molecular thermal diffusivity and turbulent thermal diffusivity which can be defined as: $\tau=\mu/Pr$ and $\tau_t=\mu_t/Pr_t$

Turbulent kinetic energy [6]:

$$\frac{\partial}{\partial_{x_j}}(\rho k u_i) = \frac{\partial}{\partial_{x_i}}\left(\alpha_k \mu_{eff}\frac{\partial k}{\partial_{x_j}}\right) + G_k - \rho\varepsilon \tag{4}$$

Turbulent kinetic energy dissipation [6]:

$$\frac{\partial}{\partial_{x_i}}(\rho\varepsilon u_i) = \frac{\partial}{\partial_{x_i}}\left(\alpha_k \mu_{eff}\frac{\partial\varepsilon}{\partial_{x_j}}\right) + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} - R_\varepsilon \tag{5}$$

Where G_k in Equations. (4) and (5), de notes the turbulence kinetic energy generation due to mean velocity gradients,

$$G_k = \left(-\rho u_i u_j \frac{\partial u_j}{\partial_{x_i}}\right) \tag{6}$$

Similarly, μ_{eff} denotes effective turbulent viscosity by considering the effect of variation in turbulent viscosity, $\mu_{eff} = \mu + \mu_t$.

2.3 Solution methods

The numerical study is performed with the Ansys Fluent 18.1 software, for a turbulent fluid regime using RNG k–ε two-equations turbulence model, with the fluid being air entering the heat exchanger a temperature of 300°. The heat exchanger tubes are at a constant temperature of 363°k.

The Nusselt number is calculated using the following expression:

$$Nu = \frac{hD_h}{k} \tag{7}$$

Zukauskas also calculates the Nusselt number according to the following correlation [9]:

$$Nu = CRe_{max}{}^{m}Pr^{0.36}\left(\frac{Pr}{Pr_w}\right)^{1/4} \tag{8}$$

Where C and m are constants.

The pressure drop is calculated using the following expression:

$$f = \frac{2\Delta P}{\rho u_{max}{}^{2} N_R} \tag{9}$$

N_R number of tube rows

u_{max} maximum velocity

Thermo-hydraulic performance efficiency is defined as:

$$\eta = \frac{(Nu/Nu_0)}{(f/f_0)^{0.33}} \tag{10}$$

2.4 Validation

The validation of the computational code was confirmed by comparing the variation of the Nusselt number as a function of the Reynolds number obtained from our CFD results with that derived from the Zukauskas correlation (Fig 2). There is a good agreement between the Nusselt number values, which are closely aligned, indicating that the code is considered valid. Fig 2 presents a comparison between the numerical results obtained with Ansys Fluent and the literature data for the variation of the Nusselt number as a function of the Reynolds number in the case of a circular tube heat exchanger without fins. The strong concordance between the two curves further confirms the validity of the numerical model and the reliability of the adopted mesh and boundary conditions. The observed discrepancies remain minor and can be attributed to differences in turbulence models, simplified geometric assumptions, or the local mesh accuracy around the tubes. This validation demonstrates that the employed simulation method accurately replicates the forced convection phenomena in the studied domain, thereby ensuring the credibility of the results for other fin configurations.

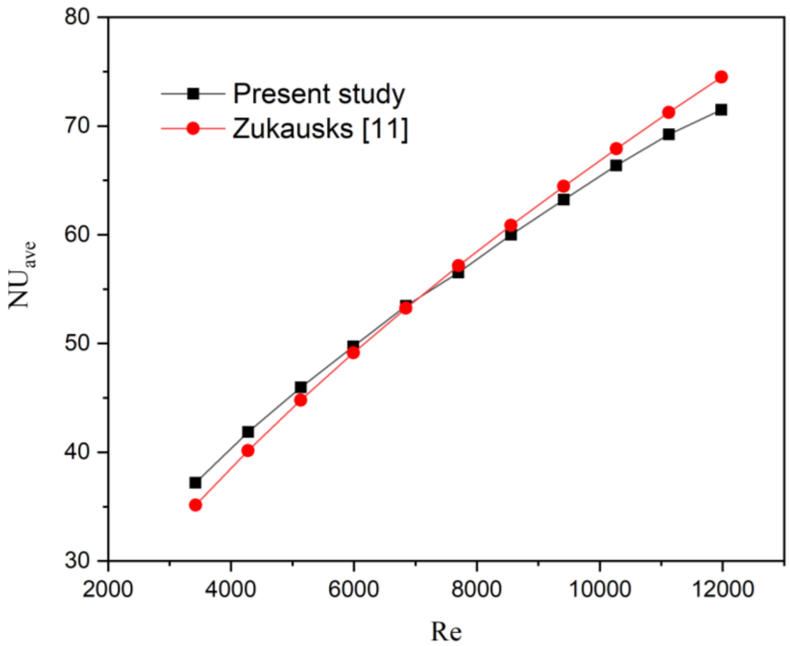

Fig. 2 Variation of the Nusselt number as a function of the Reynolds number

2.5 Grid independency

A mesh convergence study was conducted to ensure that the results were independent of mesh density before proceeding with the numerical simulations. The circular tube exchanger configuration was chosen as the reference case. Four meshes were tested, comprising 90215, 52940, 32512, and 23695 cells, respectively. The convergence analysis revealed that a variation of less than 0.008% was observed between the two finest meshes, indicating that mesh independence had been achieved. Consequently, the 52 940 cell mesh was selected for all simulations related to the study of heat transfer and fluid flow. The same mesh validation methodology was applied to heat exchangers with rectangular, hexagonal, and semi-elliptical fins to ensure the numerical accuracy and reliability of the results obtained for each configuration.

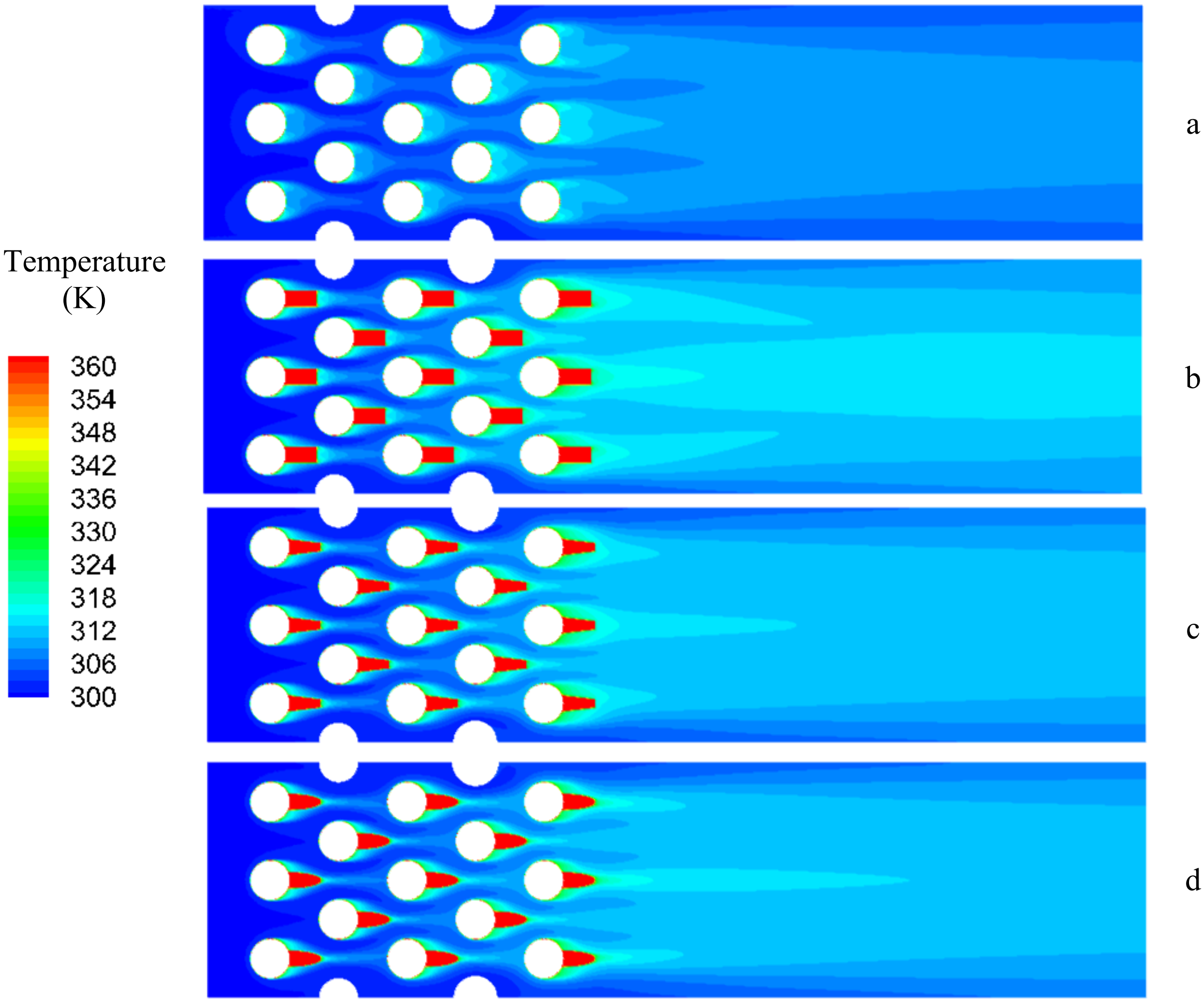

Fig. 3 Temperature distribution in the cross-flow heat exchanger a) without fins b) with rectangular fins c) with hexagonal fins d) with semi-elliptical fins

3 Results and discussion

The temperature distribution in the heat exchanger for the four different shapes is presented in Figure 3. In the case without fins (case a), large wakes are observed behind each cylinder, accompanied by low-velocity zones extending downstream. At high Reynolds numbers, these wakes become more extensive and more sensitive to instabilities, suggesting a regular separation and periodic vortex shedding. The mass and momentum transport downstream is significant, but the heat exchange between the hot and cold fluids remains limited due to the smooth shape of the cylinder. With rectangular fins (case b), the sharp edges induce early separations and generate multiple small recirculation zones near the edges. The velocity field is strongly disturbed downstream, with strong local accelerations around the corners and a wake richer in small structures, resulting in intense mixing, accentuated shear, and a notable increase in the local convection coefficient.In the case of hexagonal fins (case c), even more attachment and detachment points appear at the edges, making the flow field even more unsteady than in the rectangular case. At Re = 8557, these edges promote the formation and maintenance of small-scale vortices, producing a particularly turbulent wake; The semi-elliptical fins (case d) have a more streamlined geometry, which allows the flow to remain attached longer along the curvature, delaying or even reducing separation. The velocity field is perturbed, but more smoothly; wake formation is less abrupt than in the case of angular shapes. At high Reynolds numbers, these fins partially reduce the intensity of small dissipative structures compared to more abrupt geometries.

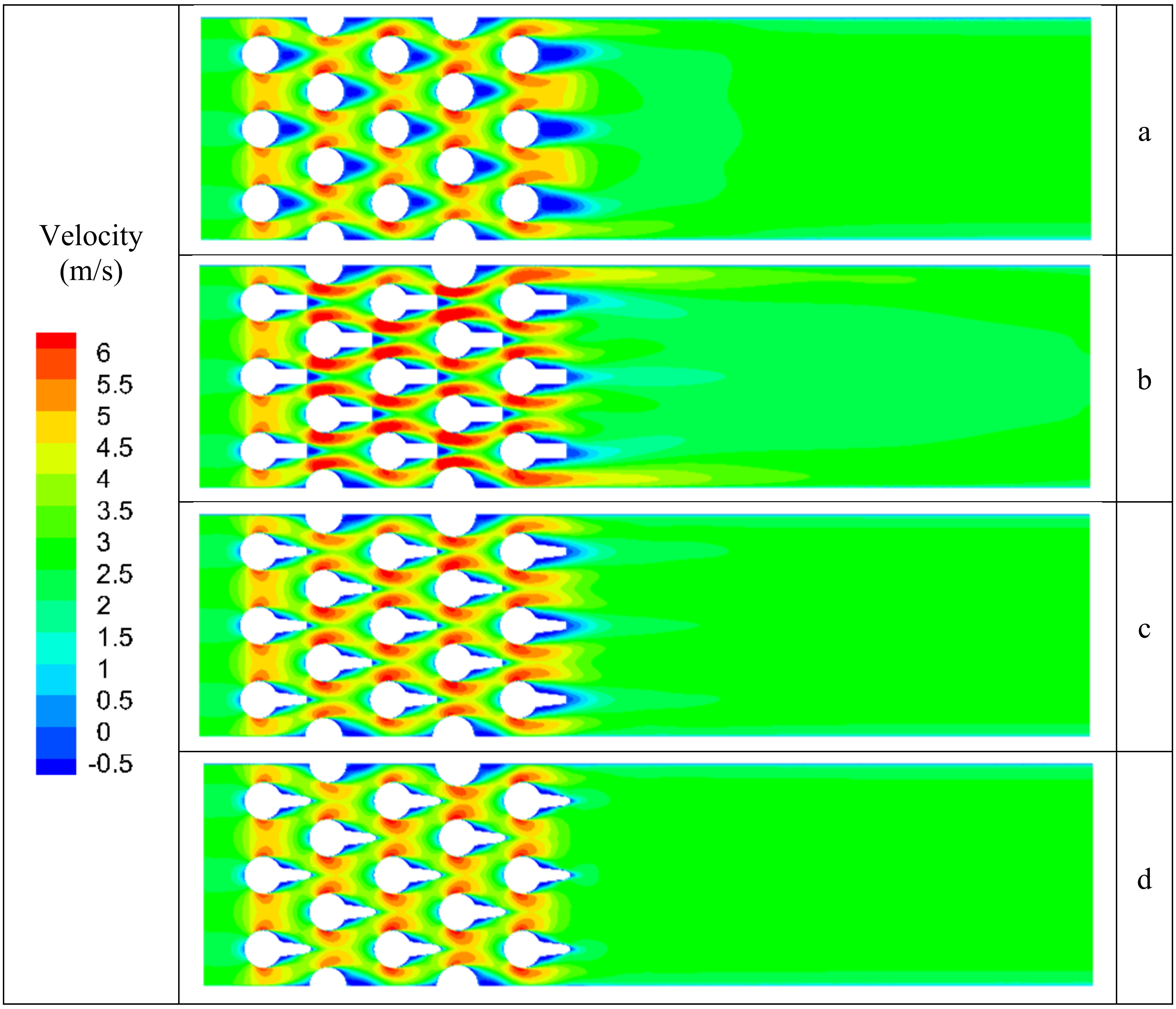

Fig. 4 Velocity distribution in the cross-flow heat exchanger a) without fins b) with rectangular fins c) with hexagonal fins d) with semi-elliptical fins

The velocity distribution inside the heat exchanger for the four fin geometries, as shown in Figure 4, shows a significant dependence on the fin shape at a Reynolds number of 8557. The flow and temperature fields clearly reveal the impact of fin geometry on flow behavior and heat transfer performance.

In the absence of fins, the flow presents highly developed recirculation zones downstream of each tube, forming low-velocity wakes. These wakes contribute to a fast thickening of the thermal boundary layer and reduce thermal mixing between hot and cold areas. This is shown by relatively uniform and widely spaced isotherms around the cylinders, indicating limited heat transfer.

The introduction of rectangular fins causes early flow separation at the edges of the fins, generating strong local shear zones. These zones improve fluid mixing and reduce the thickness of the thermal boundary layer, as shown by the distorted and tightly packed isotherms near the heated surfaces. This results in a significant improvement in heat transfer, but at the cost of increased pressure drop.

The hexagonal fins enhance these effects even further. The multiple sharp edges break up the wake more effectively and amplify the downstream vortex structures, increasing convective mixing. This results in more pronounced isothermal distortion and a more uniform temperature distribution in the exchanger. However, these improvements are accompanied by a significant increase in flow resistance.

The semi-elliptical geometry of the fins, on the other hand, promotes more uniform flow characteristics due to the slower separation induced by its streamlined shape. Velocity contours become more continuous and temperature gradients are more evenly distributed around the tubes. This configuration maintains efficient heat transfer while minimizing pressure drop. At Re = 8557, it offers the best compromise between thermal performance and hydraulic efficiency, combining high energy efficiency with stable flow characteristics.

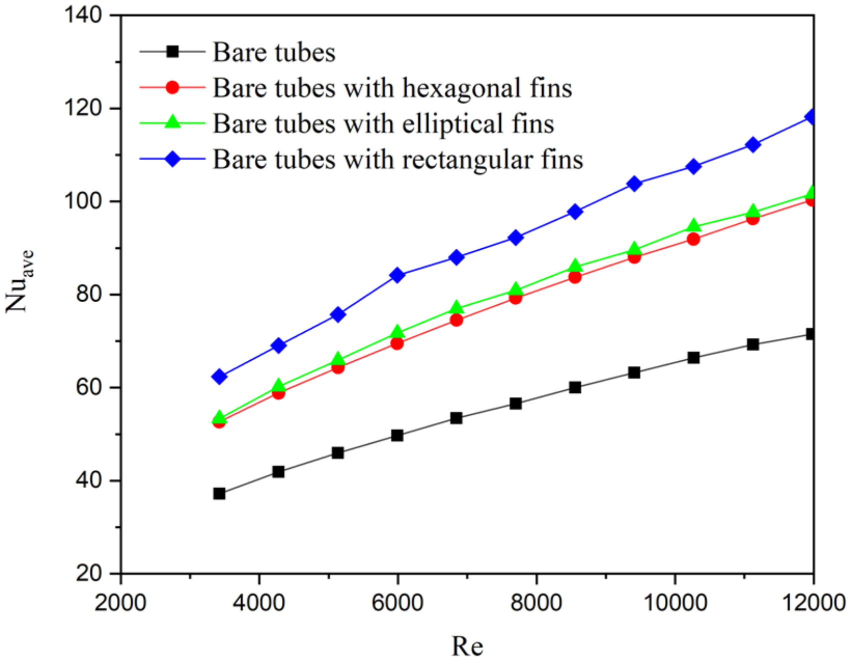

Fig. 5 Variation in Nusselt number as a function of Reynolds number for different heat exchangers

Figure 5 illustrates the evolution of the average Nusselt number (Nu) as a function of the Reynolds number (Re) for the four heat exchanger configurations: without fins, and with rectangular, hexagonal, and semi-elliptical fins. In all configurations, the Nusselt number increases with the Reynolds number, reflecting the improvement in forced convection as the flow velocity increases.

However, the amplitude of Nu varies considerably depending on the geometry of the fins. The configuration without fins has the smallest Nu values, mainly due to its limited heat transfer surface and relatively stable flow, which limit mixing and heat exchange. In contrast, rectangular and hexagonal fin arrangements show a noticeable improvement in thermal performance. This improvement is attributed to the generation of secondary vortices and increased disruption of the thermal boundary layer, which promote more efficient convective heat transfer.

The semi-elliptical configuration of the fins also allows high Nusselt values to be achieved, but with a more regular and gradual increase as Reynold increases. This behavior indicates an efficient and stable convection regime. Consequently, the addition of fins significantly improves thermal efficiency, with the semi-elliptical geometry standing out as the optimal compromise between heat transfer performance and flow stability.

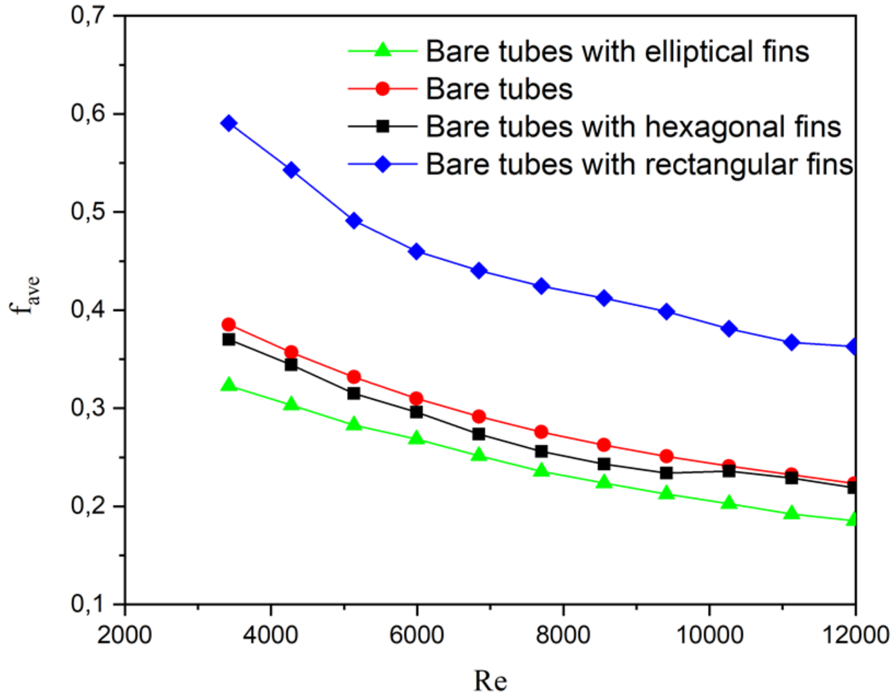

Fig. 6 Variation in pressure drop as a function of Reynolds number

Figure 6 presents the variation of pressure drop as a function of the Reynolds number for the different heat exchanger designs. As expected, pressure losses increase with Re due to the rising influence of inertial forces and wall friction at higher flow velocities.

While the addition of fins enhances heat transfer performance, it also introduces additional flow resistance. Among the finned configurations, the hexagonal and rectangular fins result in the highest pressure drops. This is primarily due to their sharp edges and the associated formation of strong recirculation zones, which increase flow obstruction.

In contrast, the semi-elliptical fins exhibit significantly lower pressure drop, benefiting from their streamlined shape which reduces flow separation while maintaining effective fluid mixing. The finless configuration, although exhibiting the lowest hydraulic resistance, offers the least efficient thermal performance.

These results highlight the necessity of achieving a balanced design that optimizes heat transfer while minimizing the energy cost associated with fluid pumping.

Figure 7 illustrates the variation of the overall performance coefficient (η), defined as the ratio between thermal enhancement represented by the increase in Nusselt number and the hydraulic penalty, quantified by the rise in pressure drop (ΔP). The data show that the performance coefficient increases with Reynolds number up to a certain threshold, beyond which it begins to level off. This trend indicates a gradual balance between improved heat transfer and growing flow resistance.

Among all the configurations, the semi-elliptical fin geometry consistently achieves the highest performance coefficient values, highlighting its superior compromise between thermal efficiency and flow resistance. In contrast, although the rectangular and hexagonal fins lead to higher Nusselt numbers, their overall performance is reduced due to significant pressure losses.

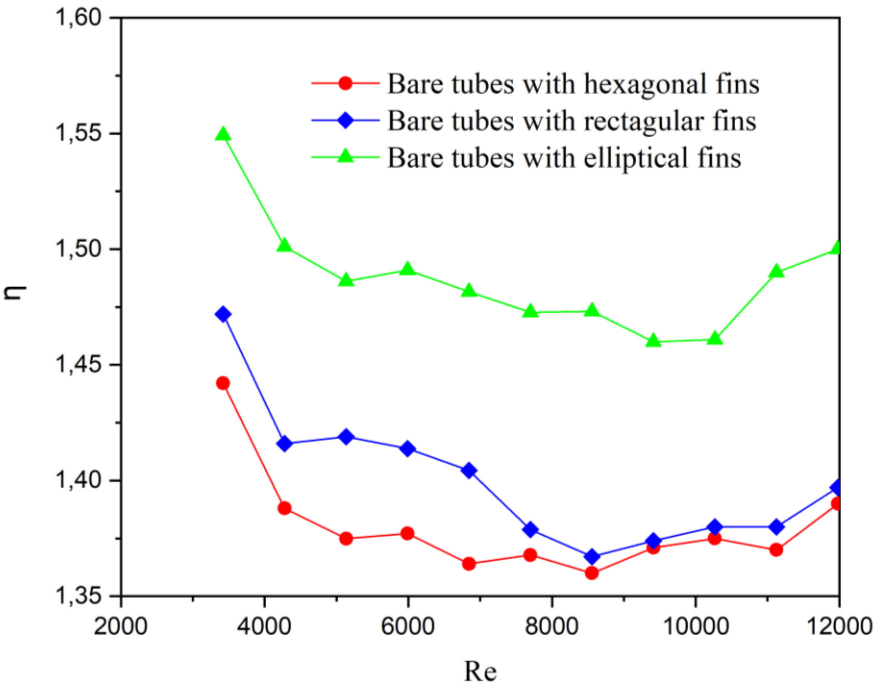

Fig. 7 variation in performance coefficient as a function of Reynolds number

The finless configuration, while exhibiting the lowest hydraulic resistance, underperforms in terms of heat exchange, resulting in a low overall efficiency. These findings underscore the energy efficiency of the semi-elliptical design, especially in moderate to high Reynolds number regimes, where both heat transfer and pressure drop must be carefully balanced.

CONCLUSION

The simulation results clearly demonstrate the significant influence of fin geometry on fluid dynamics and heat transfer efficiency within the heat exchanger. In the configuration without fins, the flow tends to develop recirculation zones that inhibit thermal mixing and reduce overall heat transfer performance. In contrast, the addition of fins, whether rectangular, hexagonal, or semi-elliptical, significantly improves thermal efficiency.The rectangular and hexagonal fins intensify fluid mixing by promoting flow perturbations, which improves convective heat transfer. However, this improvement is accompanied by an increase in pressure drop due to greater flow resistance. On the other hand, semi-elliptical fins, thanks to their aerodynamic profile, provide a better balance between heat transfer and flow resistance. This results in efficient thermal performance while minimizing energy losses associated with fluid pumping.The global performance evaluation, expressed by the performance coefficient, indicates that the semi-elliptical fin configuration improves performance compared to the other configurations, particularly in regimes with moderate to high Reynolds numbers. These results demonstrate that optimizing heat exchanger design is crucial to achieving maximum thermal efficiency while minimizing hydraulic energy costs, making the semi-elliptical geometry a particularly suitable choice for industrial applications.

NOMENCLATURE

D	Diameter of the tubes (mm)	k	Thermal Conductivity, W/m.K
D_h	hydraulic diameter, m	p	pressure, N/m^2
S_T	Transverse and longitudinal tube pitch (mm)	ΔP	Pressure drop across the tube bank (Pa)
S_L	Longitudinal tube pitch (mm)	Nu	Nusselt number for test configuration
T	Temperature (K)	Nu_o	Nusselt number for bare tube bank
T_S	Surface temperature of the tube (K)	η	Thermo-hydraulic performance efficiency
u_{max}	Maximum velocity of air (m/s)	u	velocity, m/s
f	Friction factor for test configuration	x,y	cartesian coordinates, m
f_o	Friction factor for bare tube bank	μ	dynamic viscosity, kg/ms
h	heat transfer coefficient, W/m^2K	ρ	density, kg/m^3

REFERENCES

[1] Guželа, Š., Dzianik, F., Juriga, M., Kabát, J. "Shell and Tube Heat Exchanger – the Heat Transfer Area Design Process", Strojnícky časopis – Journal of Mechanical Engineering 67 (2), pp. 13 – 24, **2017**. DOI: 10.1515/scjme-2017-0014

[2] Guželа, Š., Dzianik, F. "Countercurrent index determination for the recuperative heat exchangers", Strojnícky časopis – Journal of Mechanical Engineering 71 (1), pp. 51 – 60, **2021**. DOI: 10.2478/scjme-2021-0005

[3] Ahmed, M.A., Yaseen, M.M., Yusoff, M.Z. "Numerical Study of Convective Heat Transfer from Tube Bank in Cross Flow Using Nanofluid", Case Studies in Thermal Engineering 10, pp. 560 – 569, **2017**. DOI: 10.1016/j.csite.2017.11.002

[4] Karali, A. et al "Effect of surfaces roughness of a staggered tube bank in cross flow with air on heat transfer and pressure drop", Case Studies in Thermal Engineering 43, 102779, **2023**. DOI: 10.1016/j.csite.2023.102779

[5] Choudhary, A., Kumar, M., Patil, A. K., Chamoli, S. "Enhanced thermal and fluid flow performance of cross flow tube bank with perforated splitter plate", Experimental Heat Transfer 34 (4), pp. 329 – 341, **2020**. DOI:10.1080/08916152.2020.1749190

[6] Mohanan, A. K., Prasad, B. V. S. S. S., Vengadesan, S. "Flow and Heat Transfer Characteristics of a Cross-Flow Heat Exchanger with Elliptical Tubes", Heat Transfer Engineering 42 (21), pp. 1846 –1860, **2020**. DOI: 10.1080/01457632.2020.1826742.

[7] Reynoso-Jardón, E. et al "Numerical study of a cross flow heat exchanger varying the transverse and longitudinal space to increase heat transfer", International Journal of Combinatorial Optimization Problems and Informatics 11(1), pp. 106 – 114, **2020**. ISSN: 2007-1558

[8] Abraham, J. D., Dhoble, A. S., Mangrulkar, C. K. "Numerical analysis for thermo-hydraulic performance of staggered cross flow tube bank with longitudinal tapered fins", International Communications in Heat and Mass Transfer 118, 104905, **2020**. DOI: 10.1016/j.icheatmasstransfer.2020.104905

[9] Ahmed, M. N. E. et al "Performance enhancement for tube bank staggered configuration heat exchanger – CFD Study" ,Chemical Engineering and Processing- Process Intensification 164, 108392, **2021**. DOI: 10.1016/j.cep.2021.108392

[10] Mangrulkar, C. K., Dhoble, A. S., Chakrabarty, S. G., Wankhede, U. S. "Experimental and CFD prediction of heat transfer and friction factor characteristics in cross flow tube bank with integral splitter plate", International Journal of Heat and Mass Transfer 104, pp. 964 – 978, **2017**. DOI: 10.1016/j.ijheatmasstransfer.2016.09.013

[11] Žukauskas, A. "Heat transfer from tubes in cross flow", Adv. Heat Transf, pp. 93 – 160, **1972**. DOI: 10.1016/S0065-2717(08)70038-8