



Effect of Fluid Temperature, Reynolds Number, and Baffle Design on the Hydrothermal Behavior of Tubular Heat Exchangers

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ABSTRACT

Published: december 31st 2025

Keywords:

Heat exchanger, Baffle geometry, Thermal performance factor, Numerical simulation

This study investigates the thermal and hydrodynamic performance of a tubular heat exchanger equipped with internal baffles of different geometries. Numerical simulations were conducted to evaluate the influence of key operating parameters, including fluid temperature (313 K, 333 K, and 353 K) and Reynolds number. The results show that increasing the fluid temperature enhances heat transfer due to the larger temperature gradient between the fluid and the wall. Similarly, higher Reynolds numbers promote stronger flow turbulence, leading to significant disruption of the thermal boundary layer and improved heat transfer performance. The performance factor increases from 1.51% to 2.26% as the Reynolds number rises from 30 to 50, confirming its major effect on thermal enhancement. Furthermore, the analysis of baffle geometries reveals that Configuration No. 1 provides the best performance, offering an optimal balance between heat transfer enhancement and pressure drop. Overall, the study demonstrates that combining higher fluid temperature, increased Reynolds number, and an optimized baffle design effectively improves the thermal efficiency of heat exchangers.

1. INTRODUCTION

Enhancing heat transfer in tubular heat exchangers is a major challenge for improving the energy efficiency of industrial processes. These devices are widely used in chemical engineering, heat transfer applications—particularly in polymerization processes—petrochemical industries, and compact cooling systems. Optimizing their performance relies on improving both heat transfer and hydrodynamic behavior while minimizing pressure drop and energy consumption.

In tubular heat exchangers, heat transfer is generally limited by the formation of a thermal boundary layer. Introducing internal structures such as baffles (helical, square, triangular, etc.) helps disrupt this boundary layer, increase turbulence, and promote better cross-flow mixing. This approach has been the subject of numerous experimental and numerical studies aiming to analyze the

effect of baffle shape on the thermal performance of heat exchangers [1–2].

Among the most widely studied solutions, different baffle configurations play a major role. Transverse, longitudinal, V-shaped, W-shaped, and delta-shaped baffles have all demonstrated their ability to significantly enhance the heat transfer coefficient, although often at the cost of increased pressure drop [3–7]. Several studies have highlighted the strong influence of parameters such as baffle inclination, height, and spacing on flow distribution and the formation of recirculation zones, which are essential for thermal mixing. Recent research focuses on geometric optimization of these parameters to maximize the thermal performance–pressure drop ratio [8–11].

In parallel, extended surfaces—such as fins or corrugated channels—have also attracted considerable interest. Heat exchangers equipped with corrugated channels provide a larger contact surface and a continuous disturbance of the boundary layer, offering an attractive compromise

2. DESCRIPTION OF THE PROBLEM STUDIED

The geometry of the problem under consideration is illustrated in Figure 1. It is a rectangular channel whose upper and lower walls are equipped with rectangular baffles. All the necessary geometric parameters are detailed in Table 1 and Figure 1. The fluid used is a Ketrol solution with the following rheological properties: consistency index $k=9.5 \text{ Pa sn}$, flow index $n=0.8$, and density $\rho= 997 \text{ kg/m}^3$. The inlet temperature of the hot fluid is 333 K, while the temperature of the upper and lower walls of the channel is maintained at 268 K. Table 1: Details on the geometric parameters of the problem studied

H/L	l/L	h/L	W/L	w/L
0.05	0.156	0.025	0.156	0.005

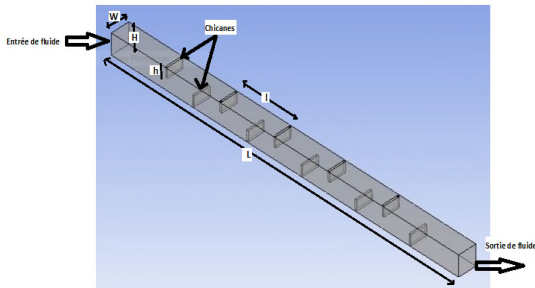


Figure. 1 Dimensionssion of the geometry

2 RESULTS AND DISCUSSION

3. NUMERICAL VALIDATION

Figure 2, as shown above, illustrates the axial velocity for $Re=200$. We validated our model by comparing the axial velocity with the experimental results of De Martini et al. (2005) and the numerical results of Ameer H. (2020). The results obtained show good agreement with these studies.

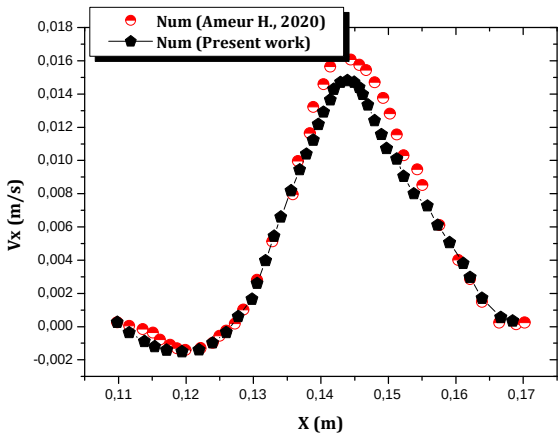
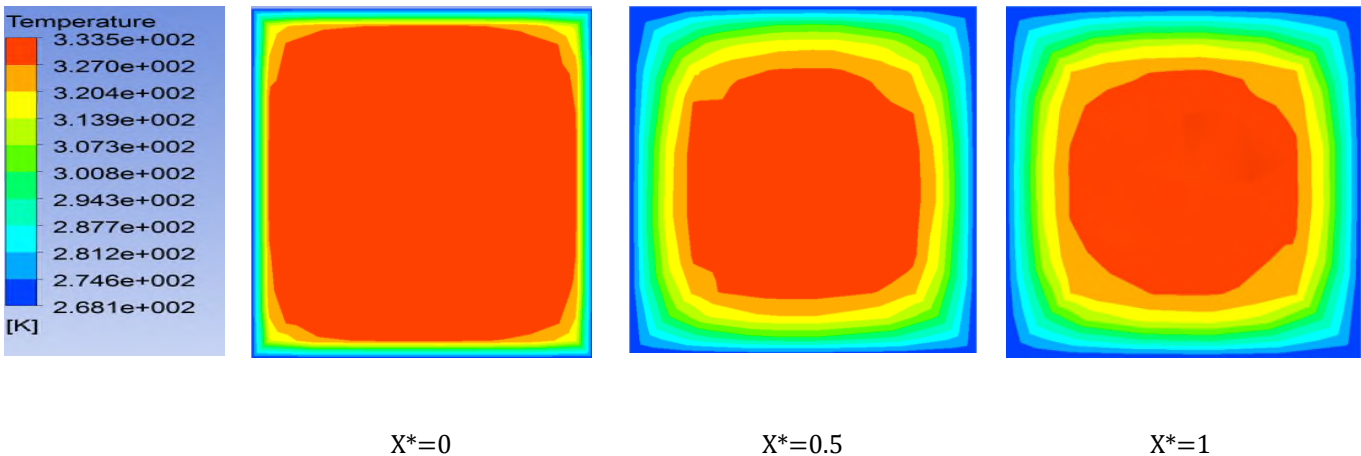


Figure. 2 Axiale Velocity for $Re=200$

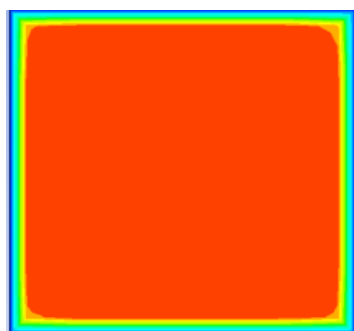
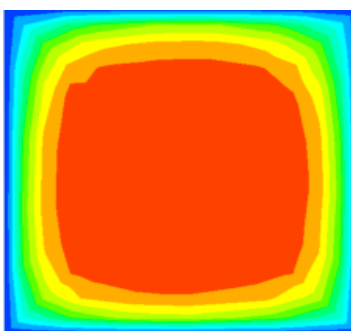
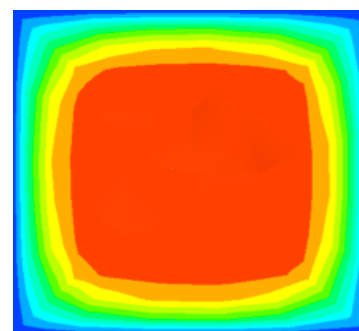
3.1. REYNOLDS NUMBER EFFECT

Figure 3 illustrates the temperature field in a rectangular channel for low Reynolds number flow, varying from 10 to 50. In this configuration, the initial temperature of the fluid is set at 313K, while the temperature of the channel walls is maintained at 278K. The temperature profiles are analyzed in three transverse planes corresponding respectively to $X^* = 0$ (channel inlet), $X^* = 0.5$ (mid-section), and $X^* = 1$ (channel outlet). It can be seen that at $X^* = 0$, for all Reynolds values studied, the fluid temperature is uniformly equal to 313K, which confirms the initial condition at the inlet. However, at $X^* = 0.5$ and $X^* = 1$, the results show a significant decrease in fluid temperature near the walls for $Re = 10$ and $Re = 20$, reaching approximately 293 K, which indicates efficient heat transfer in these cases. Conversely, for $Re = 30, 40$, and 50, the fluid temperature remains close to 313K throughout the channel, with negligible variation, reflecting weaker thermal interaction between the fluid and the walls. This behavior can be explained by the reduction in contact time between the fluid and the wall at higher Reynolds numbers, thus limiting heat transfer.

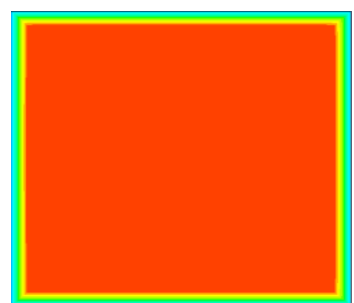
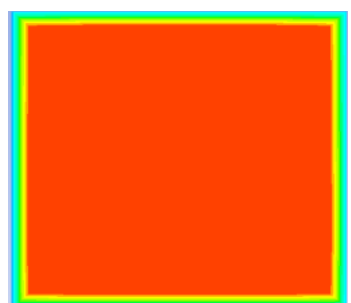
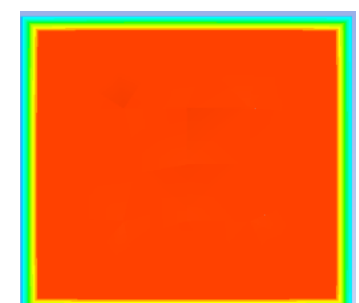
$Re= 10$



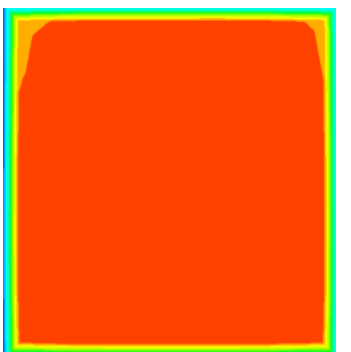
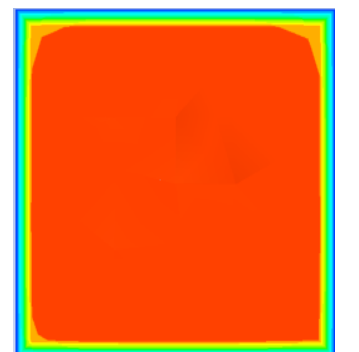
Re= 20

 $X^*=0$  $X^*=0.5$  $X^*=1$

Re= 30

 $X^*=0$  $X^*=0.5$  $X^*=1$

Re= 40

 $X^*=0$  $X^*=0.5$  $X^*=1$

Re= 50

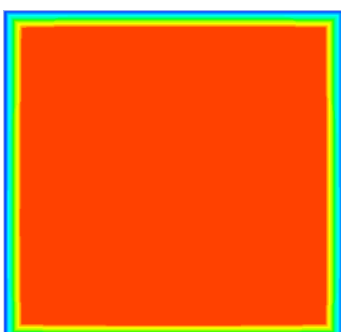
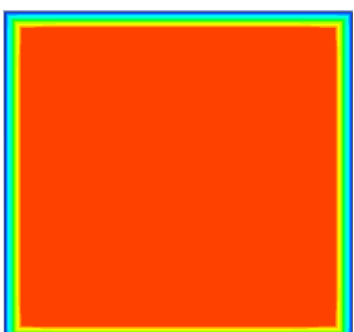


Figure 3: Temperature fields for different Re values, $T_f=60$

Figure 4 illustrates the evolution of fluid temperature along a channel equipped with baffles, for Reynolds numbers ranging from 10 to 50. We observe that the average fluid temperature is higher for $Re = 50$ and lower for $Re = 10$.

For $Re = 10$, the flow is slow, so the fluid remains in contact with the cold walls for longer, allowing for greater heat exchange. The prolonged residence time thus promotes cooling of the fluid, which explains why the temperature along the channel is lower in this case.

On the other hand, for $Re = 50$, the higher flow velocity reduces the contact time between the hot fluid and the cold wall. Although the Nusselt number is higher locally due to the intensification of convection, the overall heat transfer is less efficient, which explains why the fluid maintains a higher temperature throughout the channel.

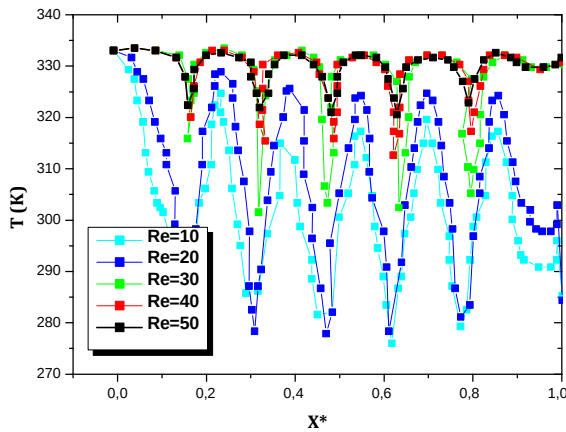


Figure. 4 Fluid temperature evolution for different Re values, $T_f=333$ K, $T_p=278$ K

Figures 5, 6, and 7 show the evolution of the fluid temperature at the inlet ($X^* = 0$), in the middle ($X^* = 0.5$), and at the outlet ($X^* = 1$) of the channel, respectively. We can see that at the inlet of the channel ($X^* = 0$), the maximum fluid temperature is identical for the three Reynolds numbers, i.e., 333 K, which corresponds to the initial temperature imposed at the inlet.

However, at the outlet of the channel ($X^* = 1$), there is a noticeable decrease in fluid temperature, particularly for low Reynolds numbers. This decrease is explained by the heat transfer between the hot fluid and the cold walls (maintained at 283 K) throughout the flow. More specifically, when the Reynolds number is low ($Re = 10$), the fluid velocity is reduced, which increases the contact time between the fluid and the wall. This prolonged residence time allows for more efficient heat exchange, resulting in significant cooling of the fluid at the outlet.

Thus, the drop in temperature at the channel outlet is directly related to the efficiency of heat transfer, which

depends on the flow regime and the interaction time between the fluid and the wall.

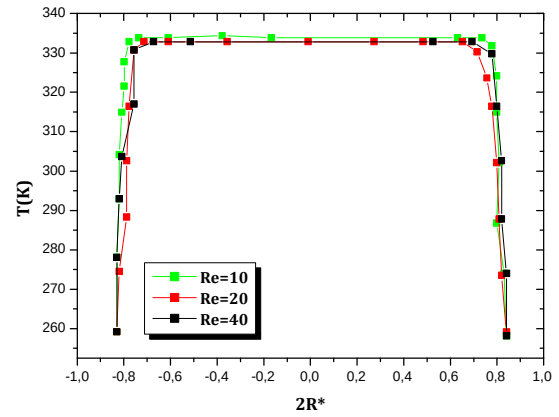


Figure 5 Fluid temperature evolution for different Re values, $T_f=333$ K, $T_p=278$ K, $X^*=0$

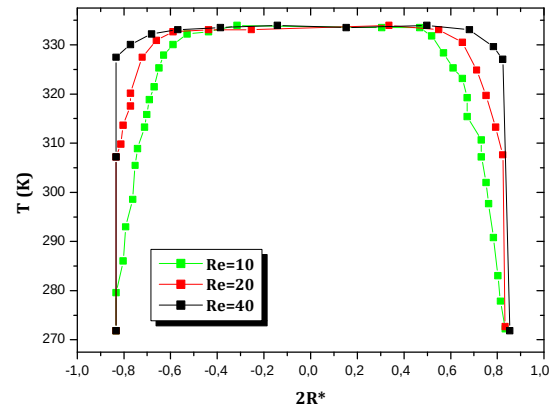


Figure 6 Fluid temperature evolution for different Re values, $T_f=333$ K, $T_p=278$ K, $X^*=0.5$

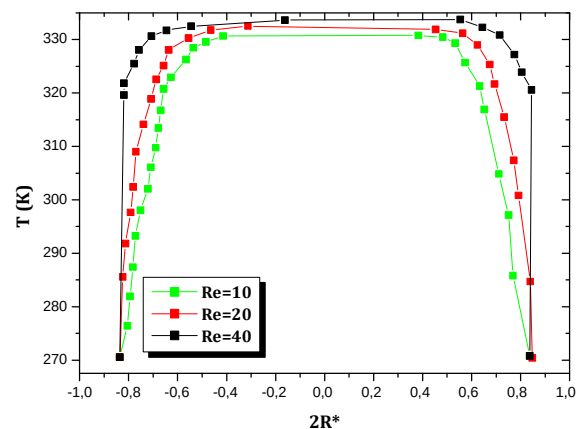


Figure 7 Fluid temperature evolution for different Re values, $T_f=333$ K, $T_p=278$ K, $X^*=1$

3.2. EFFECT OF GEOMETRY ON HEAT EXCHANGE

To analyze the impact of baffle shape on heat exchange, three geometric configurations were studied:

Case 1 corresponds to a geometry equipped with rectangular baffles, while cases 2 and 3 feature corrugated baffles of different shapes.

The detailed dimensions of these configurations are shown in Figure 8 and Table 2.

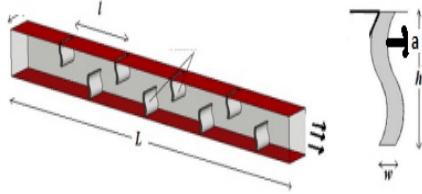


Figure 8. Geometry with corrugated baffles

Table 2. Dimensions of geometries

Cas	H/L	l/L	a/L	w/L
Cas N°1	0.05	0.156	0	0.005
Cas N°2	0.05	0.156	0.005	0.005
Cas N°3	0.05	0.156	0.01	0.005

Figure 9 shows the variation in the Nusselt number for different geometric configurations at a constant Reynolds number ($Re = 30$). It can be seen that the configuration in case 1, which uses rectangular baffles, provides higher heat transfer between the fluid and the channel walls compared to the configurations in cases 2 and 3, which use corrugated baffles.

This superior thermal performance can be attributed to the geometry with rectangular baffles, which generates more intense disturbances in the flow. In fact, baffles with straight edges cause flow separations, followed by more pronounced recirculation zones, which improves fluid mixing and reduces the thickness of the thermal boundary layer at the walls. This phenomenon therefore promotes more efficient convective heat exchange.

In comparison, corrugated baffles, although they also induce disturbances in the flow, tend to create a smoother and more gradual flow, with less boundary layer disruption. This limits the level of turbulence generated and therefore reduces the intensity of heat transfer in these configurations.

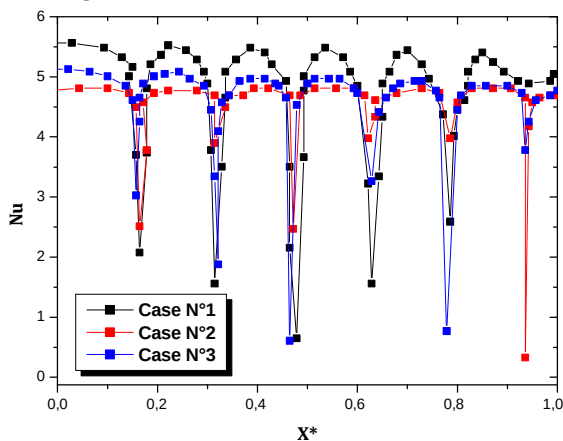


Figure 9. Nusselt number for different cases

3.3. COEFFICIENT OF PERFORMANCE (COP)

The coefficient of performance (COP) is expressed as:

$$COP = Nu / f$$

where Nu is the Nusselt number, which quantifies heat transfer by convection.

And f is the friction factor, which represents the pressure drop in the system, given by the following relationship:

$$f = \frac{2}{L/D_h} \cdot \frac{\Delta p}{\rho U^2}$$

Figure III.17 shows the evolution of the coefficient of performance for the three geometric configurations studied. It can be seen that the configuration in case No. 3 is the best compromise compared to the other two cases. This increase can be explained by the fact that case No. 3, which uses corrugated baffles, reduces pressure losses compared to case No. 1, which is characterized by rectangular baffles. The corrugated shape of the baffles in case no. 3 induces a more fluid flow with less abrupt disturbance, which limits the formation of recirculation zones behind the baffles. These recirculation zones, often at low speed, can lead to the formation of hot pockets, which are detrimental to heat transfer efficiency.

By reducing these undesirable effects, the configuration in case no. 3 optimizes the balance between heat transfer and pressure drop. This results in a 12% improvement in thermal efficiency compared to case no. 1, for $Re = 50$.

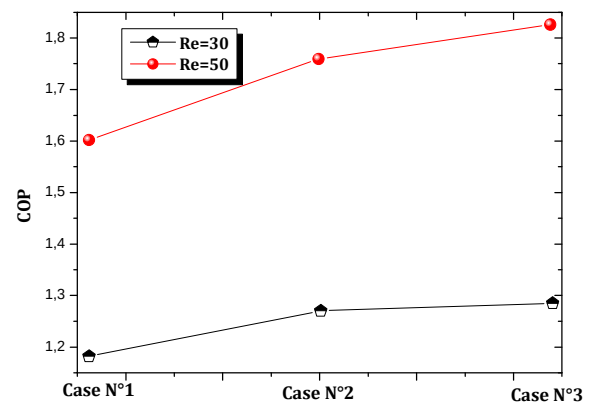


Figure 10. coefficient of performance (COP)

4. CONCLUSIONS

In this study, we evaluated the effect of several key parameters on the thermal and hydrodynamic performance of the exchanger. The results showed that increasing the fluid temperature ($T = 313 \text{ K}$, 333 K , and 353 K) significantly improves heat transfer. In fact, a higher temperature difference between the fluid and the wall intensifies heat exchange.

The influence of the Reynolds number was also clearly observed. Increasing Re promotes more turbulent flow, which disrupts the thermal boundary layer and leads to a significant increase in the heat transfer coefficient. The performance factor thus increased from 1.51 to 2.26 when the Reynolds number rose from 30 to 50,

confirming the major impact of this parameter on the overall efficiency of the system.

In addition, the study of the baffle design identified significant effects on the flow field distribution and the level of thermal mixing. The different geometric configurations analyzed showed contrasting behaviors, but the overall results highlighted that case No. 1 is the most efficient configuration, offering the best compromise between heat transfer intensification and pressure drop control.

Overall, this study highlights that the combination of a higher fluid temperature, a high Reynolds number, and optimized baffle geometry is a promising way to enhance the thermal performance of heat exchangers.

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