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COMPARATIVE STUDY OF THE THERMO-CONVECTIVE BEHAVIOR OF A TURBULENT FLOW IN A RECTANGULAR DUCT IN THE PRESENCE OF THREE PLANAR BAFFLES AND/OR CORRUGATED (WAVED)

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Abstract

This study numerically examines the behaviour of a two-dimensional turbulent flow of air in a rectangular pipe with three flat or corrugated baffles. A realizable k-ε turbulence model is used and the conservation equations are solved by the finite volume method using the SIMPLE algorithm. The study examined the dynamic and thermal behaviour of the flow by focusing on a presentation of several parameters, namely the turbulence intensity, the ratios of the friction coefficients and the Nusselt coefficients as well as the drag coefficients and thermal expansion, for all the geometry and for a Reynolds number varying from 5000 to 20000. The study showed that the presence of three baffles in a pipe governs a maximum turbulence intensity of the order of 206.49% to 110.72% for the small and the large inter-fin distance, which consequently guarantees a reinforcement of the heat exchange, of the order of 1.775 to 2.83 times compared with a smooth pipe, and on the other hand, an increase in friction with the walls of the order of 7 to 12 times. The study also revealed the existence of an optimum located between 0.14 and 0.16 m leading to the lowest friction ratio.

Keywords: turbulence baffles, forced convection, planar, corrugated.

Nomenclature

A_0, A_s	The constants of RNG turbulence model
C_f	Friction coefficient.
C_p	Specific heat at constant pressure, J/kg.K
C_1	Constant used in the standard k-ε mode
C_2	Constant used in the standard k-ε model
C_μ	Constant used in the standard k-ε model
E	Width of baffle, m
G_k	The generation of turbulence kinetic energy due to the mean velocity gradients, m^3/s^2
G_p	The generation of turbulence kinetic energy due to buoyancy
h	Baffle height, m
H	Height of air tunnel in pipe, m
k	Turbulent kinetic energy
L	Channel length, m
l_1,l_2,l_3	The three inter fin spacing, m
Nu	Averaged Nusselt number
Nu_x	Local Nusselt number
P	Pressure, Pa
Pi	Distance between two baffles, m
Re	Reynolds number
S_k, S_ε S_{ij},	Source term for k et ε
S_{ji}	Rate-of-strain tensor
$S\varphi$	Limit of source for the general variable
u, v	Velocities component, m/s
Y_M	The contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate

Greek Symbols

ε	dissipation rate of turbulence energy
ρ	Density of the air, kg/m
μ_l, μ_t μ_e	Molecular, turbulent and effective viscosity, Pa.s
v	Kinematics viscosity, pl
Φ	Stand for the reliant variables u, v, k and ε
σ_k	model constant for the k-equation
σ_ε	model constant for the ε-equation
Δp	Pressure losses
∇^2	Laplacian operator
∇	Divergent operator
$\Omega i,j$	The mean rate-of-rotation tensor
η	The improvement factor thermal performance

Abbreviations

in, o*ut*	Inlet and outlet of the simulated test section
t	Turbulent
w	Wall
f	Fluid
s	Solid

1. Introduction

The study of forced convection in complex geometry receives considerable attention due to its importance in many engineering applications. Many researchers have studied the hydrodynamic characteristics of flow and improved heat transfer performance, especially in heat exchangers and thermo-solar converters, Saim et al. [1- 3].

The technique of increasing heat exchange is based on the use of tabulators also known as vortex generators, are baffles, generally attached to the heated surface so as to provide surface heat transfer And to promote turbulence, for which several forms and orientations of baffles have been the subject of several investigations.

Considerable works has been done, in recent years, on the investigations of the flow and heat transfer processes in these geometries, include the numerical study by Jedsadaratanachai et al. [4] treats a periodic laminar flow and heat transfer characteristics in a channel inclined at 30° and equipped with baffles inserted at intervals. In order to generate rotating vortices, the authors placed repeatedly inclined deflectors on the lower and upper walls of the channel. The study showed that the longitudinal eddies created effectively increased the rate of heat transfer in the canal. The heat transfer enhancement order is about 1 to 9.2 times the smooth channel. However, this improvement in heat transfer is associated with the pressure loss ranging from 1 to 21.5 times that of the smooth channel.

Roetzeli [5] carried out an experimental study of turbulent flows to treat the role of baffles on heat transfer in a tubular heat exchanger. The impact of the distance between the baffles and their locations relative to the shell on thermal performance was also analysed.

An experimental and numerical study was investigated by Parkpoom et al. [6]. In this experiment, the baffles are placed in zigzag shape (shaped baffle Z) aligned in series on a heated top wall, similar to the absorber plate for a solar panel to air. The effect of the height and spacing are examined to find the optimal thermal performance for a Reynolds number ranging from 4400 to 20400.

Shivani et al. [7] have shown that the flow is significantly modified in the presence of a baffle inside the channel. The baffle used in this work is attached to the lower wall of the channel and blocks nearly 15% of the cross section, although it is small, experimental results published a considerable increase in the rate of production and dissipation energy. Turbulence has been improved more efficiently at a distance close to the baffle and precisely just downstream of it. The turbulence in this area was several times greater than without baffle.

Another experimental study in a rectangular channel with perforated baffles was investigated by Rajendra et al. [8] This article has shown that the Nusselt number for solid baffles is 73.7 to 82.7% higher than Of the smooth pipe, whereas for perforated baffles it varies from 60.6 to 62.9% and from 45.0 to 49.7%; This rate decreases with the increase in the perforation diameter of these baffles. The coefficient of friction for solid baffles is 9 to 11 times higher than that of the smooth pipe, this coefficient has shown a significant decrease for the perforated baffles and especially by increasing the diameter of these perforations.

Promvonge et al. [9] carried out a numerical study of a turbulent flow and the heat transfer behaviour in a square channel placed diagonally at 30 ° equipped with a finned strip for a Re = 4000 to 20 000. The vortex created by the insertion of the fin strip and there contributes to a drastic increase in heat transfer. The increase in the Nusselt number is about 150-650% *BR* = 0.1 to 0.3, (*BR* = *b* / *H*) .However, the fin strip causes a pressure loss from 2 to 55 times that of the smooth pipe. The factor for improving the thermal performance in the case of the insertion of the fin strip is much higher than the unit and its top is approximately 1.95 located at the lower Reynolds number indicating better performance heat relative to that of the smooth pipe.

The objective of this work is to study the dynamic and thermal behaviour of the airflow in the presence of a wave-shaped baffle, it is rarely used in the literature, the choice of this form is due increasing the exchange surface while maintaining the same height of the baffle, for this turbulent flow within an rectangular channel containing three flat or corrugated baffles is studied for different spacing and Reynolds numbers. The analysis of the variation of the Nusselt and coefficients of friction, trolling, and thermal expansion was addressed to numerically quantify the thermal performance and pressure drop in the channel.

2. Mathematical Formulation

2.1. Position of the problem

The geometry of the problem is shown in Figs. 1(a), (b) and (c). It is a rectangular pipe provided with three baffles. Two different forms were analysed, flat shape (Figs.1 (a) and (b)) and a wavy shape (Fig. 1 (c)). This conduct is crossed by turbulent airflow satisfying the following assumptions: thermo-physical properties of the assumed constant fluid; uniform velocity profile at the entrance; heat transfer by radiation is negligible; the flow is assumed to be steady.

The third fin has been installed in the middle of the lower wall, while the other two are placed, upstream and downstream, on the top wall. The three fins spacing are respectively $L_1=S/2$ =0.071 m, $L_2 = S$ =0.142 m and $L_3=3S/2$=0.213 m were used, identified with respect to the central baffle.

The flow of a stationary incompressible fluid and the heat transfer in the computational domain is governed by the conservation equations, namely the continuity equations, the Navier-Stokes equations and the energy equation, Write in the following form:

$$\Delta \vec{V} = 0 \tag{1}$$

$$\rho\left(\vec{V}.\nabla.\vec{V}\right) = -\nabla P + \mu_f \nabla^2 \vec{V} \tag{2}$$

$$\rho C_p\left(\vec{V}.\nabla T\right) = K_f \nabla^2 T \tag{3}$$

The governing equations based on the realizable k-ε model [10] used to model the turbulence, are solved by the finite volume method [11]. The terms of velocity and pressure equations of motion are solved by the SIMPLE algorithm [12].

$$\frac{\partial}{\partial x_j}\left(\rho.k.u_j\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_M + S_k \tag{4}$$

$$\frac{\partial}{\partial x_j}\left(\rho.\varepsilon.u_j\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \rho C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k+\sqrt{v\varepsilon}} + C_{1\varepsilon}\frac{\varepsilon}{k} C_{3\varepsilon} G_b + S_\varepsilon \tag{5}$$

where:

$C_1 = max\left[0.43, \frac{\eta}{\eta+5}\right]$, $\eta = S\frac{k}{\varepsilon}$, $C_{3\varepsilon} = tanh\left|\frac{v}{u}\right|$

where v and u are respectively the parallel and perpendicular component of the velocity of the flow to the gravitational vector.

The turbulent viscosity is given by:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{6}$$

The difference with the standard model is in the term of C_μ which is given by:

$$C_\mu = \frac{1}{A_0 + A_s \frac{kU^*}{\varepsilon}} \tag{7}$$

where: $U^* = \sqrt{S_{ij}S_{ij} + \tilde{\Omega}_{ij}\tilde{\Omega}_{ij}}$ and $\tilde{\Omega}_{ij} = \Omega_{ij} - 2\varepsilon_{ij}\omega_k$,

$\Omega_{ij} = \bar{\Omega}_{ij} - \varepsilon_{ij}\omega_k$

where: $\bar{\Omega}_{ij}$ is the rotation coefficient tensor obtained from the angular velocity ω_K.

The model constants A_0 and A_s are given by: $A_0 = 4.04$, $A_s = \sqrt{6}\cos\phi$

where: $\phi = \frac{1}{3}cos^{-1}(\sqrt{6}W)$, $W = \frac{S_{ij}S_{jk}S_{ki}}{\tilde{S}}$, $\tilde{S} = \sqrt{S_{ij}S_{ij}}$, $S_{ij} = \frac{1}{2}(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j})$

The constants of the model are given by:

$C_{1\varepsilon} = 1.4$, $C_{2\varepsilon} = 1{,}9$, $\sigma_k = 1$, $\sigma_\varepsilon = 1{,}2$

The relationship between the Reynolds number and the friction coefficient can be expressed as:

$$\left(f.Re^3\right)_0 = \left(f.Re^3\right) \tag{8}$$

$$Re_0 = Re\left(f/f_0\right)^{1/3} \tag{9}$$

The improvement factor η thermal performance is defined as follows:

$$\eta = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(\frac{Nu}{Nu_0}\right)\left(\frac{f}{f_0}\right)^{-1/3} \tag{10}$$

2.2. Boundary Condition and Grid Sensitivity

The dynamic and thermal behavior of the air to the three positions of the baffles was analyzed for Reynolds numbers equal to 5000, 10000, 15000 and 20000, for this, a uniform speed is applied as a boundary condition to the hydraulic input of the computational domain. And as a condition for the thermal limit, a constant temperature of T_w=102 ° C (375 K) was applied on the two horizontal walls of the computational domain. The temperature of the fluid used was set at T_{in}=27 ° C (300 K) to the inlet side. The conditions for hydraulic and thermal limits are chosen in accordance with that section of Nasiruddin et al. [13].

A structured, variable grid with quadrilateral elements, highly concentrated in the vicinity of the walls and baffles was generated in the Gambit preprocessor. To test their independence on the results, a series of test results obtained for the two components of velocity and stream function was performed. Finally and after reaching a difference of results, between two grids, less than 1%, the mesh grid with 195×82 nodes was chosen and that will allow obtaining solutions with a reasonable error. Subsequently, a Fluent calculation code (V. 6.3.26), based on the finite volume method, is used to perform the numerical simulations.

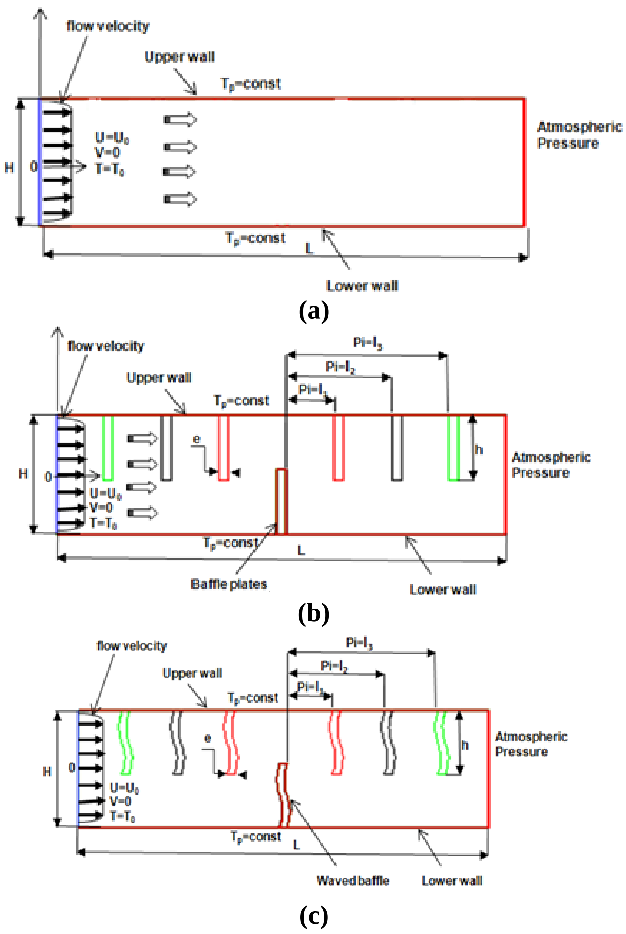


Fig. 1. Geometry of the system studied (a) Canal smooth wall, (b) Channel with flat baffles (c) Channel with wavy baffles.

3. Results and Discussion

3.1. Validation

These numerical results on the heat transfer and friction characteristics in a smooth-walled channel are first validated in terms of Nusselt number and coefficient of friction. The Nusselt number and the coefficient of friction obtained in a smooth walled channel are compared, respectively, with respect to correlation Dittus-Boelter and Blasius [14, 15] for turbulent flow in pipes.

- Dittus-Boelter Correlation [14]:

$$Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \quad for\ \mathrm{Re} \geq 10000 \tag{11}$$

- Blasius Correlation [15]:

$$Cf = 0.316\ \mathrm{Re}^{-0.25} \tag{12}$$

The results shown in Figs. 2 (a) and (b) agree reasonably well for the two friction coefficient of correlation and the correlation of Blasius Nusselt number of Dittus-Boelter.

3.2. Turbulence Intensity

For a Reynolds number equal to 5000, the contours of the intensity of the turbulence in the direction of flow for the three different spacing used in a pipe with three disruptive of different shapes are shown in Fig. 3.

The turbulence intensity increases from the first chicane and is growing more and more, approaching towards the exit of the pipe, where the highest values are observed.

Maximum turbulence of the rate based on the average speed achieved in the pipe with three baffles: 206.49% for the small distance where $L_1 = S/2$,124.71% to the average distance where $L_2=S$ and 110.72% for $L_3=3S/2$, the greater the distance reduces the turbulence increases, in other words there is an inverse proportionality between the elevation of the turbulence and spacing inter fin.

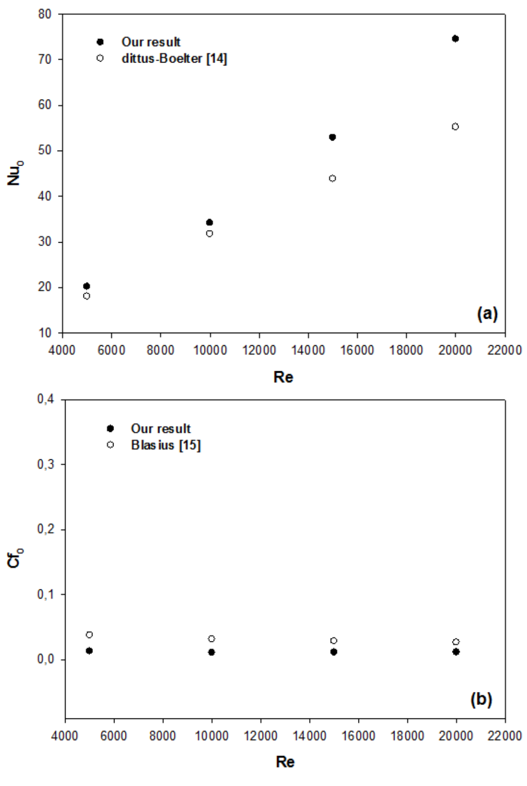
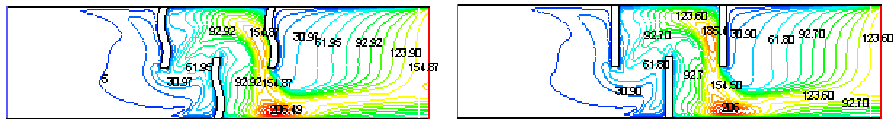
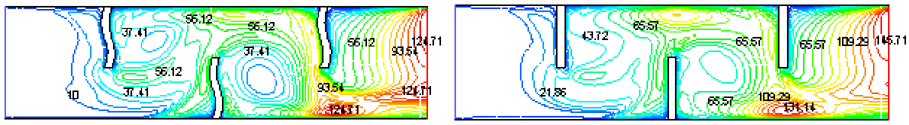


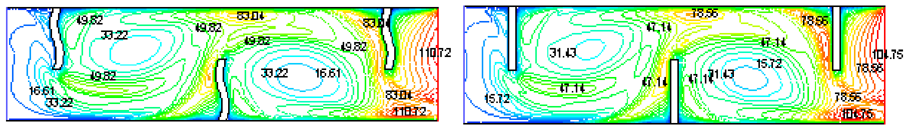
Fig. 2. Comparison with different correlations:
(a) Number of Nusselt, (b) Coefficient of friction for a smooth wall channel.



(a)



(b)



(c)

Fig. 3. Turbulence intensity [%] for both flat and waved forms and for different spacing (a) L_1=S/2, (b) L_2=S, (c) L_3=3S/2.

These turbulence intensity contours show the role of baffles in flow disturbance which promote turbulence and consequently increase the heat transfer capacities between the solid and adjacent fluid particles and between the fluid particles themselves.

3.3. Friction coefficient

It should be noted that for a Reynolds ranging from 5000 to 20000, for the two form of baffle (flat or corrugated) and for the different locations, the ratio of the coefficient of friction shows a decrease with the increase of the number of Reynolds. The presence of three baffles in a pipe leads to a considerable increase in the friction with the walls; the improvement is of the order seven to twelve times with respect to a smooth pipe.

Analysis of friction according to the distances between fins for both forms, shows that it is the intermediate case L_2 corresponding to the lowest friction Fig. 4, which suggests the existence of an optimum located between 0.14 and 0.16 as it is presented and reassured in Fig. 5, which shows the evolution of the ratio of the friction coefficients according to the distance between fins for different Reynolds numbers.

Figure 5 confirms that there is an inverse proportionality between the elevation of the friction coefficient ratio and the Reynolds number for both forms and different distances between the baffles. Note also that the values of the higher friction ratios are recorded for smaller distances, then decreases more with increasing the distance between fin until reaching the minimum (optimum) in the vicinity of a distance L_2 = 0.14 m and just after, reports began to increase with increasing distance.

To quantify the pressure losses for both forms, it is necessary not to take into account the viscous friction through the walls of the lead and baffles, however it is interesting to involve the concept of drag coefficient (Fig. 6 and Table 1), which is related to the form and which is due to the pressure difference between the upstream and downstream of the obstacle.

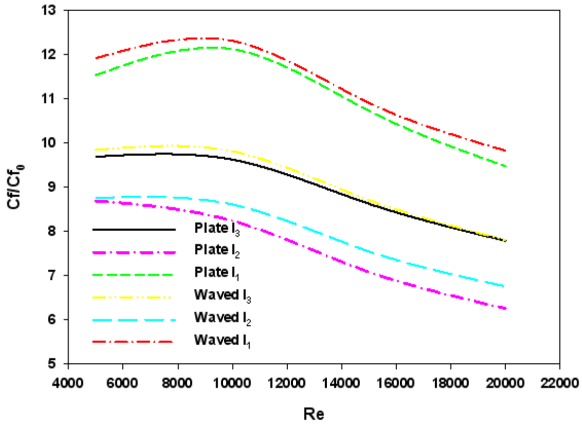


Fig. 4. Means friction coefficients report depending on the Reynolds number.

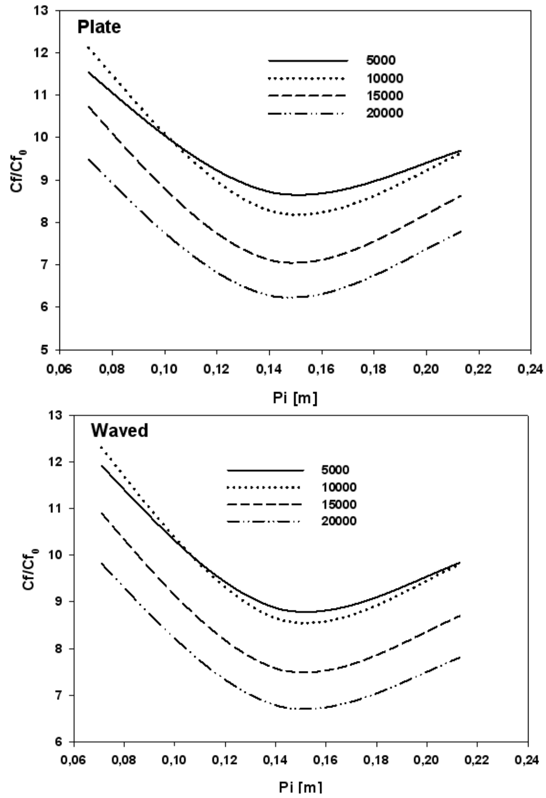


Fig. 5. Report of the coefficients of friction means a function of the spacing between vanes, for different Reynolds numbers.

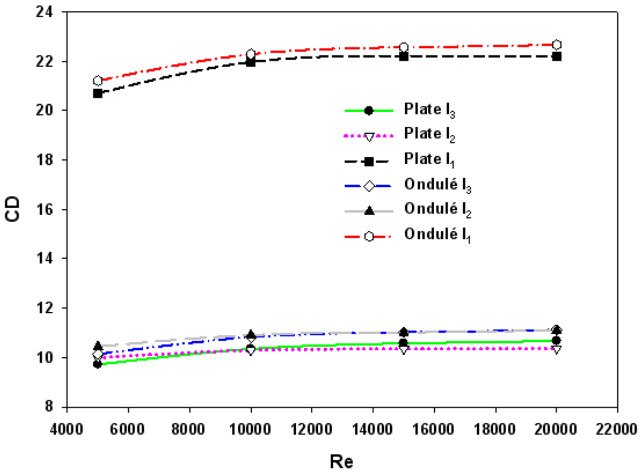


Fig. 6. Mean Drag coefficients as a function of the Reynolds number.

Table1. The average coefficients of friction and dragged to different spacing.

	$\overline{Cf}$		
	L_1	L_2	L_3
Plate	0.103	0.087	0.126
waved	0.104	0.092	0.13
	CD		
Plate	10.329	10.247	21.777
waved	10.777	10.868	22.184

For all the cases studied and presented in Table 1, the results of friction and dragged show values decreases to the planar shape by providing the wavy form.

3.4. Heat transfer

The report of the Nusselt number, Nu / Nu_0, defined as a Nusselt number ratio increased compared to the smooth channel Nusselt number plotted against the value of the Reynolds number, is shown in Fig.7 and Table 2. This report of Nusselt number tends to decrease with increasing Reynolds number in the range 5000-20000 for all cases treated.

The presence of three corrugated or flat baffles in a drive ensures enhanced exchange heat, of the order of 1.775 to 2.83 times compared to smooth channel. This is explained by the extension of the trajectories of the fluid particles created by the obstacles, which promotes a better heat exchange by convection and consequently a significant improvement in the thermal efficiency.

There is still an inverse proportionality between the relative elevation of the Nusselt number and the Reynolds for different inter-vane distances.

Low ratios of Nusselt are observed for the case of medium distance (L_2=S), while the larger ratios are those for smaller distance.

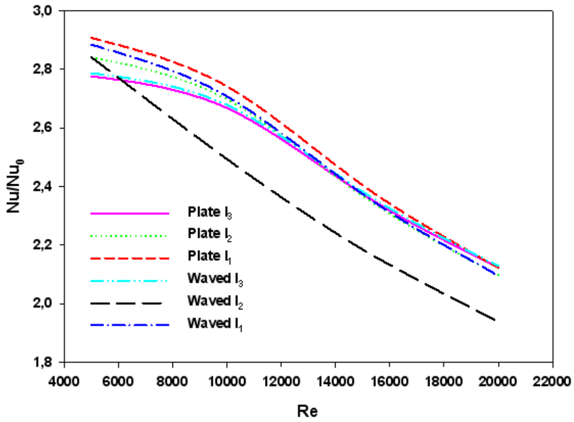


Fig. 7. Report the average Nusselt number as a function of the Reynolds number for different distances between fins.

Table 2. Average Nusselt coefficients for different spacing and shapes.

	Nu/Nu₀		
	L_1	L_2	L_3
Plate	2.48	2.50	2.54
Waved	2.49	2.32	2.51

The inclusion of the three baffles in the channel induces a significant improvement in heat transfer but causes additional pressure drop. The thermal improvement factor (η) is therefore a duality between the control of thermal performance and hydraulic performance.

Figure 8 and Table 3 illustrate the evolution of the thermal improvement factor (η) as a function of the Reynolds number for all the treated cases, and shows that the factor decreases with the increase of the Reynolds number values. It should also be noted that the increase in the distance between the fins contributes to an increase in the thermal expansion factor. The best case is that of L_1.

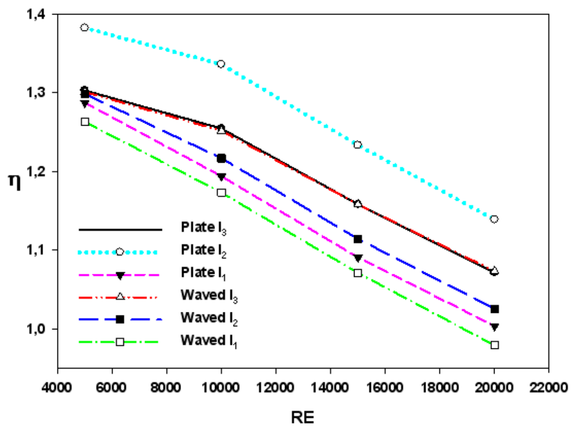


Fig. 8. Thermal enhancement factor as a function of the Reynolds number for the three distances between fins.

Table 3. Thermal enhancement factors for different spacing and shapes.

	η		
	L_1	L_2	L_3
Plate	1.196	1.272	1.143
Waved	1.196	1.163	1.121

4. Conclusion

This article aims to present the behavior of air flow around baffles flat or wavy shapes. The Fluent CFD code was adopted to simulate the steady flow of a Newtonian fluid, incompressible and turbulent.

The effect of three chicanes shapes (flat or corrugated), purely vertical, on improving Nusselt number and pressure loss was studied for different Reynolds number. This work allowed us to draw from rich observations for different Reynolds number.

- The evolution of the friction coefficient profiles along both walls and especially when it is weak interlayer distance fin.
- Insert three baffles in the channel leading to significant friction values on the Reynolds range from 5000 to 20000.
- Significant improvement on the heat exchange along the two walls and for different Reynolds numbers.
- In case the shortest distance between fins (L_1), dynamic configurations and heat flow are almost identical for the two forms of baffles.
- Improving η to say in the case of flat baffles and from the intermediate distance between blades.
- In both forms, the most favorable and most successful case is the shortest distance.

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