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Numerical Study on Heat Transfer Enhancement in a Curved Channel with Baffles

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ABSTRACT

Curved channel is commonly used in cement mill's journal bearing. In this study, heat transfer enhancement of curved channel having a rectangular cross-section with using inclined shape baffles is numerically investigated. The effects of different parameters of baffles, i.e. attack angle (α= 45°, 60°, and 90°), and the number of baffles (NB=9 and 13 baffles) are examined. The water is selected as working fluid for laminar and turbulent flow region. A standard k-ε turbulence model together with enhanced wall treatment is applied to solve the complex flow in Re of 500–5000. Influences of those parameters on heat transfer and friction performances in terms of Nusselt number, friction factor , Nussellt number enhancement ratio, and thermal performance factor, respectively are studied. The results show that the best condition to achieve maximum heat transfer at angle $\alpha = 90^{\circ}$, NB=13 and Re=5000 compared with other conditions. Furthermore, the maximum thermal performance factor (TEF) of the curved channel with using baffles is 4.4 at the same condition. This indicated that the geometry of baffles inside curved channel can improve the heat transfer significantly with reasonable increase in friction losses.

Keywords:
Heat transfer enhancement; Baffles; Nusselt number; Thermal enhancement factor

1. Introduction

Vortex-flow device is considered one of the most important techniques to thin and destroy the boundary layer near the wall in order to achieve a higher heat transfer coefficient. In other words, because of the need to achieve higher thermal performance and promote heat transfer in most engineering applications, it is essential that new procedures and mechanisms be considered. In general, vortex/swirl flow is widely applied in cooling/heating systems in many industrial/engineering applications such as drying or curing of agricultural/industrial products, cooling of the gas turbine blades, chemical process plants, cooling of thermal load or heating process from duct airflow through

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the attic space of a house and so on. Therefore, finding methods with low cost and high performance is one of the challenges for modern heat transfer.

Nasiruddin and Siddiqui [1] performed a numerical study on heat transfer enhancement in a heat exchanger tube by installing a baffle with a focus on baffle size and orientation. The results showed that the Nusselt number enhancement is almost independent of the baffle inclination angle. The majority result of this research suggested that a significant heat transfer enhancement in a heat exchanger tube can be achieved by introducing a baffle inclined towards the downstream side, with the minimum pressure loss.

Supattarachai *et al.,* [2] conducted an experimental study of heat transfer enhancement, friction factor and pressure drop in a square duct heat exchanger fitted with 45◦ and 90◦ diagonal obstacles (ribs). The study used air as a working fluid over Reynolds number ranges from 4000 to 26,000. The result showed that the obstacle with 45◦ provides the maximum value of heat transfer and pressure drop than 90◦ for all obstacle pitch ratio. Also, they claimed that the average Nusselt number for the two baffles case is 20% higher than the one baffle case and 82% higher than the no baffle case.

Promvonge *et al.,* [3] conducted a numerical study of a turbulent flow and the heat transfer characteristic in a square duct put in inclined plane at 30 ° fitted out with a finned strip for a Re from 4000 to 20000. The results showed that maximum thermal performance is found to be 1.95 for using the finned tape at fin blockage ratio (*BR* = 0.2) and pitch ratio (*PR* = 1) whereas the Nusselt number ratio is about 4.5 at lower *Re*.

In another respect, Promvonge *et al.,* [4] examined the influence of inclined horseshoes baffles placed repeatedly in a tubular heat exchanger on thermal enhancement factor. The horseshoe baffle elements with an inclination angle of 20 were inserted periodically into the test tube at three different baffle-pitch ratios (PR = 0.5, 1.0 and 2) and -width or blockage ratios (BR = 0.1, 0.15 and 0.2). The experimental results revealed that the tube fitted with inclined horseshoes baffles provides considerable improvement of the heat transfer rate over the plain tube around 92-208% while the friction factor is increased at about 1.76-6.37 times.

Sriromreun *et al.,* [5] studied experimentally and numerically the influence of baffle turbulators on heat transfer augmentation in a rectangular channel. The results showed a significant effect of the presence of the Z-baffle on the heat transfer rate and friction loss over the smooth channel with no baffle.

The heat transfer behavior of the solar air channel of aspect ratio of 10.0 with 60° angled broken multiple V-type baffles is analyzed experimentally by Kumar *et al.,* [6]. The obtained experimental results showed that higher overall thermal performance occurred at a relative baffle width of 5.0. Also, the results revealed that the broken multiple V-type baffles are thermo-hydraulically superior as compared to the other baffles shaped solar air channels.

Sahel *et al.,* [7] studied the effects of baffle design on the heat transfer phenomenon in the channel, numerically. The study employed a perforated baffle having a row of four holes placed at three different positions. The obtained results showed that a new design of baffle with Pores Axis Ratio (PAR) of 0.190 led to increasing the heat transfer rate from 2% to 65% compared with the simple baffle. Du *et al.,* [8] explored the effects of five geometric parameters on the efficiency of overlapped helical baffled heat exchangers and which are: diameter of tube, overlap size, helix angle, tube layout and central distance of tube. They reported that the effect of overlap size is most significant than the other parameters.

Nanan *et al.,* [9] investigated numerically and experimentally heat transfer, friction loss and thermal performance factor associated with the use of the baffles in circular tubes. The studies encompassed three different baffle width ratios (w/D = 0.1, 0.2 and 0.3), three baffle twist ratios (y/w = 2.0, 3.0 and 4.0), and Reynolds numbers from 6000 to 20,000. The results showed that the transverse twisted-baffles with the smallest twist ratio (y/w = 2.0) give higher thermal performance factors than the ones with 3.0 and 4.0 by around 4.7–6.1 and 10.2–15 %, respectively. Lei *et al.,* [10] studied the influences of various baffle inclination angles on fluid flow and heat transfer of heat exchangers with helical baffles. According to the results, the pressure drop decreases with increasing of baffle diagonal angle. Furthermore, they found that when consuming the same pumping power; it is the heat exchangers will have a higher heat transfer coefficient with helical baffles. The thermal performance enhancement factor increased when the baffle diagonal angle $\alpha < 45^{\circ}$, and decreased at $\alpha > 45^{\circ}$.

Karwa and Maheshwari [11] carried out an experimental study of heat transfer and friction in a rectangular section duct with fully perforated baffles (open area ratio of 46.8%) or half perforated baffles open area ratio of 26%) at relative roughness pitch of 7.2-28.8 affixed to one of the broader walls. The baffle wall was uniformly heated while the remaining walls are insulated. The said study showed that the half perforated baffles are thermo-hydraulically better to the fully perforated baffles at the same pitch. Amirtharaj [12] studied heat transfer and fluid flow in shell and tube heat exchanger with various design methods of baffle. They used CFD simulation for two models of shell and tube heat exchanger, the first with sector baffles and second with diagonal baffles. The results showed a high effect of heat transfer performance with diagonal baffles for heat exchanger (shell and tube). Skullong *et al.,* [13] performed a numerical and experimental investigation to study the heat transfer enhancement in a heat exchanger square-duct fitted with 30_ oblique horseshoe baffles (HB). Various geometrical parameters of baffles are applied; air flow and heat transfer behaviors are presented for turbulent flow region, Reynolds number ranging from 4000 to 25,000. Numerically, the study displayed heat transfer behaviors such as streamlines, temperature and Nusselt number contours of the duct flow model. They reported that the thermal performance of using the HB is much higher than that of the wire coil insert. Li and Gao [14] reported that when baffle height equals to corrugation height, friction factor is no longer proportional to Re in the channel with 60_ and 90_ apex angle. This research used delta-shaped baffles for further enhance heat transfer in cross-corrugated triangular ducts. Also, a standard k-e turbulence model together with enhanced wall treatment is applied to solve the turbulent flow in Re of 1000–6000.

In the literature review above, most of the investigations focused on heat exchanger with different baffles arrangements in the turbulent flow regime using a rectangular and straight channel for thermal performance improvement. Therefore, further investigations need for using a new design of baffles inside a curved channel instead of the traditional channel. In this research, baffles with special sections same as horseshoe baffles used to enhance heat transfer through laminar and turbulent regimes across curved channel. Regards the baffle, different parameters such as baffles angles (α= 45°, 60°, and 90°) and different numbers (9, 13) have been studied.

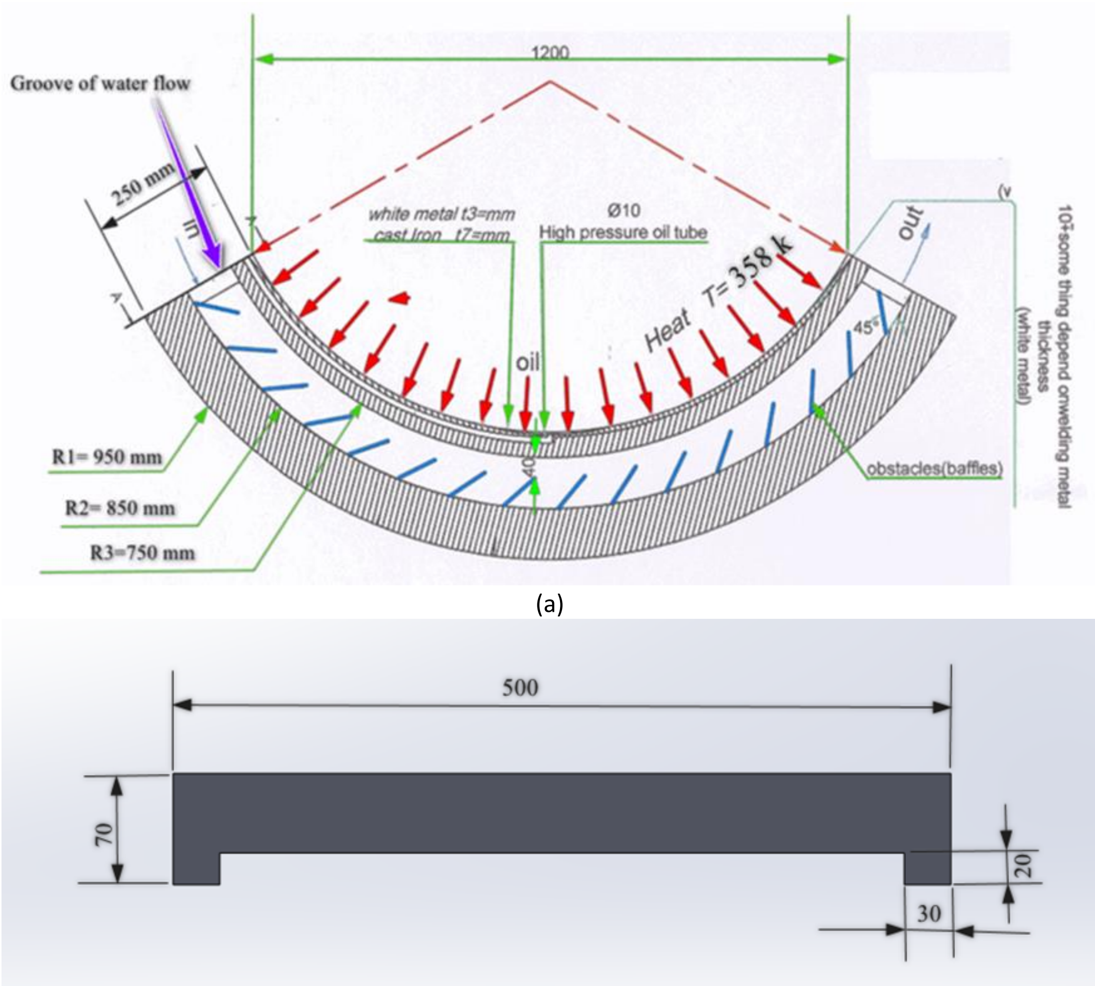
2. Problem Description

Figure 1 shows the 2-dimensional geometrical model of the current study; (a) curved channel with baffles, and (b) baffles. The basic geometry consists of two walls, the upper and lower walls, and the average distance between the walls is 100 mm, while the width and length of the channel are 500 mm and 1572 mm, respectively.

The inside radius and outside radius of the curved channel are 750 mm and 850 mm, respectively. It can be assumed that the flow is fully developed, laminar and turbulent, incompressible, 3-dimensional and steady.

The description of the horseshoe-baffle elements repeatedly inserted into the test channel is shown in Figure 1(b). The thickness of the baffle element was 2 mm and formed in horseshoe baffle.

In this study, baffles have placed in various angles (α= 45°,60°, and 90°) and different numbers (9, 13).



(a)

(b)

Fig. 1. Schematic diagram (a) curved channel,(b) beffles

3. Governing Equations and Boundary Conditions

A curved duct model were analyzed by equation of the continuity, equation of the momentum, equation of the energy in the Cartesian tensor, these equations can be represented as follows

Equation of the continuity

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

Equation of the momentum

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

Equation of the energy

$$\frac{\partial}{\partial x_j}\left(\rho\overline{u_j T}\right) = \frac{\partial}{\partial x_j}\left(\left(\frac{\mu}{Pr}+\frac{\mu_t}{Pr_t}\right)\frac{\partial \overline{T}}{\partial x_j}\right) \tag{3}$$

where ρ is represent the density of the fluid(kg/m^3) that mean density of water in this paper, P is Pressure of fluid, u_i is velocity of fluid $(m/s)$and μ is dynamic viscosity of fluid $(kg/m.s)$

$$-\rho\overline{u_i' u_j'} = \mu\left(\frac{\partial \overline{u_i}}{\partial x_j}+\frac{\partial \overline{u_j}}{\partial x_i}\right)-\frac{2}{3}k\rho\delta_{ij} \tag{4}$$

where
δ_{ij}= kronecker baffle function

$$\mu_t = C_\mu \rho \frac{K^2}{\varepsilon} \tag{5}$$

where μ_t is turbulent viscosity, k is turbulent kinetic energy, and ε is the rate of turbulent dissipation. Also it can represent transport equations of (k-ε) model as in below:

$$\overline{u_j}\frac{\partial k}{\partial x_j} = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k-\varepsilon \tag{6}$$

$$\overline{u_j}\frac{\partial \varepsilon}{\partial x_j} = \frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_1\frac{\varepsilon}{K}G_k-C_2\frac{\varepsilon^2}{\mathrm{K}} \tag{7}$$

The value of turbulent kinetic energy (G_k) can be calculated in the following equation

$$G_k = \mu\left(\frac{\partial \overline{u_i}}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\frac{\partial u_i}{\partial x_i} \tag{8}$$

Computational domains and boundary conditions are applied at the curved channel having baffles which included velocity inlet and temperature of 303.15 K, pressure outlet condition. In addition, uniform temperature (358.15 K) on the upper wall whereas adiabatic condition applied for the lower wall and baffles.

The hydraulic diameter is computed based on cross section area (A_{cross}) and the perimeter of wetted (P) as

$$D_h = \frac{4A_{cross}}{P} \tag{9}$$

$$Re = \frac{\rho D_h v}{\mu} \tag{10}$$

And, average Nusselt number can be defined as below

$$\overline{Nu} = \frac{\overline{h}D_h}{K} \tag{11}$$

The heat transfer coefficient is evaluated from

$$\bar{h} = q'' \cdot \frac{\ln\left(\frac{T_w - T_{m,in}}{T_w - T_{m,out}}\right)}{(T_w - T_{m,in}) - (T_w - T_{m,out})} \tag{12}$$

$$q'' = \dot{m}\, C_p\, (T_{m,in} - T_{m,out})/ A \tag{13}$$

The pressure drop can also be obtained as

$$f_b = \frac{\Delta p}{[L/D_h]} \times \left(\frac{\rho_{water} U^2}{2}\right) \tag{14}$$

Then, thermal Performance Enhancement Factor [15]

$$\text{TEF} = \left[\frac{Nu_b}{Nu_o}\right] \times \left[\frac{f_b}{f_o}\right]^{\frac{1}{3}} \tag{15}$$

where

q'' : Heat flux (w/m^2)
Δp : Pressure drop (Pa)
D_h : Hydraulic diameter
TEF : Thermal Performance Enhancement Factor
Nu_b : Nusselt of duct with baffles
Nu_o : Nusselt of duct without baffles or smooth
f_b : friction factor of duct with baffles
f_o : friction factor of duct without baffles

4. Numerical Method and Grid Independence Test

The Finite Volume Method (FVM) is used to solve the governing equations with corresponding boundary conditions by employing the CFD commercial software ANSYS FLUENT 18.2. The k-ε turbulent model with enhanced wall treatment is applied and diffusion term in the energy equation is discretized by Quadratic upstream interpolation for convective kinetics differencing scheme (QUICK). The governing equations were discretized by the QUICK numerical scheme, decoupling with the Semi-implicit method for pressure-linked equations (SIMPLE) algorithm and solved using a finite volume approach. In the current investigation, the convergence criterion is considered as Eq. (1) and (2) for continuity, momentum and turbulence equations while Eq. (3) for energy equation.

In the current study, the procedure of the computational domain of fluid flow for the curved channel is conducted by regular elements of Cartesian or tetrahedron. According to this module, a grid independence test was included with variant numbers of cells about 550000, 2250500, 3425500 4236494, and 5122000. The average Nusselt number is calculated at angle $\alpha = 90^{\circ}$ in case NB=13 and Re=500 as shown in Table 1.

Table 1 briefly demonstrates the relative error defined in Eq. (16) between the new result of Nusselt number (N1) and the previous result of Nusselt number (N2). By comparing the outcomes in terms of Nuav to study the effect of grid size, it is detected that the mesh size of 413520 elements can achieve and confirm grid independence solution as shown in Table 1. Consequently, the increase in the number of cells after these values, it was no very useful. Therefore, the grid independence of 4236494 cells was used in this simulation to save computation time.

$$Relative\ error = \left|\frac{N_1 - N_2}{N_1}\right| \times 100 \tag{16}$$

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Table 1
Grid independence test

Grid size	Average Nusselt number	Relative error%
550000	521	-
2250500	521.379	0.072691842
3425500	521.645	0.050992533
4236494*	521.854	0.040049516
5122000	521.898	0.008430766

5. Code Validation

For validation of the accuracy of the numerical procedure, the average Nusselt number obtained from the current numerical approach is compared with the experimental data reported by Promvonge *et al.,* [4] as displayed in Figure 2. The numerical method is carried out for the smooth tube (without baffles) and with horseshoe-baffles condition at blockage ratios=0.2 and pitch ratio=0.5. It is noticed in Figure 2 that there is an appropriate consistency between the results, which illustrates the validity of the numerical method.

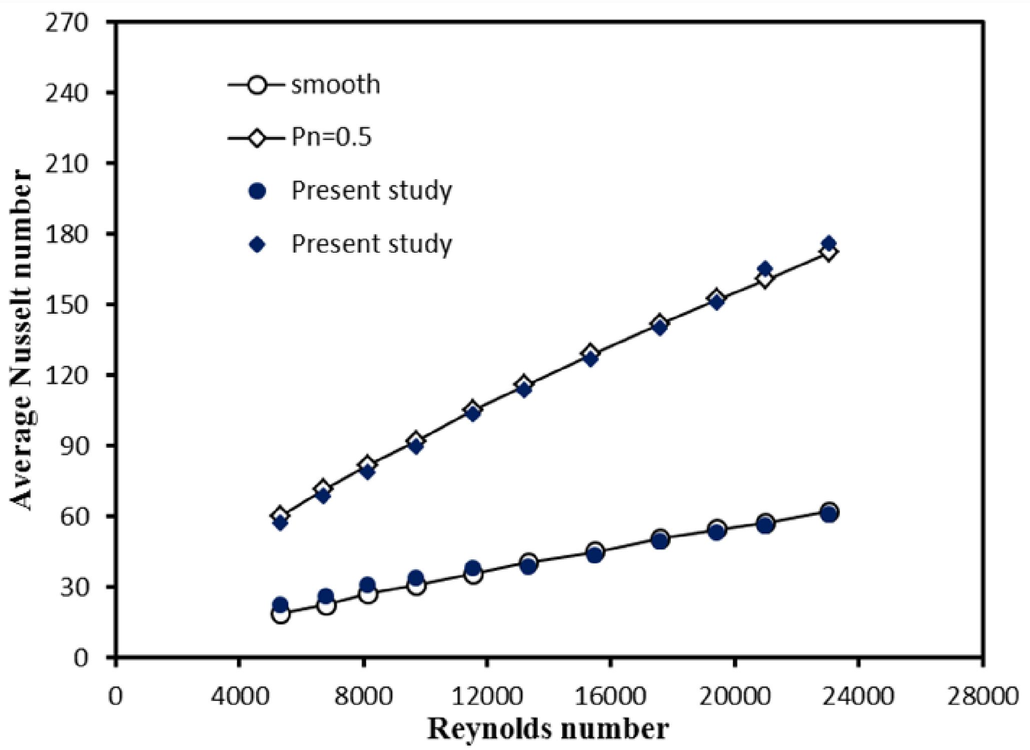


Fig. 2. Comparison between the current study and the results of experimental data by Promvonge *et al.,* [4]

6. Results and Discussion

Laminar and turbulent forced convection flow of water through curved channel having baffles has been numerically investigated over Reynolds number ranges of 500-5000. Figure 3 shows the effect of using baffles on heat transfer enhancement via displays velocity streamlines.

In flow channel without baffle, parallel flow is the main flow pattern in the curved channel. Therefore, there is no swirl flow inside tested channel. And only a small quantity of fluid flows in the lower wall, creating clockwise vortices. These longitudinal vortices are considered to be the reason for momentum and heat transfer enhancement. That is also why we try to add baffles to create more strong vortices in the curved channel. As shown in these figure, the vortices are indeed generated before and after the baffles. Because the baffle has the same sectional area of channel inlet, incoming flow is entirely obstructed and impinges to the lower wall of channel. Subsequently, it turns towards the upper space in these troughs, and veers across the next obstructive baffle. High-velocity gradients exist in the location near the baffle and the lower wall, where flow separation occurs.

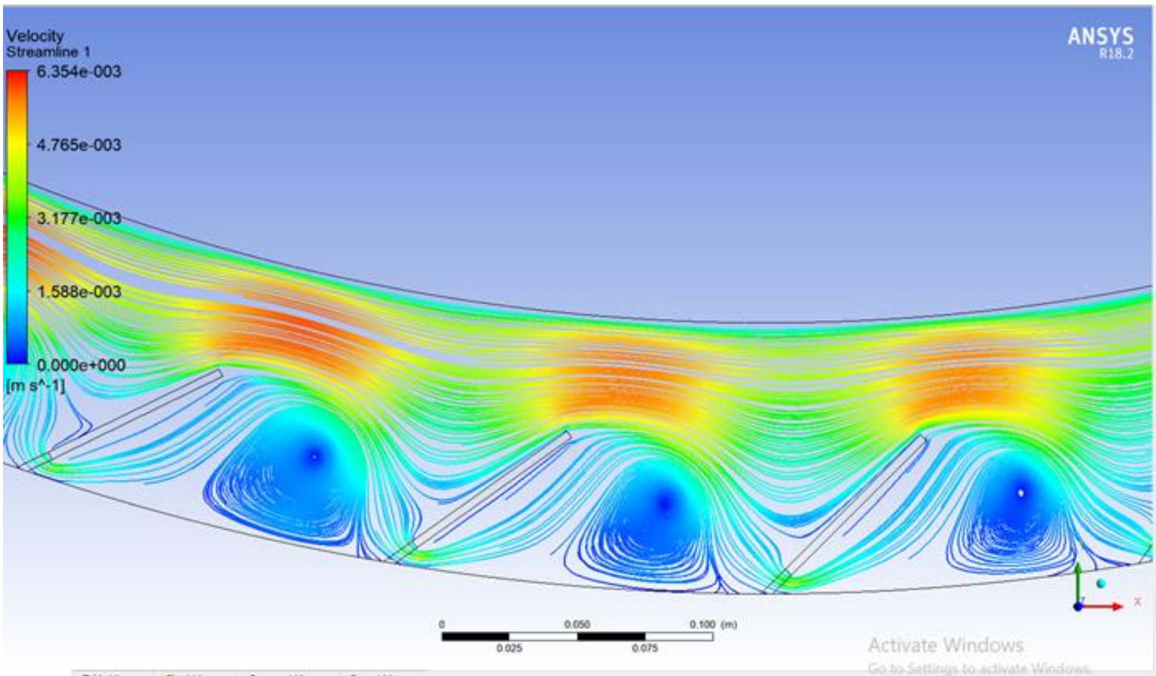


Fig. 3. Velocity streamlines peak angle of $\alpha = 45°$ and NB=9 at Re=5000

Figure 4 represents the variation of friction factor (f) and Reynolds Number (Re) with different values of peak angles (α=90°, 45° and 60°) and the number of baffles (9 and 13). It should be noted that the friction factor decreases as the Reynolds number increases for all values of α and NB. The reason behind that back to the complex flow inside the curved channel which can be considered as a finned channel. In this case there are two forces effect on the friction, the first force is friction drag and it commensurate with (Re) and second force is pressure drag which it resulted by separation of boundary layer.

By comparing the peak angles of the channel, the friction factor is affected by the peak angle, where friction factor of peak angle ($f_{90^\circ,13}$) is higher than that of ($f_{45^\circ,13}$) and ($f_{60^\circ,13}$). Also the friction factor is affected by the number of baffles where friction factor in ($f_{90^\circ,13}$) is higher than ($f_{90^\circ,9}$), and ($f_{45^\circ,13}$) is higher than ($f_{45^\circ,9}$) and ($f_{60^\circ,13}$) is higher than ($f_{60^\circ,9}$). Consequently, it can be concluded that with bigger peak angle trends to obtain higher friction factor with the increase of the number of baffles.

Figure 5 shows difference of mean Nusselt number with different peak angles and numbers of baffle. In general, Nusselt number increases with increasing baffles number. This perhaps feature to larger vortexes and turbulence resulted by obstructive baffles. Also, it should be noted that the parameters, α and (NB), have a significant effect on the average Nusselt number. At any value of α and NB, the average Nusselt number increases with Reynolds number due to the increases of the convection effect. In addition, the average Nusselt number increases with the number of baffle and peak angle at a given Reynolds number. It can be concluded that the number of baffle and peak angle have a positive effect on heat transfer. For instance, at α=60˚ and Re=5000, average Nusselt number at NB=13 and 9 is 1442.83 and 1433.71, respectively. Also, at α=90˚, Nu= 576.77 at NB=13 while it equal 521.854 when NB=9 at the same Re=500.

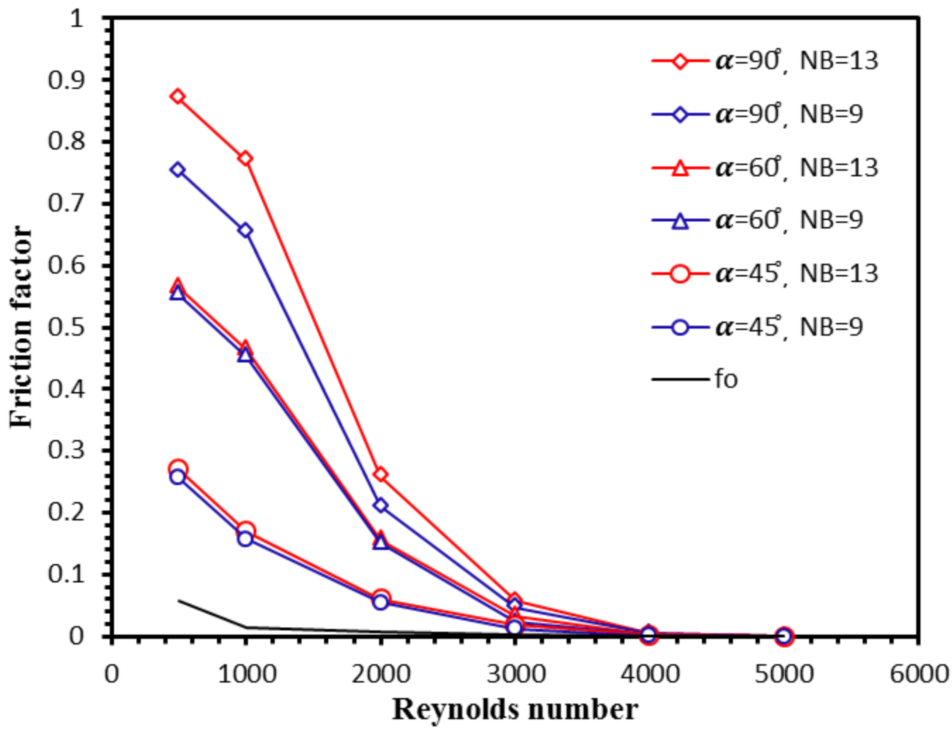


Fig. 4. Friction factor for various α and NB

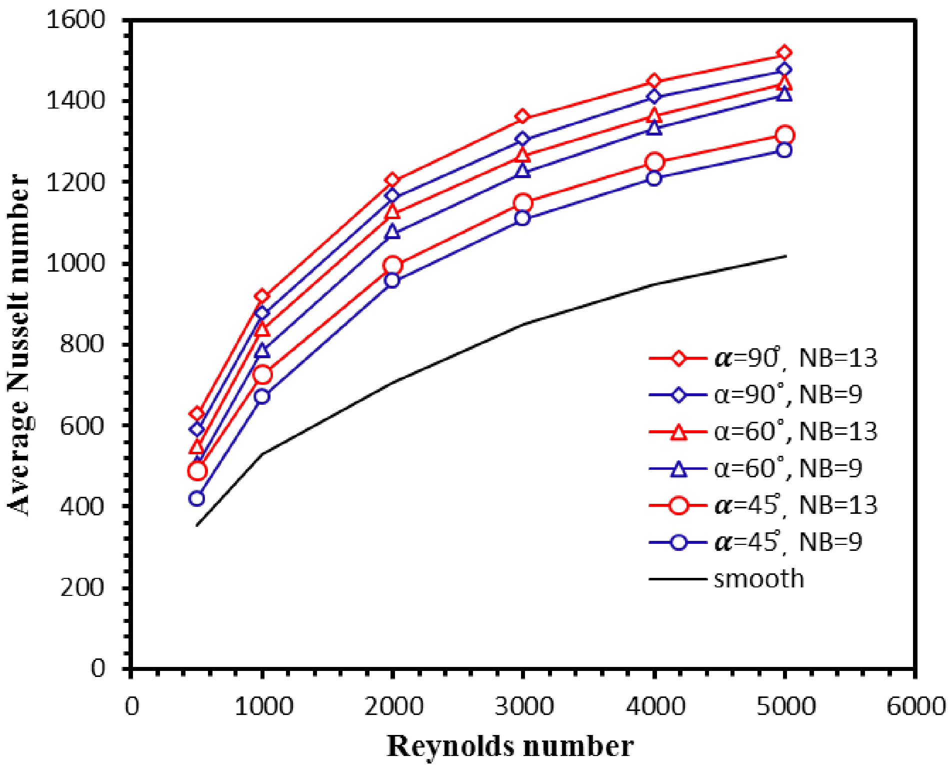
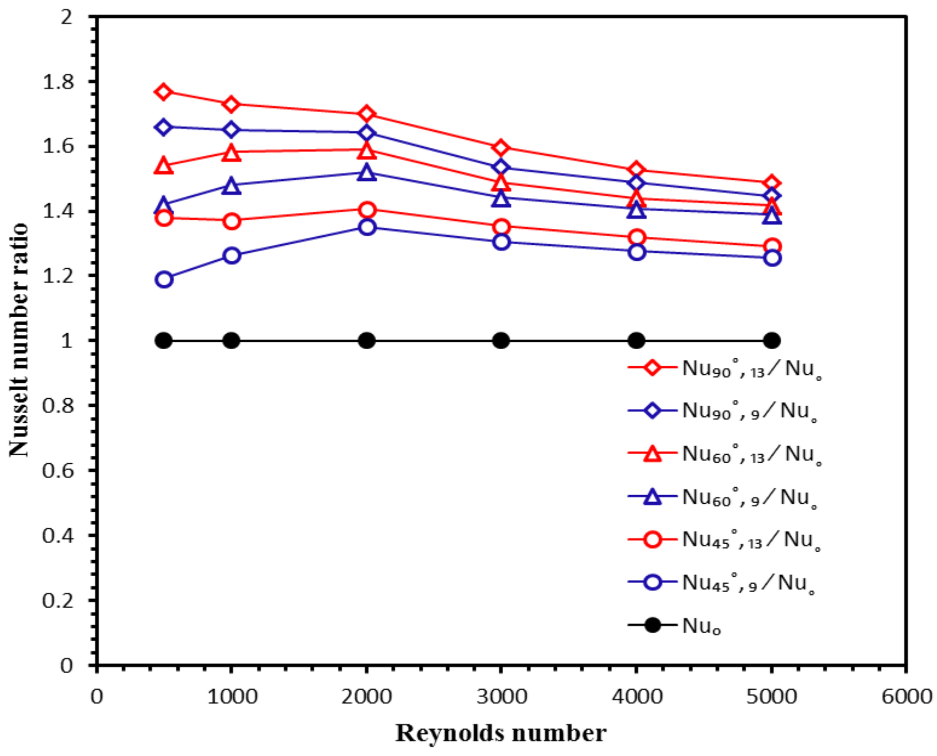


Fig. 5. Average Nusselt number for various α and NB

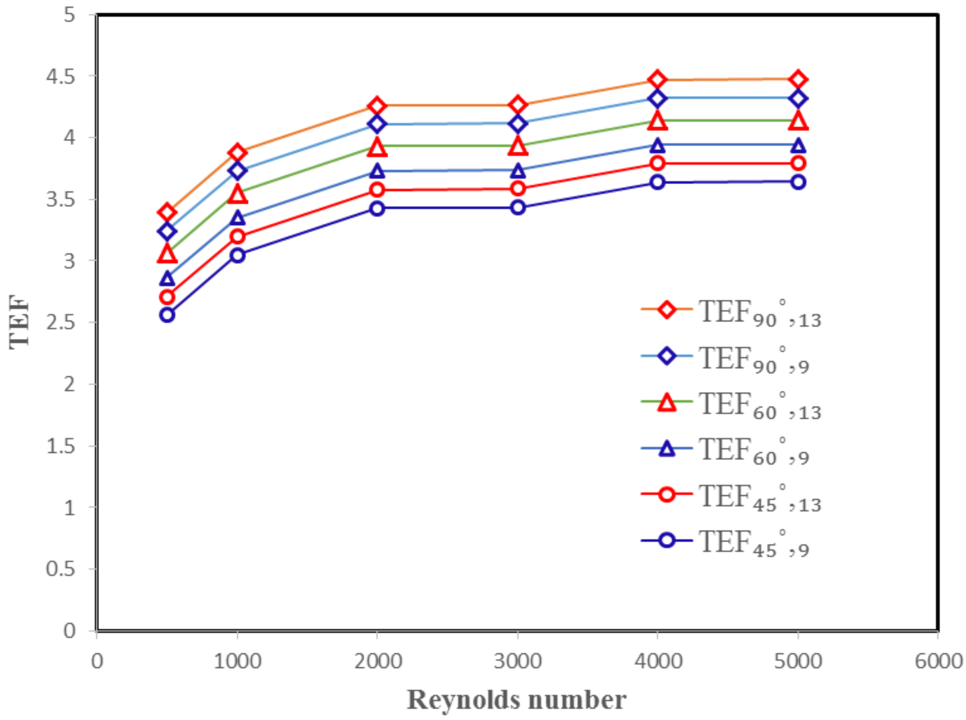
The performance evaluations in the curved channel with a rectangle section for inserted horseshoe baffles have planned in regard to the Nusselt number ratio (Nu/Nu_o) and thermal performance enhancement factor (TEF).

Figure 6(a) introduces by and large the differences of the Nu/Nu_o with the Reynolds number at dissimilar status for the curved channel with the horseshoe baffles. The horseshoe baffles in the curved channel with a rectangle section can enhance the heat transfer rate higher than the smooth channel if the values of ($Nu / Nu_o > 1$). Therefore, Nusselt number ratios are larger than unity, implying that the proposed baffles through curved channel are advantageous over the smooth channel. It is seen that the ratio of $Nu_{90^\circ,13}/Nu_\circ$ at Re =500 gives the height value of the heat transfer rate about 76% while $Nu_{45^\circ,9}/Nu_\circ$ at Re =500 gives the low value of the heat transfer rate. The explanation behind this is the fact that the blending of the working fluid in curved channel turns out to be better with the higher angle and leads to improve heat transfer.

Figure 6(b) represents the relation of the Thermal performance enhancement factor TEF with the Reynolds number (Re) at different cases. In general, it can note that the TEF tends to little increase somewhat with raising the Reynolds number in laminar flow cases while it tends to little decrease with augmenting the Reynolds number (Re) in turbulent flow cases. Generally, the TEF increases with increasing Reynolds number for every case of channel. Likewise, it can be plainly observed that the $TEF_{90^\circ,13}$ case of curved channel gives the best performance over Reynolds number range. In contrast, the $\mathrm{TEF}_{45^\circ,9}$ has a lower performance when compared to the other configuration. This is because thermal performance enhancement depends on the heat transfer enhancement which can be deduced from enhanced Nu results. Also, the results indicated that TEF more than unit for all tested cases. In other words, this means that heat gained or heat transfer enhancement is much more than pressure loss. In addition, this gives indicator that the baffle profile had a significant impact on the thermal performance enhancement.



(a)



(b)

Fig. 6. (a) Nusselt number enhancement ratio, (b) thermal performance enhancement factors (TEF)

7. Conclusions

Laminar flow, turbulent flow, and enhancement heat transfer in a curved channel with a rectangle section concerning inclined baffles influence are numerically studied in this research. Accordingly, the basic conclusions were obtained as follows:

i. The baffles technique of curved channel had a great impact on thermal performance enhancement compare with similar cases without using baffles.
ii. Numerical results show that the average Nusselt number enhanced with increasing Reynolds number, baffle number and peak angle of the curved channel, but the friction factor will also increase.
iii. The adopted geometry of curved channel with baffles can improve heat transfer enhancement in the range of 2.5-3.8 times that of same channel without baffles.
iv. Thermal performance enhancement factors (TEF) increases with increasing baffle number and peak angle and best value at specific geometrical parameters.
v. Thermal maximum performance enhancement factors (TEF) is 4.4 can achieved geometrical parameters of baffles at α=90 ˚ and NB=13.

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