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Research Paper

Numerical Analysis of the Baffles Inclination on Fluid Behavior in a Shell and Tube Heat Exchanger

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Abstract. The thermo-hydraulic performances of the shell-and-tube heat exchangers with different baffles inclination angle α =10°, α =20°, and α = 40° are investigated. The numerical analysis has been evaluated using ANSYS Fluent with the finite volume method for Reynolds number varying between 24000 and 27000. In all heat exchangers, the characteristics studied are the velocity, the temperature in the shell, the heat transfer coefficient, the pressure. The results showed small dead zones for the baffles inclination angle of 40°. The results showed that the temperature increases by 3.4 K, the heat transfer coefficient decreased by 0.983 %, the pressure drop decreased by 0.992 %, the overall performance factor decreased by 0.83 % when the baffles inclination angle α is increased from 10° to 40°.

Keywords: Shell and tube heat exchanger, CFD, Pressure drop, Baffle design.

1. Introduction

The presence of obstacles in the channels increases the flow energy performance and gives a better heat transfer coefficient. Over 35-40% of the heat exchangers are shell-and-tubes type [1] because of a higher temperature operating and their robust mechanical construction.

The use of the segmental baffles improves heat transfer and increases turbulence [2], however, this type has major disadvantages [3, 4]: very high-pressure drop, low heat transfer, stagnation of the fluid in the dead zones, strong vibration and higher pumping power. As a result, it is necessary to use new types of baffles to improve heat transfer. Anas et al. [5], a comparison between three types of heat exchangers, the results show that the use of helical baffles gives a higher thermo-hydraulic performance, high pressure drop by the baffles lattice holes. Gabriel et al. [6], a numerical analysis of a shell-and-tube heat exchanger with segmental baffles by two software packages, HTRI and Fluent 15.0, the leakage rate decrease the fluid temperature and pressure drop by 8 K and 40%. Cong et al. [7], performed a numerical analysis on six types of baffles, helical, elliptical, and segmental, the results show that the form 20° has the optimum characteristics. Sepehr et al. [8], an optimization study of a shell-and-tube heat exchanger to maximize efficiency and minimize total cost, by varying the design parameters. Dogan. [9], a thermo-economic analysis to give the optimal ratio between the baffles and show that, the ratio and the shell diameter as affected by the different values of the geometric parameters. Kunal et al. [10], a comparative study between two heat exchangers showed that the pressure drop is less than about 8.3%, and the overall performance higher by 4.52% in the heat exchanger with inclined baffles. Sunilkumar et al. [11], a comparison for the angles 15°, 25°, 35°, 45°, 55° in heat exchangers with helical baffles, the results show a decrease of heat transfer coefficient and pressure drop and increase in the overall performance factor with increasing helix angle. Youcef et al. [12] investigated by numerical simulations the performances of the heat exchangers with new shape baffles, their results showed that the wing baffles losses pressure drop and improve the heat transfer coefficient. Houari [13], analysis the effects of the waviness angle baffles 0°, 22.5°, 45° on convection and pressure losses in a heat exchanger, the results showed that the performance factor increased by 1.27 to 1.53 when the waviness angle increased from 0° to 45°. Ahmed et al. [14,15] a comparative study for giving the effect of baffles in the shell and tubes heat exchanger, the results show that the heat transfers and pressure drop increased by 1.86% and 21.67%. Cong et al. [16], have studied the impact of folded helical baffles, the results show that, under the same flow, the overall performance of these baffles is higher than those of segmental baffles. Devvrat et al. [17], carried a numerical analysis of helical angle between 0°, and 20°, the results conclude that the temperature, the velocity and the heat transfer coefficient increase with the angle of the helical baffle. Kunal et al. [18], a comparative study between two heat exchangers, the results showed that the pressure drop is less than about 8.3%, and the overall performance higher by 4.52% in the heat exchanger with inclined baffles.

The present study explores the influence of baffles inclination angle on the phenomenon of thermal transfer, effects of inclination angle 10°, 20°, 40° for Reynolds number varying from 24000 to 27000 in a shell-and-tube heat exchanger on heat transfer coefficient and pressure drop are highlighted.

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2. Mathematical Formulation

2.1 Geometry of the problem

The schematic diagram and geometry of the problem studied are shown in Figs. 1 and 2, six baffles placed on alternate orientations in the shell of the heat exchangers, the model created by deferent baffles inclination by α =10°, α =20° and α= 40°.

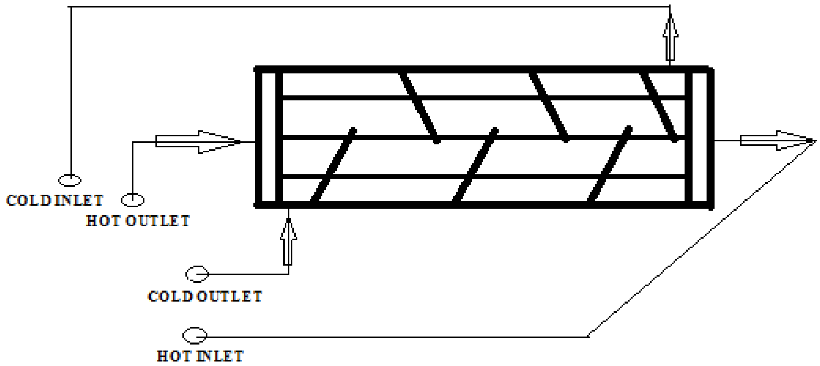


Fig. 1. Schematic diagram.

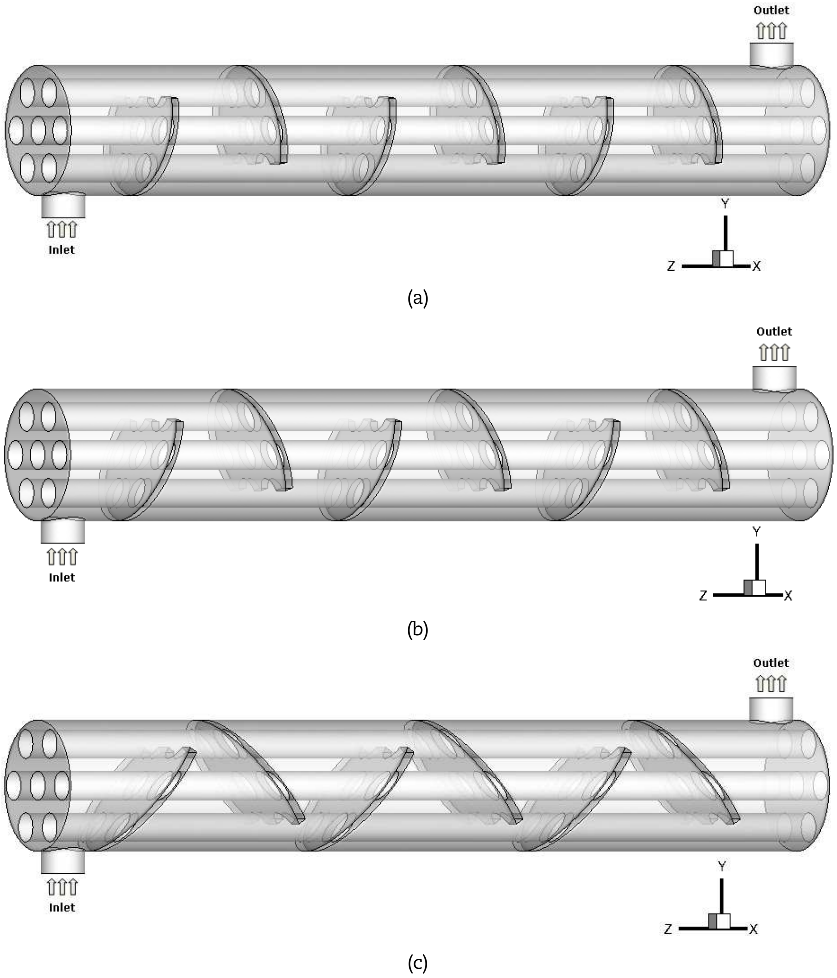


Fig. 2. Geometry of the problem (a) 10°, (b) 20°, (c) 40°.

Table 1. Geometric parameters

Parameter	Values
Shell size, D_s	90 mm
Tube outer diameter, d_o	20 mm
Tube bundle geometry and pitch	triangular, 30 mm
Number of tubes, N_t	7
Heat exchanger length, L	600 mm
Shell side inlet temperature, T	300 K
Baffle cut, B_c	36%
Number of baffles, N_b	6
Baffles inclination α	20°, 30°,40°

2.2 Governing equations

The governing equations of the considered problem can be expressed as follows:

Continuity:

$$\frac{\partial(\rho.u_i)}{\partial x_i}=0 \tag{1}$$

Momentum:

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

Energy:

$$\frac{\partial(\rho.u_iT)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\left(\frac{\mu}{\Pr}-\frac{\mu_t}{\Pr_t}\right)\frac{\partial T}{\partial x_j}\right) \tag{3}$$

Turbulence kinetic energy k:

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

Energy dissipation ε:

$$\rho u_j\frac{\partial \varepsilon}{\partial x_j}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}G_k-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{5}$$

Turbulent viscosity:

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

The turbulence production

$$G_k=-\rho\overline{u_i'u_i'}\frac{\partial u_j}{\partial x_i} \tag{7}$$

The model constants have the following values: $C_{1\varepsilon}$=1.44, $C_{2\varepsilon}$=1.92, C_μ=0.09, σ_k=1.0, σ_ε=1.3, Pr_t=0.09.

The overall performance factor:

$$\eta=\left.\frac{h}{h_0}\right|_{pp}=\left.\frac{Nu}{Nu_0}\right|_{pp}=\left(\frac{Nu}{Nu_0}\right)\left(\frac{f}{f_0}\right)^{-1/3} \tag{8}$$

The friction factor:

$$f_s=e^{(0{,}576-0{,}19\ln\mathrm{Re})} \tag{9}$$

The local Nusselt number:

$$Nu(x)=\frac{h(x)D_h}{\lambda_f} \tag{10}$$

The Nusselt number of base case Incropera and al [19]:

$$Nu_0=\frac{\left(\frac{f}{8}\right)\mathrm{Re}\,D\,\Pr}{1.07+12.7\left(\frac{f}{8}\right)^{1/2}\left(\Pr^{2/3}-1\right)} \tag{11}$$

The friction factor of the base case [19]:

$$f_0=(0.79\ln\mathrm{Re}-1.64)^{-2} \tag{12}$$

2.3 Boundary conditions

A uniform velocity has applied to the inlet of the computation domain and a constant temperature of T_w = 450 K applied to the walls of the tubes. It took the temperature of the fluid used equal to T_{in} = 300 K at the inlet of the fluid [20].

Table 2. Validation of the mesh at y=0.045 m.

Elements	U_{max}(m/s)	V_{max}(m/s)	W_{max}(m/s)	T_{max}(K)
352708	0,0002	1,922	0,00003	339,4159
666871	0,0001	1,9143	0,00002	338,5478
1021159	0,00015	1,907	0,000024	338,2547

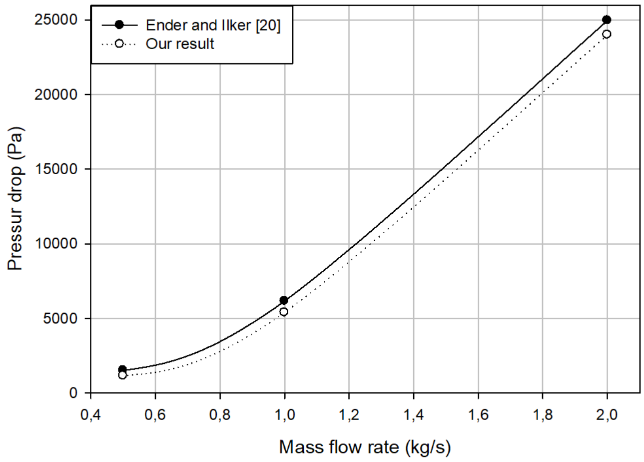


Fig. 3. Pressure drop in the shell.

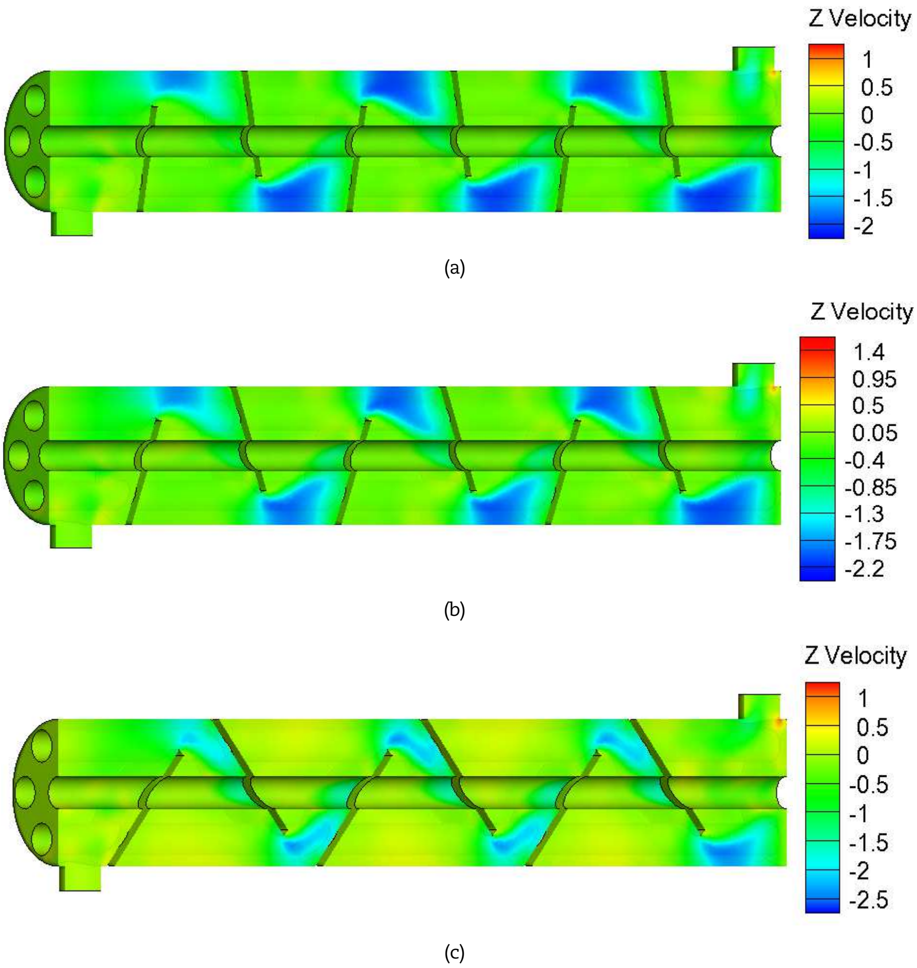


Fig. 4. Velocity contour in (m/s) (a) 10°, (b) 20°, (c) 40°.

3. Results and Discussion

3.1. Grid sensitivity

To ensure that the results are independent of mesh, several tests of mesh in the shell was generated in section y = 0.045 m presented in the Table 2. The analysis of the results shows that the choice of (1021159) cell suffices to get the independence of the parameters studied.

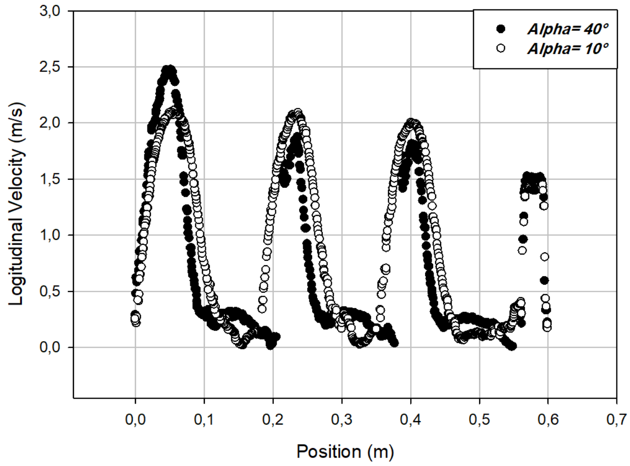
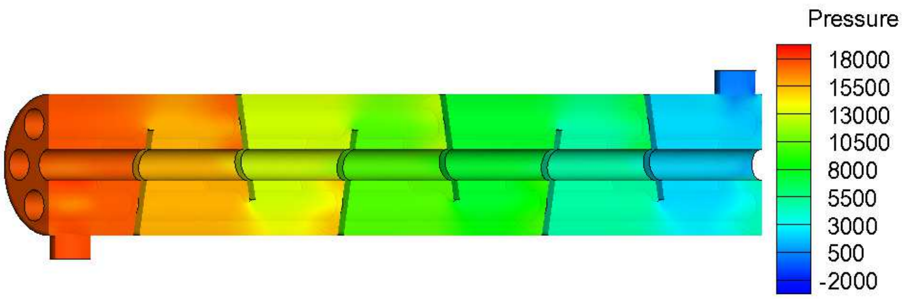
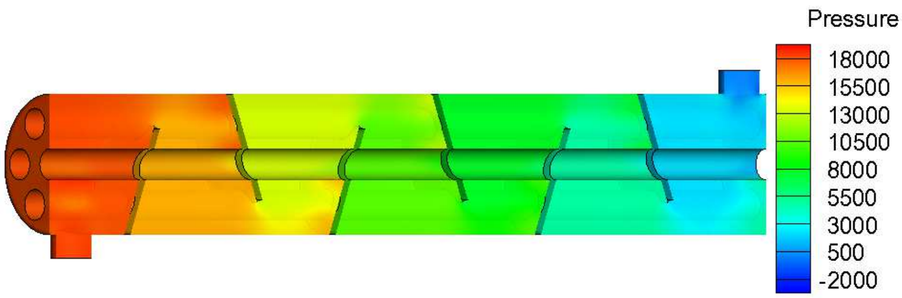


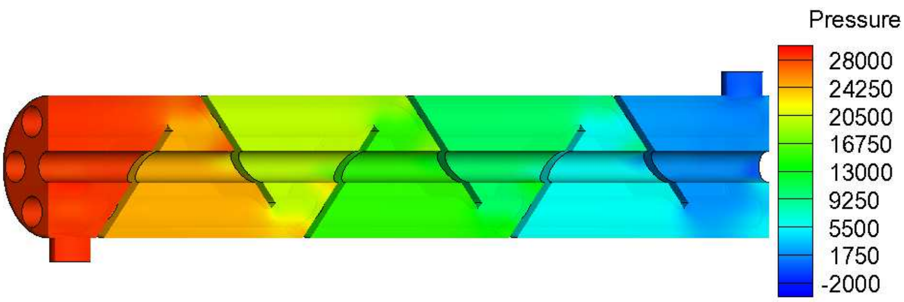
Fig. 5. Longitudinal velocity profile at y = 0.04 for α =10° and α =40°.



(a)



(b)



(c)

Fig. 6. Dynamic pressure contour in (Pa) at the symmetrical plane (a) 10°, (b) 20°, (c) 40°.

3.2. Model validation

In Fig. 2, the heat exchanger with segmental baffles α =0° compared with the data available [20] in order of pressure drop. The deviation between the two results was found to be about 4 %. From the grid independence and this validation that the model of 1021159 elements gives a good prediction for the characteristics of the heat exchanger.

3.3. Longitudinal velocity profile

In Fig. 4, the velocity contours given in the shell for the three baffles inclination angle α. The fluid hits the baffles and changed the direction, so the space of the grille behind the baffle not used effectively for the fluid. The maximum velocity for α =10°, α =20° and α =40°, is 2.28 m/s, 2.39 m/s and 3.19 m/s. We find that baffles inclination with α=40° provides a faster flow caused by the great baffles inclination, so there is an increase in velocity with increasing baffles angle at the outlet.

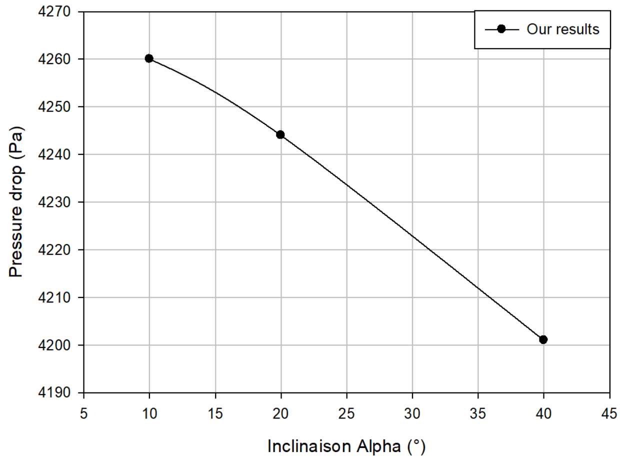


Fig. 7. Pressure drop in the shell

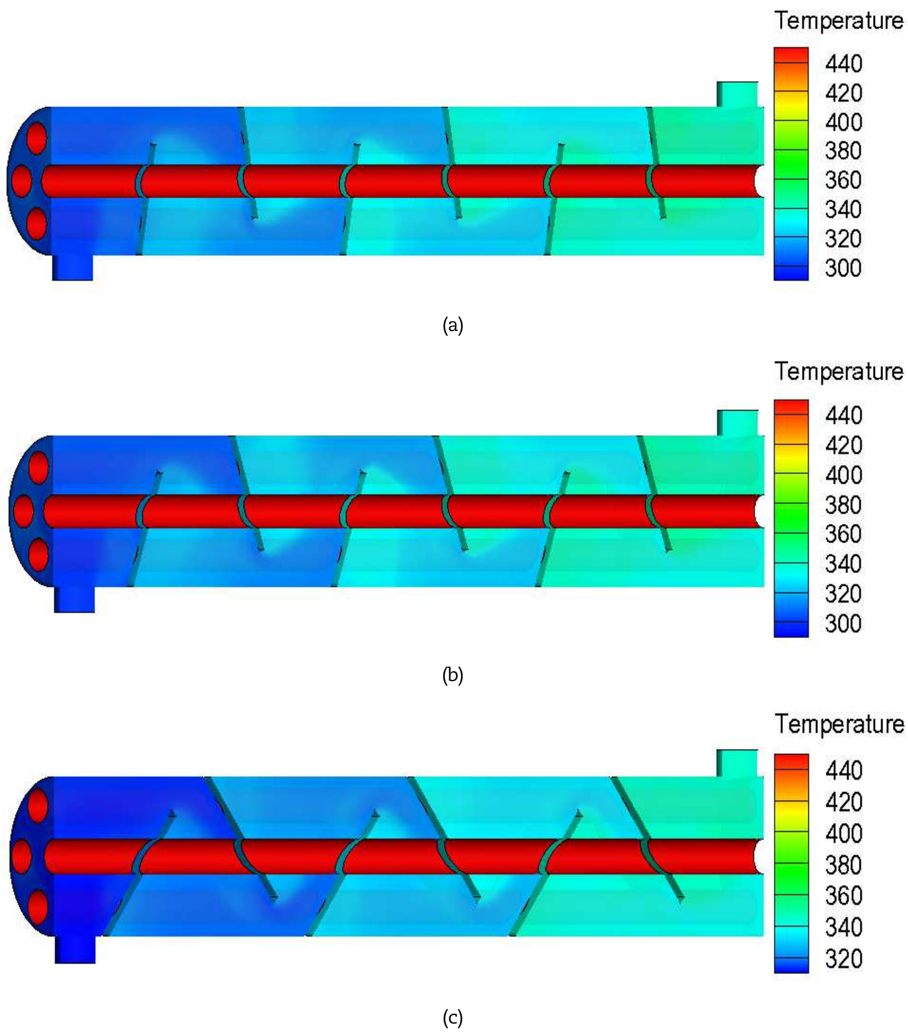


Fig. 8. Temperature contour in (K) (a) α = 10°, (b) α = 20°, (c) α = 40°.

In Fig. 5, the distribution of the velocity profile in the heat exchangers at y = 0.04 m for the baffles inclination of α =10° and α =40° are presented. The velocity increase from the inlet to the outlet for both cases. For the angle of α =10° the fluid velocity reaches 2 m/s after the second baffle, then 2.1 m/s after the fourth baffle, then 2.15 m/s at the outlet. For the baffles inclination angle of 40° the fluid velocity reaches 1.75 m/s after the second baffle, then 1.82 m/s after the fourth baffle, then 2.5 m/s at the outlet. The effects of changing the baffles inclination angle α with the same baffle free segment of 36 % result that the velocity of α=40° inclination are faster than that of α =10° inclination which just happens in the range of 0-0.1 m because a high recirculation in the baffle duct, while the velocity of α =10° inclination is all faster in the rest range of 0.1-0.6 m because the outlet of the shell in this region, also the heat transfer is the smallest value in this region.

The increasing of the baffles inclination angle α with the same baffle free segment leads to intensified performances and increase the residence time of the fluid in the shell and leading to upper gradients of temperature. Comparing these two profiles note that the variation of the velocity with the baffles inclination angle of 10 ° is higher in the range of 0.1-0.6 m and elevated the pressure drop along with the shell for this case.

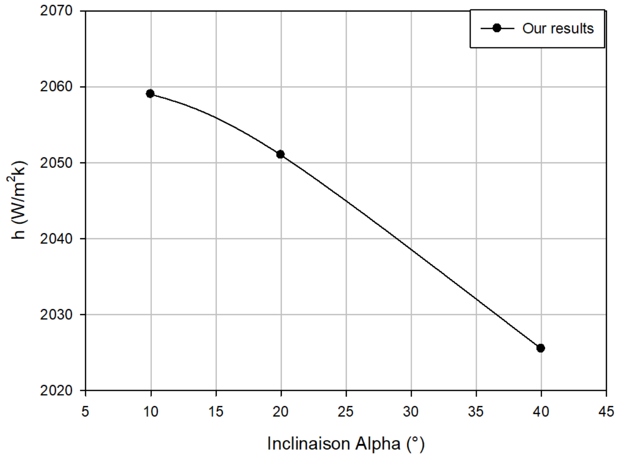


Fig. 9. Variation of average heat transfer coefficient.

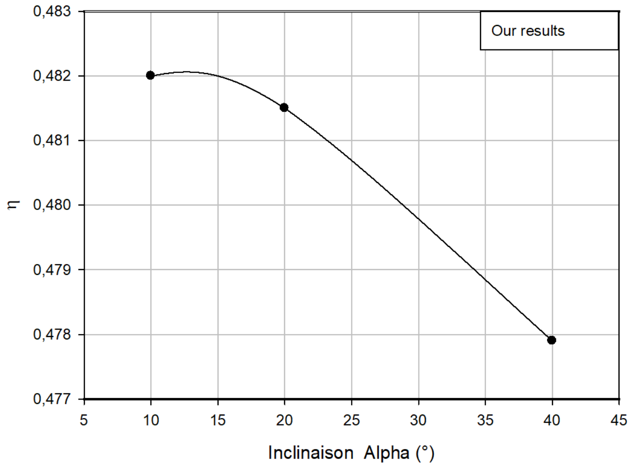


Fig. 10. Overall performance factor evaluation.

3.4. Dynamic pressure

The contours of dynamic pressure are added in Fig. 6, the pressure values are very high at the inlet of the shell and very low at the outlet because of the existence of recirculation zones in the back of the baffles. The dynamic pressure tends to increase with the increases of the baffles inclination angle α.

3.5. The pressure drop

The pressure drop (Fig. 6) decreased by approximately 0.996% and 0.992 % with increasing baffles inclination angle α from 20° to 40°. As can be shown lower pressure drop with the baffles inclination angle of 40° compared to the baffles inclination of 10° and 20° because of easier fluid guidance. However, the pressure drop in the shell side decreases with the increase of the baffles inclination angle α.

3.6. Longitudinal temperature profile

Fig. 8, depicts the contour of temperature at the symmetrical plane for angles 10°, 20°, and 40°. Not that the fluid temperature increase with the increases of the baffle's inclination angle. The heat exchangers divide into the cycles correspond to a means an average temperature. The two heat exchangers with 10° and 20° divided into 4 cycles with fluid temperatures different, the 1st limit to the inlet of the 2nd baffle, the 2nd cycle to the 2nd of the 4th baffle, the 3rd cycle to the 4 th baffle of the 6 th baffle, the 4th cycle to the 6 th baffle of the shell outlet.

The heat exchanger with baffles inclination of α =40° can divide only into 3 cycles, the 1st cycle to the inlet at the 3rd baffles, the 2nd cycle to the 3rd at the 5th baffle, the 3rd cycle to the 5th baffles at the shell outlet.

3.7. The heat transfer coefficient

The heat transfer coefficient tends to decreased by 0.996% and 0.983% with an increase of baffles inclination angle α from 20° to 40°. Note that using the great baffles inclination angle minimized the turbulence, consequently decreased the heat transfer coefficient.

3.8. Overall performance factor

Fig. 10, shows the evolution of the overall performance factor as a function of the baffles inclination angle. Not that the increase of the inclination baffles angle from 20° to 40° the overall performance factor decreases from 0.998% to 0.991% as the pressure drop remains high because of the high velocity generated by the baffles of 10°.

4. Conclusion

In this study, we have attempted to investigate the influence of various baffles inclination angles 10°, 20° and 40° on fluid flow and heat transfer characteristics in shell-and-tube heat exchangers for the same mass flow. The numerical simulations are conducted with ANSYS Fluent. The results of the simulation, pressure drop, heat transfer coefficient, overall performance factor for the heat exchangers geometry determined. The high velocity generated by the baffles near the walls of the tubes results a significant thermal improvement in the shell side. The results prove that the average temperature at the outlet increases by 3.4 K, the average heat transfer coefficient decreased by 0.983%, the pressure drop in the shell decreased by 0.992%, the overall performance factor decreased by 0.83% when the baffles inclination angle is increased from 10° to 40°. The higher heat transfer coefficient got in the small baffles inclination and the lower pressure drop giving by the great baffles inclination. A lower pressure drop of these heat exchangers than the heat exchanger with segmental baffles. The paper presents a novel concept of the baffles inclination for lower pumping power.

Author Contributions

A. Youcef designed, simulated the heat exchangers and discussed the results; R. Saim conducted the project and analyzed the theory validation. All authors approved the final version of the manuscript.

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Conflict of Interest

The authors declared no potential conflicts of interest with respect to the research, authorship and publication of this article.

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Nomenclature

f	Friction factor	N_b	Number of baffles
D_s	Shell diameter [m]	Nu_0	Nusselt number of the base case
d	Tube diameter	f_0	Friction factor of the base case
D_h	Hydraulic diameter [m]	Γ	Generalized diffusion coefficient
u,v,w	Average velocity [m/s]	ε	Dissipation rate of turbulence energy [m²/s3]
L	Tube length[m]	ρ	Density of the water[kg/m3]
h	Heat transfer coefficient[W/m ²K]	μ	Dynamic viscosity (kg/(m s)
Nu	Averaged *Nusselt* number	λ	Thermal conductivity [W/m K]
T_{in}	Inlet temperature [K]	σ_k	Prandtl number for k
T_{out}	Outlet temperature [K]	σ_ε	Prandtl number for ε
P	Pressure [Pa]	η	Overall performance factor
B_c	Distance between two baffles [m]	α	Baffles Inclination Angle (°)
Re	Reynolds number		

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