International Journal of Science and Research (IJSR)
ISSN (Online): 2319-7064
Impact Factor (2012): 3.358

Numerical Study of Fluid Flow and Heat Transfer over a Circular Tubes Bank of Heat Exchanger with Different Shapes of Vortex Generator

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Abstract: ***A numerical study is carried out to study the heat transfer and flow in the plate-fin and tube heat exchangers with three different shaped vortex generators mounted behind the tubes with Reynolds number ranging (7000 ≤ Re ≤ 11000). A computational fluid dynamics code (FLUENT) is used to solve steady (2-D) Navier- Stokes and energy equations. (k-ε) model used to remedy the turbulent effects. The effects of three shapes of winglets is looked at (airfoil, rectangle and triangle) with different angles of attack (30 and 45) has been investigated on average heat transfer (Nu), friction coefficient and pressure drop. The results show that there is an effect for using winglet pairs on heat transfer, friction coefficient and pressure drop, also, heat transfer depends on the shape, angle of attack of winglet. The triangle is the best shape for enhancing heat transfer and (α = 45) is the best angle of attack for enhancing heat transfer.***

Keywords: Vortex Generator, Heat transfer enhancement, heat exchangers

1. Introduction

Convectional types of heat exchanger are externally plain tube in cross flow. They are widely used in chemical, petrochemical, automotive industry, cooling towers, heated pipes refrigerators of power plants as well as in applications for heating, refrigeration and air conditioning. In finned tube heat exchanger liquid or steam flows through the tube and gas through fin ducts. Heat transfer is closely related to fluid dynamics. That is why heat transfer is considered simultaneously with fluid dynamics.[1] As we know, how to reduce the thermal resistance is the key for the heat transfer enhancement. One frequently used method for heat transfer enhancement employs surfaces that are interrupted periodically along the stream wise direction. Typically, these surfaces are in the form of wavy, louver, slit, or offset strip fins. Despite the fact that interrupted surfaces can significantly improve the heat transfer performance, the associated penalty of pressure drop is also tremendous [2].The (V.Gs) are small plates placing in the stream flow for mixed flow, disturbing flow and controlling the growth of boundary layer [3] Vortex generators such as pins, ribs, wings, winglets ---etc have been successfully used as a powerful way for enhanced heat transfer in the development of modern heat exchangers. Vortex generators form secondary flow by swirl and destabilize the flow. They generate the longitudinal vortices and create rotating and secondary flow in the main flow which can raise turbulent intensity, mix the warm and cold fluid near and in the center of channel and increase the heat transfer in the heat exchangers [4]. Sohal and O,Brien 2001, summarized the researches about enhancement heat transfer from fin-tube heat exchanger by using winglet. All experimental researches have been done by (1- INEEL researchers Idaho National Engineering &Environmental laboratory, 2-YNU Yokohama National University). The INEEL researchers investigated the local heat transfer in the fin in the wake region behind the cylinder. The heat transfer enhancement is about 25-35% for a Reynolds number ranging from 1200-2000, pressure drop also increase by moderate amount. YNU researchers investigated local heat transfer from the fin in fin-tube heat exchanger by placing a winglet behind the tube banks, the heat transfer enhanced by 10-25 % and pressure drop increase by 20-35 % with Reynolds number ranging from 350-2100 [5]. Sohankar and Davidson 2003, investigated the flow and heat transfer structures in plat fin heat exchanger with vortex generators numerically by FVM method with collocated grid, the grid distribution was non-uniform body fitting coordinate. The type of vortex generators used in this study was rectangular winglet. In this study the Reynolds number was 2000 and Prandtl number 0.71 and the flow in channel plat simulate as a part of plat-fin heat exchanger. The results show that the heat transfer enhanced by using vortex generators with increasing pressure losses, also heat transfer increases with increase angle of attack [3]. Biswas et al. 1994, numerically investigated the flow structure and heat transfer in a three-row fin-tube heat exchanger with built-in delta winglet pairs. The staggered array of tube rows, and a punched-out delta winglet pair with an aspect ratio of 2 and an attack angle of 45° was located behind each tube with $\Delta x/D = 0.5$ and $\Delta y/D = 1.0$. At a Reynolds number of 500, the local heat transfer was found to increase by more than 240% at a location about 12 times the channel height downstream of the inlet. The span wise average Nusselt number at Re=646 compared favorably with the experimental results from the same geometry for most of the stream wise locations [6]. Fiebig et al. 1994 measured the heat transfer enhancement and flow losses for a fin-tube heat exchanger with three rows of flat tubes in a staggered arrangement. Delta winglet pairs with an aspect ratio of 2 and an attack angle of 45° were used. The results were compared to similar experimental results for round tubes. The heat transfer was found to increase 100% for flat tubes while only 10% for round tubes. It was found that the heat exchanger element with flat tubes and vortex generators gave nearly twice as much heat transfer and only half as much pressure drop as the corresponding heat exchanger element with round tubes [7]. Leu et al. 2004, numerically and experimentally investigated the heat transfer and fluid flow in a plate-fin and tube heat exchangers with an inclined block shaped vortex generator mounted behind the tubes. The effect of the span angle (30°, 45° and 60°) in the Reynolds range 400 to 3000 was investigated. Experiments were also performed to visualize the temperature distribution and local flow structure. The case with span angle of 45° provided the best heat transfer augmentation. The reduction in the fin area of 25% was obtained at Re=500 [8]. Joardar and Jacobi 2008, experimentally evaluated the potential of winglet type vortex generator (VG) arrays, for air-side heat transfer enhancement, by full-scale wind-tunnel testing of a plain-fin-and tube heat exchanger. The effectiveness of a 3VG alternate-tube inline array of vortex generators was compared to a single-row vortex generator design and the baseline configuration [9]. J.M. Wu and W.Q. Tao 2008, achieve heat transfer enhancement and lower pressure loss penalty, even reduction in pressure loss; two novel fin-tube surfaces with two rows of tubes in different diameters are presented in this paper. Numerical simulation results show that the fin-tube surface with first row tube in smaller size and second row tube in larger size can lead to an increase of heat transfer and decrease of pressure drop in comparison with the traditional fin-tube surface with two rows of tubes in the same size[10]. Chu et al. 2009, further numerically investigated the flow structure and heat transfer enhancement in a full scale fin and tube heat exchanger with a rectangular winglet pair in the Reynolds number range of 500 to 800.Three configurations inline RWP, inline 3RWP and 7 RWP were compared with exchanger without RWP. It was found that the heat transfer coefficient increased by 28.1% to 43.9%,71.3% to 87.6% and 98.9% to 131% along with pressure drop penalty increase of 11.3% to 25.1%,54.4% to 72% and 88.8% to 121.4% respectively. Among these configurations the inline 1RWP obtained the best overall performance and inline 3 RWP was better than 7 RWP. The numerical results were also analyzed on the basis of synergy [11]. Numerical analyses by K.Thirumalai kannan, B.Senthil Kumar 2011, were carried out to study the heat transfer and flow in the plate-fin and tube heat exchangers with different shaped vortex generators mounted behind the tubes. The effects of different span angles a (α = 30°, 45° and 60°) are investigated in detail for the Reynolds number ranging from 500 to 2500. Numerical simulation was performed by computational fluid dynamics of the heat transfer and fluid flow. The results indicated that the triangle shaped winglet is able to generate longitudinal vortices and improve the heat transfer performance in the wake regions. The case of α = 45° provides the best heat transfer augmentation than rectangle shape winglet generator in case of inline tubes [12]. Abdulmajeed A. Ramadhan 2012, a numerical investigation of forced laminar flow heat transfer over a 3-rows oval-tube bank in staggered arrangement with rectangular longitudinal vortex generators (LVGs) placed behind each tube was carried out to study The effects of Reynolds number, the positions and angles of attack of rectangular VGs. The results showed increasing in the heat transfer and skin friction coefficient with the increasing of Re number and decreasing the relative distance of positions of LVGs. It has been observed that the overall Nuav number of three oval-tubes increases by 10–20.4% and by 10.4–27.7% with angles of 30o and 45o respectively, with increasing in the overall average of skin friction coefficient of three oval-tubes reached to 53% and 72% with two angles used respectively, in comparison with the case without VGs[2].

2. Computational Models and Numerical Methods

2.1 Computational Domain

The schematic diagram of fin-and-tube heat exchanger in staggered arrangement with LVG is shown in Figure 1. A pair of winglets with three different shape of cross –section area (Airfoil, Rectangle and Triangle) is punched out from the fin symmetrically behind every round tube to delay the separation of the boundary layer from the tube surface, and the wake region behind the tube may be reduced, even be eliminated by the LVG.All the shapes of LVG have equal cross –section area and characteristic length. Due to the symmetric arrangement, the region enclosed by the dashed lines in Figure 1 is selected as the computational element. The angles of attack α of LVG are varied by (30 and 45). This geometry can be specified by the parameters such as the tube diameter, D, the spacing between tubes, the location of vortex generator, the angle of attack, the length and area of the vortex generator. All geometric parameters for the configuration are selected similar to those employed by [4] see Table 1, while the table 2 show the nomenclature.

Table 1: The geometric parameters according to [4].

Parameter	*Symbol / unit*	*Size or value*
Length of computational domain	*L / mm*	216
Outside diameter of tube D	*D / mm*	32
Longitudinal tube pitch	*Pl / mm*	72
Transverse tube pitch	*Pt / mm*	64
Number of tube row	*N*	3
Length of LVG	*Lv / mm*	16
Area of LVG	*Av* / mm^2	25.6
Wall temperature	*Tw / k*	353
Inlet temperature of air	*To / k*	313
Frontal velocity	u_i */ m/sec*	3.195 – 5.02

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Table 2: The nomenclature

Cf skin friction coefficient
k : thermal conductivity (W/m K)
Nu : average Nusselt number= h D / k
P : pressure (Nm-2)
Re_D : Reynolds number based on the diameter of tube
u, v : velocity components (ms-1)
U, V : non-dimensional velocity components
W : height of the inflow and outflow openings
Gk : Turbulent production term
x, y : Cartesian coordinates (m)
LVG : Longitudinal vortex generator
Greek symbols
α: angle of attack (o)
ν : kinematic viscosity of the fluid (m2s-1)
ρ: density of the fluid (kgm-3)
τ : shear stress (Pa)
Cμ, Cε1, Cε2: Constants in k-ε equation
σ_k, σ_ε : Effective Prandtl
μ : dynamic viscosity, kg/m.s
μ_T : Turbulent viscosity, kg/m.s
ΔX streamwise distance (m)
ΔY spanwise distance (m)

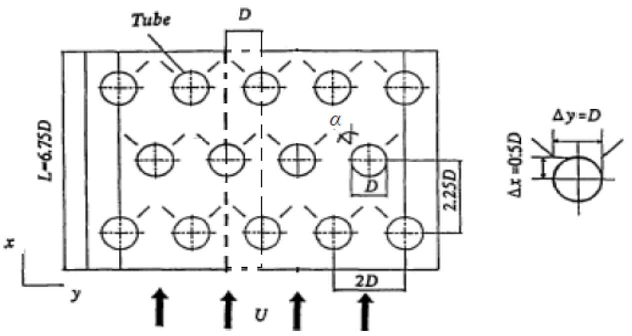

Figure 1: The schematic diagram of computational domain and location of LVG.

2.2 Governing Equations

The two-dimensional instantaneous governing equation of mass, momentum and energy equations for study incompressible in fully developed flow can be written in conservation form expressed in Cartesian coordinates as follows [13]:-

$$\frac{\partial u_i}{\partial x_i}=0 \quad (1)$$

$$\frac{\partial(\rho u_i u_j)}{\partial x_j}=-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\overline{\rho u_i' u_j'}\right] \quad (2)$$

$$\frac{\partial \rho u_i T}{\partial x_j}=\frac{\partial}{\partial x_j}\frac{k}{cp}\left(\frac{\partial T}{\partial x_j}\right) \quad (3)$$

The Reynolds stress tensor $-\overline{\rho u_i' u_j'}$ can be determined according to the Boussinesq assumption as

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

Where μt is the turbulent eddy viscosity and is estimated by the (k-ε) two equations turbulent model.

μt=cμρk2/ε (5)

The differential equation of k and ε are given as

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

$$\frac{\partial(\rho u_j \varepsilon)}{\partial x_j}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{\varepsilon 1}G_\varepsilon\frac{\varepsilon}{k}-C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k} \quad (7)$$

where $Gk=-\rho u_i' u_j'\left(\frac{\partial u_j}{\partial x_i}\right)$ is the turbulent production term.

The remaining coefficients that appeared in the above equation are as quoted by[13] : C_μ=0.09,$C_{\varepsilon 1}$=1.44, $C_{\varepsilon 2}$=1.92, σ_k=1 and σ_ε=1.3.

2.3 Boundary Conditions

The boundary conditions for this analysis are shown in figure 2:

- At the inlet: *U=u_i ; V=0; T= To*
- At the outlet: *P*=0
- At the Tube walls: *U*=0; *V*=0; T= *Tw*.
- At the rest of the adiabatic walls :*U*=*V*=0,$\frac{\partial T}{\partial X}$ =0, $\frac{\partial T}{\partial Y}$ = 0.
- At the top and bottom surfaces of the computational domain excluding the tube

surfaces, symmetry boundary condition is used. The mathematical form of this condition:

$\frac{\partial U}{\partial Y}$ = 0, *V*=0, $\frac{\partial T}{\partial Y}$ = 0

2.4 Parameter Definitions

The average Nusselt number and friction coefficient can be calculated as follows [2] :

$$Nu=\frac{1}{\pi D/2}\int_0^{180} Nu_L\frac{D}{2}d\theta$$

$$c_f=\frac{\tau_w}{0.5\rho U^2{}_m}$$

$$Re_D=\frac{u_i D}{\gamma}$$

Where (*θ*) is the angular displacement from the front stagnation, and *τ*w is the wall shear stress.

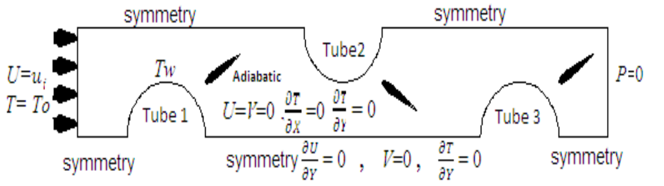

Figure 2: The computational domain and boundary conditions

2.5 Numerical Method

The above set of equations with the prescribed boundary conditions is solved by a computational fluid dynamics code (FLUENT). An unstructured quadrilateral mesh is generated. To improve the accuracy of the numerical results, the grid around the LVG and tubes is refined. The grid around the LVG and tube is shown in Figure 3. In the computation, the LVG regions are defined as solid in which the velocity is 0 and adiabatic. The convection terms in Eqs. (2) and (3) are by the second upwind scheme. The coupling between pressure and velocity is implemented by SIMPLEC algorithm. In order to ensure the accuracy of the obtained results, a grid study is performed with three resolutions (23778, 36576 and 58832 cells). The average Nusselt numbers and average skin friction coefficient are listed in Table (3**)**. As shown in this table, the difference between the Nusselt numbers and average skin friction coefficient for these three grids is small. By increasing the grid resolution, the number of iterations decreases slightly but the time convergence severely increases. Because a little difference between the results is found for the three resolutions, grid 1(23778) is also used for all simulations in this study.

Table 3: Grid independent test for the Airfoil winglet to the first raw at Re =7000.

	Grid 1 (23778)	Grid 2 (36576)	Grid 3 (58832
Nu	3432.306	3432.497	3432.714
Cf	0.062932	0.062777	0.062738

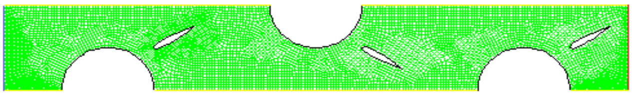

Figure 3: The numerical grid

3. Results and Discussion

Numerical study has been carried out for 2-D domain at Reynolds number (7000 ≤ Re ≤ 11000). Air has been used as the working fluid. Three shapes of winglet is used. The effects of turbulent are simulated by the (k-ε) turbulent model. The results of this present study are displayed in terms of velocity vectors and streamlines. Moreover, the effects of vortex generators on heat transfer and aerodynamic characteristic are shown in terms of average Nusselt number Nu, friction coefficient and pressure drop.

3.1 Velocity Vectors and Streamlines

Figures (4 and 5) represent the velocity vector and streamline plots of the flow over tubes with and without using winglet pairs respectively, at Re = 7000. At the baseline model velocity gradient near the solid boundary (Tubes) due to viscosity is visible; also, separation point on the tubes surface was visible. The flow over tubes with different shapes of winglet pairs is more different from the case of flow over the Tubes without using winglet pairs (free) especially near the rear of Tubes and the wake behind the Tubes. In general, the incoming flow in the case of flow over tubes together with winglet pairs is the same as the flow over tube without using winglet reaches to the center of tube, after the center of tube the difference will be showing, where separation occur in the free case but winglet pairs will limit the growth of boundary layer and delay or prevent the separation from the rear surface of tubes [the prime focus area for transport enhancement] in the case of using winglet. Figure 5 shows the streamline plots of flow field over the tubes at Re = 7000 with different shapes of winglet pairs. When incoming flow reaches the winglet, it will be changing its direction in to two ways, the first in tangential to the winglet toward the symmetry and the second direction is toward the rear of tubes and pass through the gap between the tube and the winglet. This procedure lead to high flow momentum on rear of tubes to limit the growth the boundary layer and heat transfer from this region of tube will be increasing. Also, the figure shows that there is no circulation (vortex) of flow behind tubes due to controlling the flow around tubes by winglet, also it can been seen that the Increasing in angle of attack lead to increasing pressure difference between the front surface facing the flow and suction surface of the winglet and then increasing the velocity of the fluid near the tube. When increasing the gap between the tube and the winglet, the velocity near the symmetry will be increasing, and due to low pressure behind the winglet, the flow moves from the wake region toward the winglet and due to high flow velocity in the gap between the winglet and the symmetry, the flow will change its direction and the vortex will be occur.

3.2 Heat Transfer and aerodynamics Characteristics

The effect of velocity on the average Nusselt number (Nu), Friction Coefficient (*Cf*) at Tubes wall, and pressure drop along the domain are displayed as a function of Reynolds number as shown in figures 6,7 and 8 respectively. Figure (6 A, B and C) Show the effect of Re on Nu for all shapes of winglet and display the comparison between the winglet shapes on Nu also, with the case of the flow over tubes without using winglets (Baseline) As expected, heat transfer will be increasing with increasing (Re) due to increasing the velocity (momentum) around tubes and increase the flow's capacity for convective heat. In general the results shows there is an enhancing in heat transfer when using any shape of winglet with respect of baseline. The triangle shape of winglet was appropriate shape for enhancing heat transfer because of accelerating the flow and controlled the fluid to flow on rear tubes, moreover, because of existing inclined edge, the flow will not restrict when reach the winglet and will pass over inclined edge as a swirl, the swirl will be blending the hot fluid (near tubes) with the incoming cold fluid. In the other hand, figure 7 show that the LVG increase the skin friction coefficient with increasing Reynolds number when the flow encounters the blockages created by the presence of LVG, and due to the interaction between the tubes and LVG placed which disturb the entire flow field. That is cause more friction than the case without LVG. Figure 8 shows the effect of Re on static pressure difference along the domain. As seen, by increasing the Reynolds numbers the momentum and secondary flow increases and it increases the pressure drop. In general when the winglet be near the tube and the gap be small, the static pressure along the domain will be falling, Increasing in angle of attack lead to increases the pressure different due to accelerate the flow through the domain.

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Impact Factor (2012): 3.358

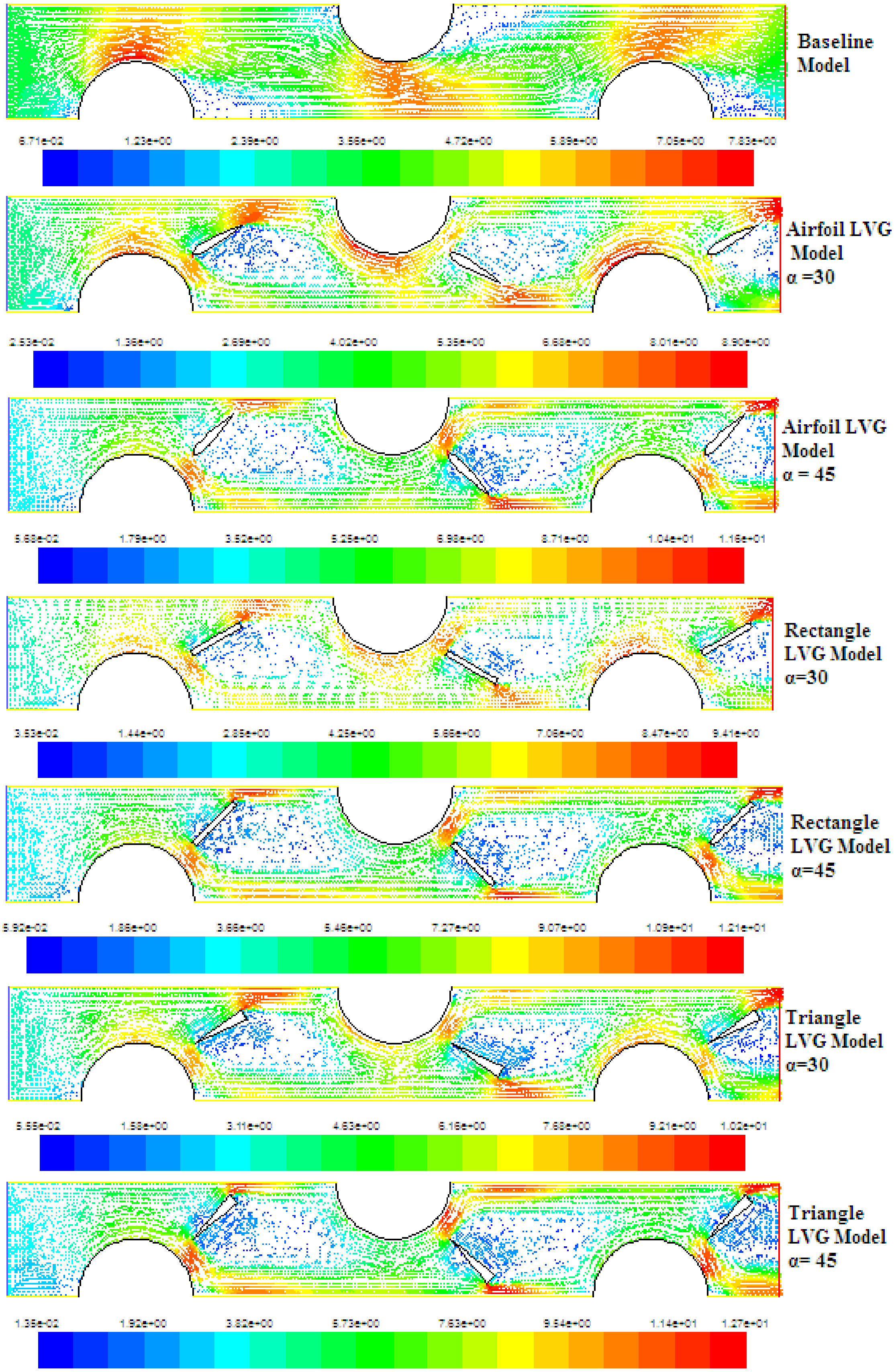


Figure 4 : Velocity distribution on the three rows of tube bank for baseline and modified models, for *Re* = 7000

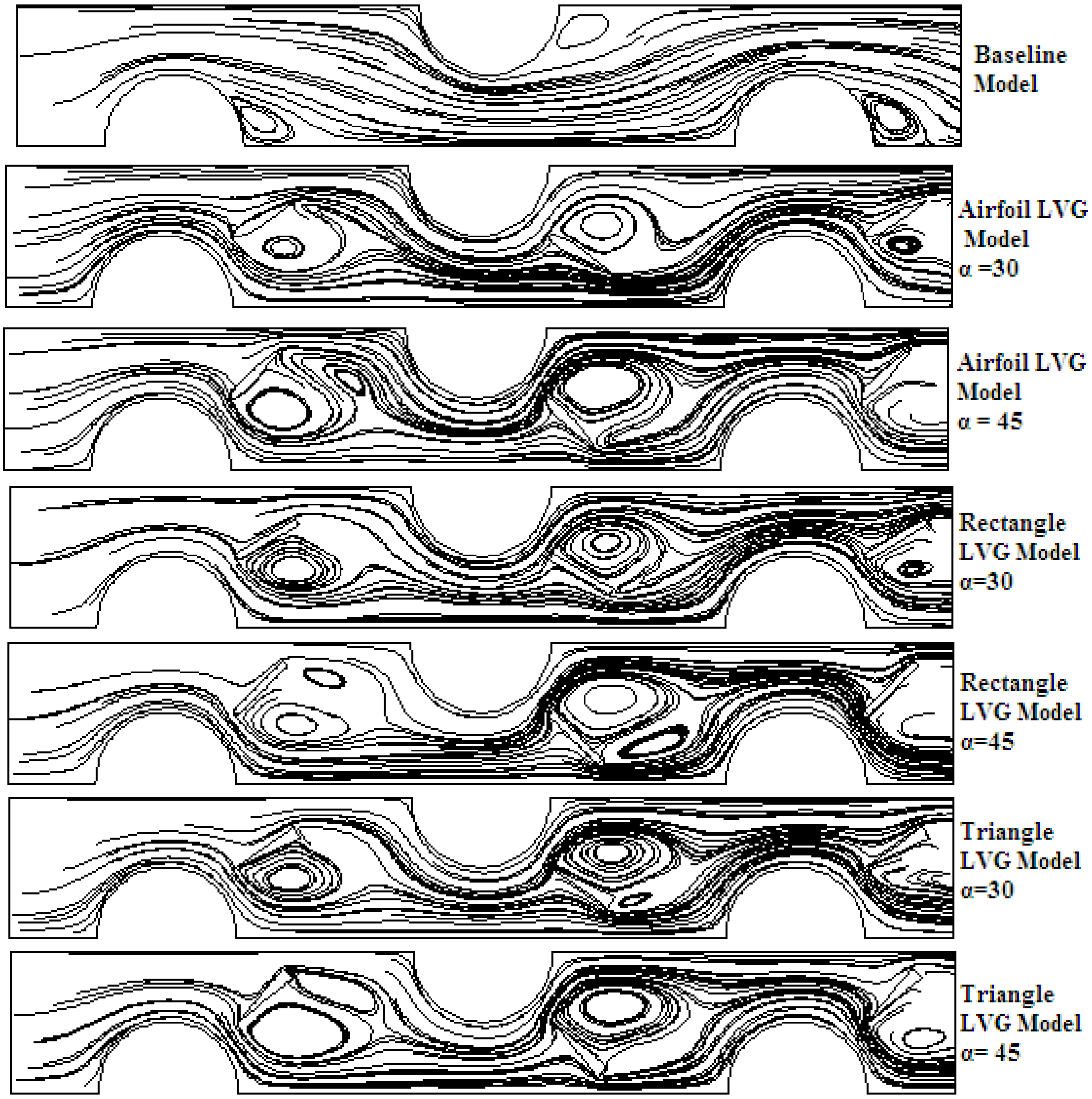
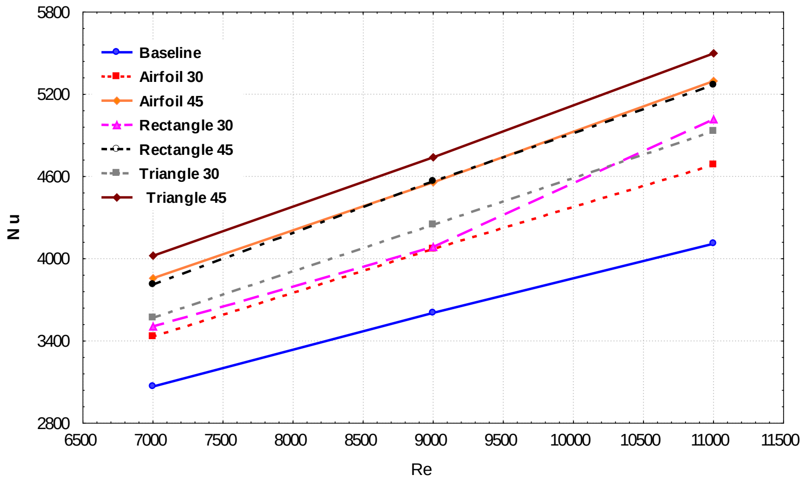
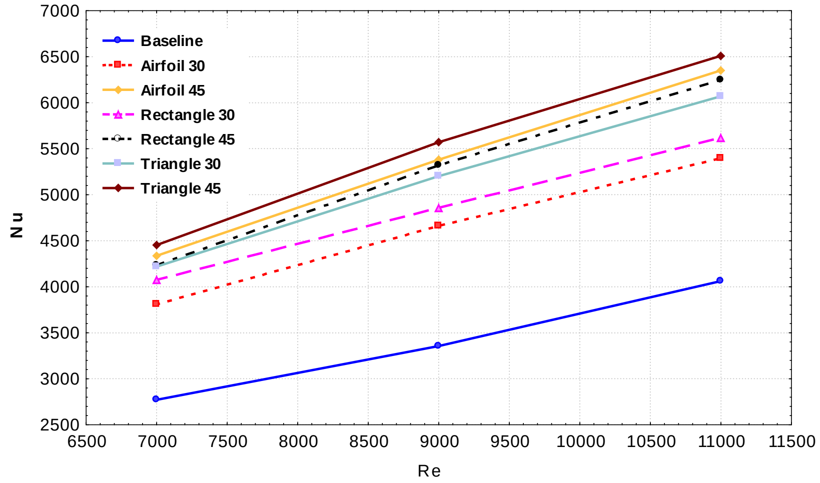


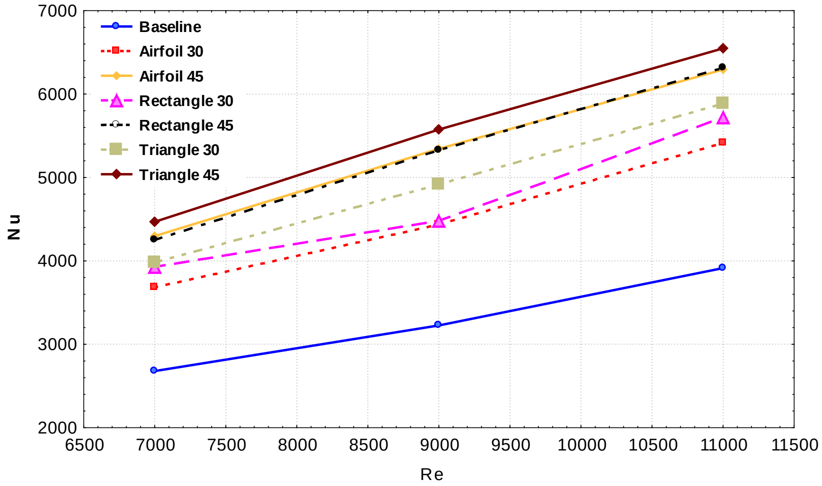
Figure 5: Streamlines on the three rows of tube bank for baseline and modified models, for *Re* = 7000.



A- First Row of Tubes

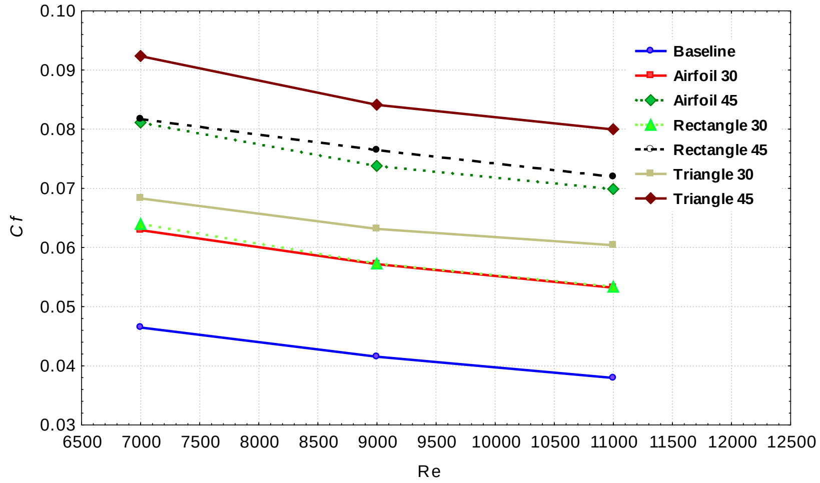


B- Second Row of Tubes

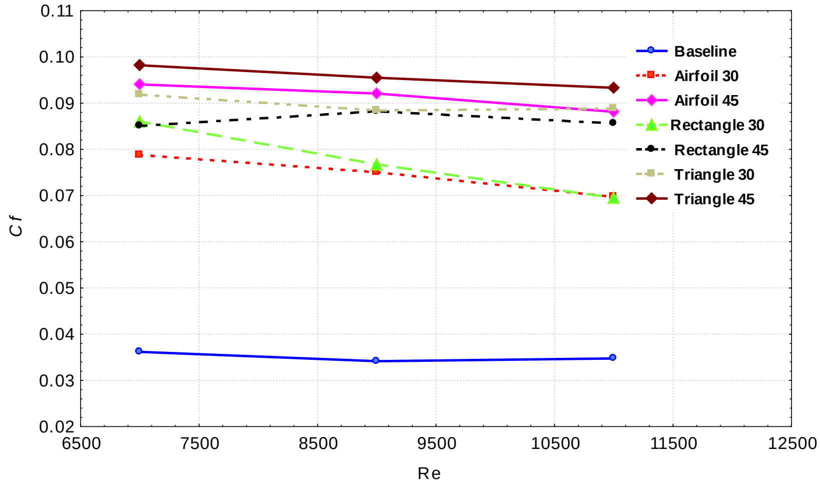


C - Third Row of Tubes

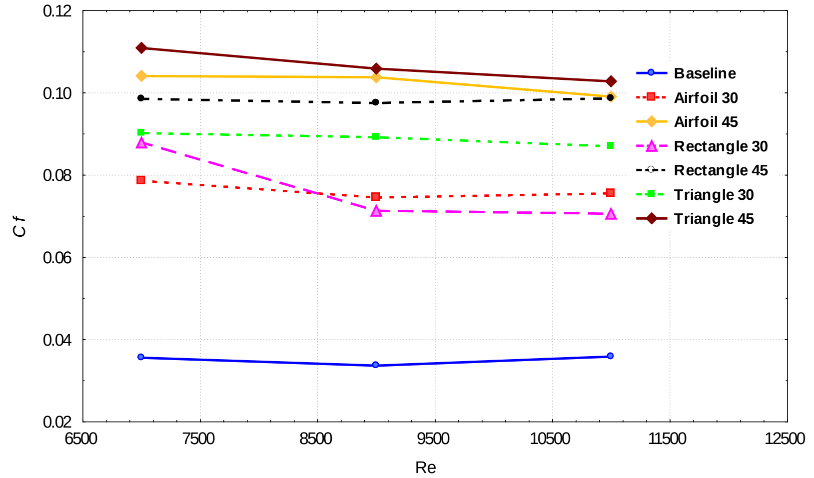
Figure 6: Variation of the average Nusselt number with Reynolds number.



A- First Row of Tubes



B- Second Row of Tubes



C- Third Row of Tubes

Figure 7: Variation of friction coefficient with Reynolds number.

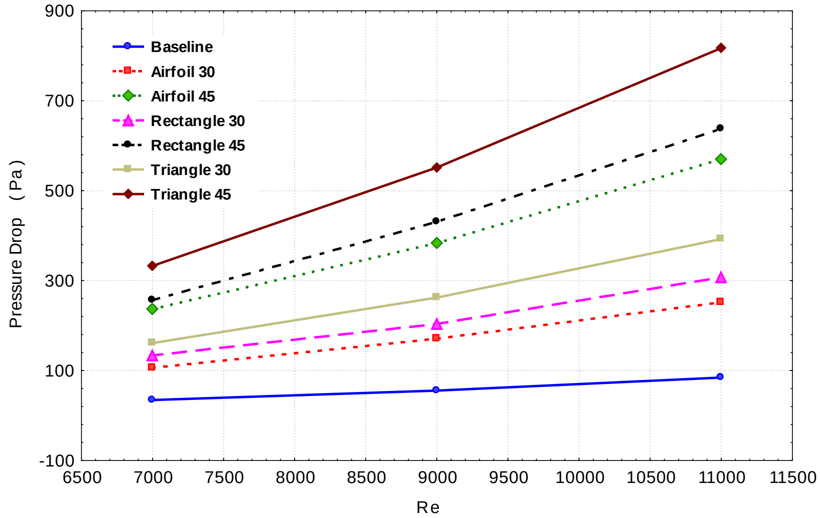


Figure 8: Variation of Pressure drop along the domain with Reynolds number

4. Conclusions

Numerical study for heat transfer around circular tube banks with using different shape of vortex generators (winglet pairs) in turbulent flow has been done. The following major conclusions may be drawn from the present study:

- There is an effect for the shapes and angle of attack of winglet on heat transfer and aerodynamic characteristic.
- The increasing in Nusselt number accompanied by increasing in pressure drop and the friction coefficient.

References

[1] Wisam Abed Kattea " An Experimental Study on the Effect of Shape and Location of Vortex Generators Ahead of a Heat Exchanger" Al-Khwarizmi Engineering Journal, Vol. 8, No. 2, PP 12- 29, 2012.
[2] Abdulmajeed A. Ramadhan “Numerical Study of Fluid Flow and Heat Transfer over a Bank of Oval-Tubes Heat Exchanger with Vortex Generators " Anbar Journal for Engineering ScienceS, Vol.5, No.1, PP 88-108, 2012.
[3] Sohankear, A. and Davidson, L. “Numerical study of heat and flow on a plat –fin heat exchanger with vortex generators” Begell house, Inc. Turbulence heat mass transfer, 2003.
[4] M. MIRZAEI AND A. SOHANKAR " HEAT TRANSFER AUGMENTATION IN PLATE FINNED TUBE HEAT EXCHANGERS WITH VORTEX GENERATORS: A COMPARISON OF ROUND AND FLAT TUBES " IJST, Transactions of Mechanical Engineering, Vol. 37, No. M1, pp 39-51, 2013.
[5] Sohal, M. S. and O,Brien, J. “Improving air cooled condenser performance using winglet and oval tube in a geothermal power plan” Geothermal resources council transaction, Vol 25, August PP 26-29, 2001.
[6] Biswas, G., Mitra, N.K. and Fiebig, M." Heat Transfer Enhancement in Fin-Tube Heat Exchangers by Winglet Type Vortex Generators “Int. J. Heat Mass Transfer 37(2), pp 283-291, 1994.
[7] Fiebig, M., Valencia, A. and Mitra, N. K"Local Heat Transfer and Flow Losses in Fin-and-Tube Heat Exchangers with Vortex Generators: A Comparison of Round and Flat Tubes" Exp.Thermal Fluid Sci. 8, pp 35-45,., 1994.
[8] Leu, J. S., Wu Y. H. and Jang, J.Y. " Heat transfer and fluid flow analysis in platefin and tube heat exchangers with a pair of block shape vortex generators" International Journal of Heat and Mass Transfer 47, pp 4327-4338, 2004.
[9] Joardar, A. and Jacobi, A.M " Heat Transfer Enhancement by Winglet-Type Vortex Generator Arrays in Compact Plain-Fin-And-Tube Heat Exchangers" International journal of refrigeration 3, pp 187–97, 2008.
[10] J.M. Wu and W.Q. Tao “Numerical study on laminar convection heat transfer in a rectangular channel with longitudinal vortex generator. Part A: Verification of field synergy principle. International Journal of Heat and Mass Transfer ". 51, pp 1179-1191, 2008.
[11] Chu, P., He, Y.L., Lei, Y.G., Tian, L.T. and Li, R. "Three-Dimensional Numerical Study on Fin-And-Oval-Tube Heat Exchanger with Longitudinal Vortex Generators" Applied Thermal Engineering 29, pp 859–876, 2009
[12] K.Thirumalai kannan, B.Senthil Kumar " HEAT TRANSFER AND FLUID FLOW ANALYSIS IN PLATE-FIN AND TUBE HEAT EXCHANGERS WITH DIFFERENT SHAPED VORTEX GENERATORS " International Journal of Soft Computing and Engineering (IJSCE) ISSN: 2231-2307, Vol 2, Issue-1, pp 357-361, March 2011.
[13] Ferzigen, J. H. and Peric, M. “Computational methods for fluid dynamic ".2th edition, Springer. Berlin, (1999)
[14] J.M. Wu, W.Q. Tao " Investigation on laminar convection heat transfer in fin-and-tube heat exchanger in aligned arrangement with longitudinal vortex generator from the viewpoint of field synergy principle " Applied Thermal Engineering 27, pp 2609–2617, (2007).

Author Profile

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