INTERNATIONAL JOURNAL OF MECHANICAL ENGINEERING AND TECHNOLOGY (IJMET)

ISSN 0976 – 6340 (Print)
ISSN 0976 – 6359 (Online)
Volume 5, Issue 12, December (2014), pp. 36-46
© IAEME: www.iaeme.com/IJMET.asp
Journal Impact Factor (2014): 7.5377 (Calculated by GISI)
www.jifactor.com

IJMET

©IAEME

A NUMERICAL STUDY OF HEAT TRANSFER AND FLUID FLOW IN A BANK OF TUBES WITH INTEGRAL WAKE SPLITTER

Mohammed Saad Kamel

Department of Mechanical Techniques, Al- Nassiriyah Technical Institute
Southern Technical University, Dhi –Qar/ Al-Nassiriyah, Iraq

ABSTRACT

The present work represents a two-dimensional steady state numerical investigation of heat transfer and hydrodynamic characteristics of air flow across a 3 rows of circular tube banks in triangular arrangement with triangular wake splitters placed on each tube with downstream direction (wake region). The effects of Reynolds number (from 5000 to 15000), the length of splitter are 0.5D, 1.0D, and 1.5D times of tube diameter. The study focuses on the Influence of the different parameters of splitters on heat transfer and fluid flow characteristics of three rows tube banks. The characteristics of total heat transfer per area for each tube with and without splitter and total pressure drop are studied numerically by the aid of the computational fluid dynamics (CFD) commercial code of FLUENT 6.3. Velocity vector and streamlines on the three rows tube bank for baseline and modified models are plotted. The results showed increasing in the heat transfer with the increasing of Reynolds number and that Increase in total heat transfer when the triangular wake splitter attachment as a result of extra surface area generated by the splitter plate it observed that about (45.14%, 45.67% and 64.65%) increasing in total heat transfer from tube1, tube2 and tube3 respectively with 1.5D chord at Reynolds number 15000 compared with tubes without splitter. Also, it observed that the reduction in total pressure drop of triangular wake splitter with length 1.5D at Reynolds 15000 about (9.79%) compared with tubes without splitter.

Keyword: Fluent-CFD, Tube Bank, Reynolds Number, Triangular Wake Splitter, Pressure Drop.

1. INTRODUCTION

In the Present study considers numerical work for flow over tubes bank with and without wake splitter (triangular splitter plate), And the effect of chord length on hydrodynamics and heat transfer characteristics. Flow over tube bank is a fundamental heat transfer problem which is of practical importance having large number of application. Applications which involve flow past cylinder include cross flow around rod bundles in heat exchangers of nuclear reactors, cooling of electronic equipments, air flow around cooling towers, flow past flame stabilizers in high speed combustion chamber, pipelines etc. However, as Reynolds number increases, flow begins to separate behind the cylinder causing symmetric wake. Wake is a region of recirculation flow behind a body caused due to flow separation. Attached and symmetric flow takes place for very low Reynolds number. Wake formation reduces convective heat transfer downstream because of low velocity recirculation. To overcome this problem, splitter plates are used downstream. Splitter plates are wake stabilizers and have been used as a means of controlling various aspects of wake formation and vortex shedding. Wake splitter is a rigid attachment to the body which alters shedding frequency and increases base pressure resulting in overall reduction of drag. Even though it reduces average heat transfer coefficient, it provides additional surface for convective heat transfer and increases overall heat transfer. Various configurations of splitter plates can be used for enhancement of heat transfer and controlling vortex shedding [1].

Many researchers tried to study the complex flow over cylinder and tube bank. Work has been carried out by Vivek Shrivastava and et.al [1]. Studied Effect of Triangular Wake Splitter on Flow and Heat Transfer over a Circular Cylinder for Various Chord Lengths numerically the results were found to be in good agreement with available experimental and numerical work. Heat transfer with triangular wake splitter has been found to be 17%, 53.4%, 115.7% more and drag coefficient 1.176, 7.92 and 9.01 times lower compared to bare cylinder for three different chord lengths. Performance of triangular wake splitter has been found to be similar to rectangular wake splitter. Results point towards cylinder with triangular wake splitter being more efficient than other configurations. Oosthuizen and et.al [2] on two dimensional square cylinder with splitter plates. Tiwari, and et.al [3] have worked on circular cylinder with splitter plate of different length and their effect on coefficient of pressure, local Nusselt number and overall heat transfer. Anderson and et.al [4] have worked on circular cylinder for near subcritical Reynolds number. Mahir and et.al [5] have worked on Convective heat transfer in unsteady flow past two cylinder in tandem arrangement with variation of L/D ratio(center to center distance ratio) Local Nusselt number , Strouhal number, flow parameters were studied for Re=100 and Re=200. Sudhakar and et.al [6] have carried out work on oscillating rectangular wake splitters and its effect on flow characteristics. Panchal and et.al [7] have worked on flow over square array of circular cylinders for L/D ratio (center to center distance) 1.25 for Re= 40, 50, 100, 150 & 200 for different prandlt number. Study on wake generation and change in local Nusselt number was studied Patnaik and et.al [8] have worked on heat transfer for laminar flow past circular cylinder with integral wake splitter. Chandra and et.al [9] has carried out study on. Natural convection Heat transfer from a heated semi circular cylinder for the various boundary condition ie constant temperature and constant heat flux for various Grshaoff number and prandtl number. Sparrow and Kang [10] have studied heat transfer and pressure drop characteristics on longitudinally finned tube banks. Roshko [11] studied vortex shedding suppression by using splitter plate in circular tube. Badami and et.al [12] have studied effect of rectangular and triangular splitter plates on cylinder and their effect on wake length, coefficient of pressure and coefficient of drag.

2. MODEL DESCRIPTION

2.1 Physical model

Top view of schematic diagram of a circular-tube heat exchanger in triangular arrangement with IWS is shown in Figure (1). A splitter of triangular shape is putted as a fin surface symmetrically on each circular tube. Due to the symmetric arrangement, the region occupied by dashed lines is selected as the computational domain as shown in figure (2), which is, considered as a channel of height H = D = 32 mm and length L = 10D =320 mm. The tube rows are arranged in a triangular formation with the transverse pitch-to-diameter ratio 2.0 and longitudinal pitch-diameter ratio 2.25. The splitter plate length-to-tube diameter ratio was varied 0 to 1.5 in the increment of 0.5. The splitter plate thickness was 1/5 the tube diameter

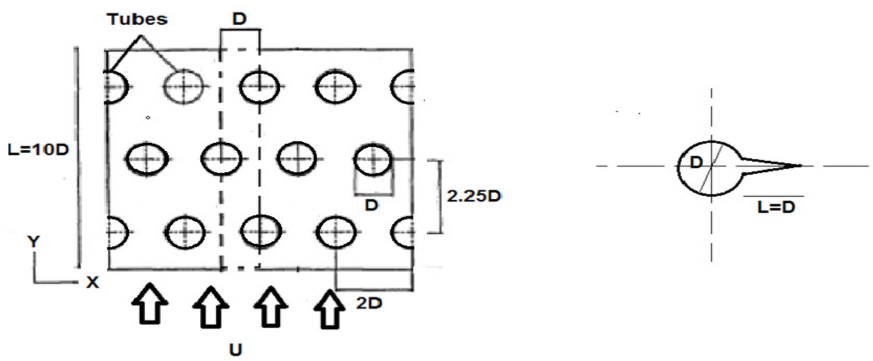

Figure (1) Schematic diagram of the physical problem

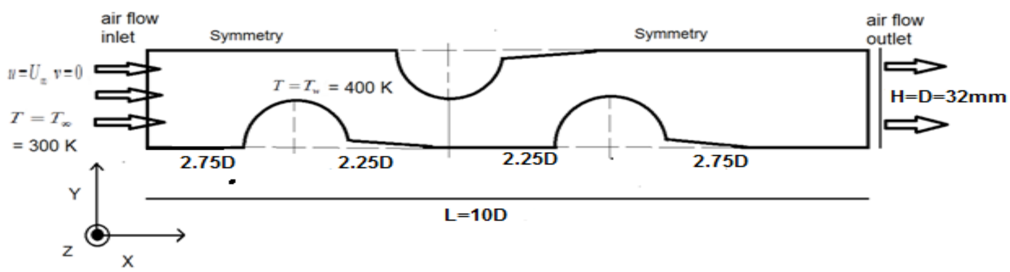

Figure (2): Computational domain

2.2 Governing equations

The arrangement of circular tube banks in staggered with cross airflow. The wall of the circular tube is heated under constant temperature, and air inlet at variable velocity V. force convection heat transfer between heated circular tube surface with and without splitters and inlet airflow in a horizontal x-y plane. 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]:

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

$$\frac{\partial(\rho u_i u_{j)}}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}[\mu(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}) - \overline{\rho u_i' u_j'}] \qquad \ldots\ldots (2)$$

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

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

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

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

μ_t= cμρk2/ε .... (5)

The differential equation of k and ε are given as

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

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

-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] : Cμ=0.09 ,$C\varepsilon_1$=1.44 , $C\varepsilon_2$=1.92 , σ_k=1 and σ_ε=1.3

2.3 Parameter definition

The Reynolds number Re is defined as:

$$\text{Re} = \rho \times U_\infty \times D/\mu \qquad \text{........... (1)}$$

Where U_∞ is the mean velocity inlet in the minimum flow cross-section of the flow channel, and D is the diameter which equals to 0.5St.

$$Q = hA(\Delta T) \qquad \text{.......... (2)}$$

Where (h) is the average heat transfer (A) surface area and ΔT is the temperature difference.

2.4 Boundary conditions

The boundary conditions for this analysis are:

- At the inlet Chanel the fluid is assumed to enter with a uniform horizontal velocity U_∞ and temperature (T_∞) $U=U_\infty$, $T=T_\infty$=300K, V=0;
- At the outlet Chanel: P=0

- Symmetry condition: For 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, $\frac{\partial T}{\partial y}=0$,
- Symmetry condition circular tube wall surface: U = V = 0, T_w = constant = 400 K.
- Triangular IWS wall surface: U = V = 0; T_w = constant = 400 K.

3. NUMERICAL METHOD

In this present work (Fluent-CFD) software used to solve equations for conservation of mass, momentum, and energy using a finite volume technique to show dynamic flow and heat transfer around circular tubes, The model geometry and mesh generation are build by (Gambit), Version 2.2.30 as a show in Figure (3). The grid is made up of triangular elements to improve the quality of the numerical prediction near the curved surfaces. The coupling between pressure and velocity is implemented by SIMPLEC algorithm. To reduce numerical errors, second order upwind discritization schemes are used in the calculations. Each computational iteration is solved implicitly. The convergence of the computational solution is determined on scaled residuals for the continuity, energy equations and for many of the predicted variables. More than 400 iterations are generally needed for convergence.

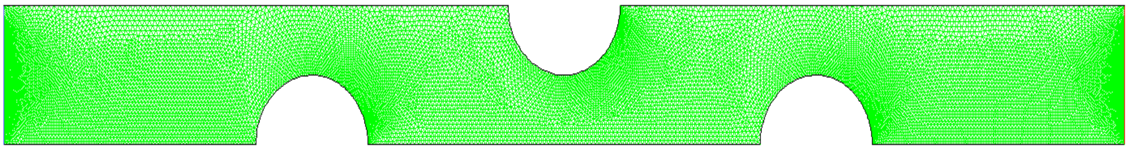

Figure (3): the computational grid in gambit 2.3.1

4. RESULTS AND DISCUSSION

Two-dimensional forced convection is studied for turbulent flow in a bank of tubes with and without wake splitter with a Prandtl number of 0.71. The controlling parameter, for the cases configurations of the geometry as defined in the table (1). The range of Reynolds number used for the simulations is $5000 \leq Re \leq 15000$. The results of this present study are displayed in terms of streamlines and velocity vector. Moreover, the effects of heat transfer per area of the tubes and the pressure drop also shown.

Table 1: The configurations of the geometry for Study Cases

Case No.	Case name	Length of splitter
Case 1	Baseline(without splitter)	0D
Case 2	Triangular splitter	0.5D
Case 3	Triangular splitter	1D
Case 4	Triangular splitter	1.5D

4.1 The grid sensitivity study

The sensitivity of the numerical results can be seen from table (2).The difference in total pressure drop at the inlet and 9.25D from computational domain are listed in it. From the table it can be clearly seen that the maximum different of the pressure drop is less than 2% for three different grid systems. For the present study, the final grid number is selected as about (25552) cells. Similar validations are also conducted for other cases.

Table 2: Grid independent test

parameter	Grid Size/cell		Diff %	Grid Size/cell	Diff %
	16236	25552		34300	
Pressure drop (pa)	**13.871**	**13.382**	**3.654**	**13.135**	**1.880**

%Diff = [(grid1)-(grid2)]*100/(grid2)

4.2 Effect of total pressure drop

The figure (4) illustrates the difference in total pressure at inlet and 9.25D from length of computational domain. The energy loss is evaluated by calculating pressure drop along section. It can be seen that the total pressure drop increase as the Reynolds number increases for all cases because the pressure loss due to friction is a function of the mean velocity of the flow that's mean when velocity increase friction increase and loss due to friction increase. The increasing in triangular splitter plat length led to reducing in pressure drop exceptional of L/D = 0.5D at Reynolds 150000 and 1.5D at Reynolds 5000 compared with baseline (without splitter) that Because the larger pressure drag in the larger separation region, when added triangular wake splitter that will serve to reduce shedding vortices, also the triangular Wake splitter arranged fluid flow makes vortices region separated. The area of vortex region reduces which made the pressure has reduced.

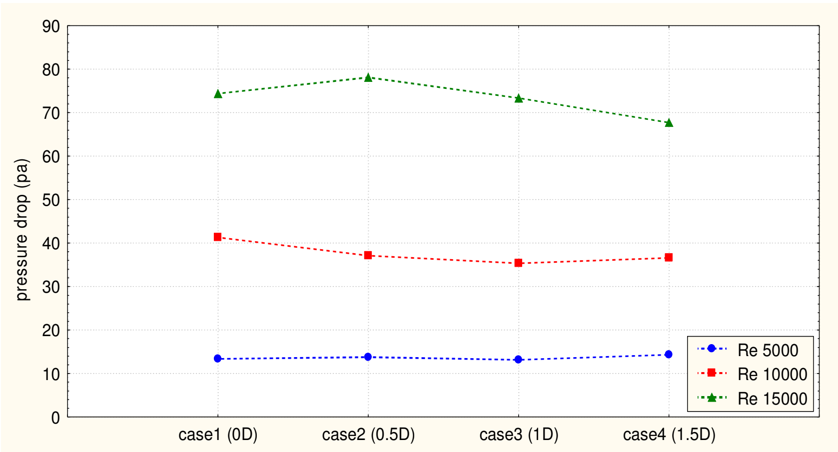

Figure (4): total pressure drop for various L/D and Reynolds number.

4.3 Heat transfer from tube banks with splitter plate

From figure (5) it can be observed that Increase in total heat transfer when the triangular wake splitter attachment as a result of extra surface area generated by the splitter plate. For all tube banks, The flow starts to create the vortexes behind the tube when going to high Reynolds number (Re>5000).the movement of fluid around the surfaces heat transfer will be fast at high Reynolds number, which makes average heat transfer Coefficient increase with increasing fluid flow. Because the temperature difference between the fluid and heat transfer surface is becoming high. However, according to figure(), the total heat transfer rate per area improves since the splitter plates also act as fins, which increases the surface contact area for better overall heat transfer process. Heat transfer characteristics were studied for tubes with different L/D ratios under constant temperature conditions. Tube banks with L/D = 1.5 yielded the highest heat transfer rates.

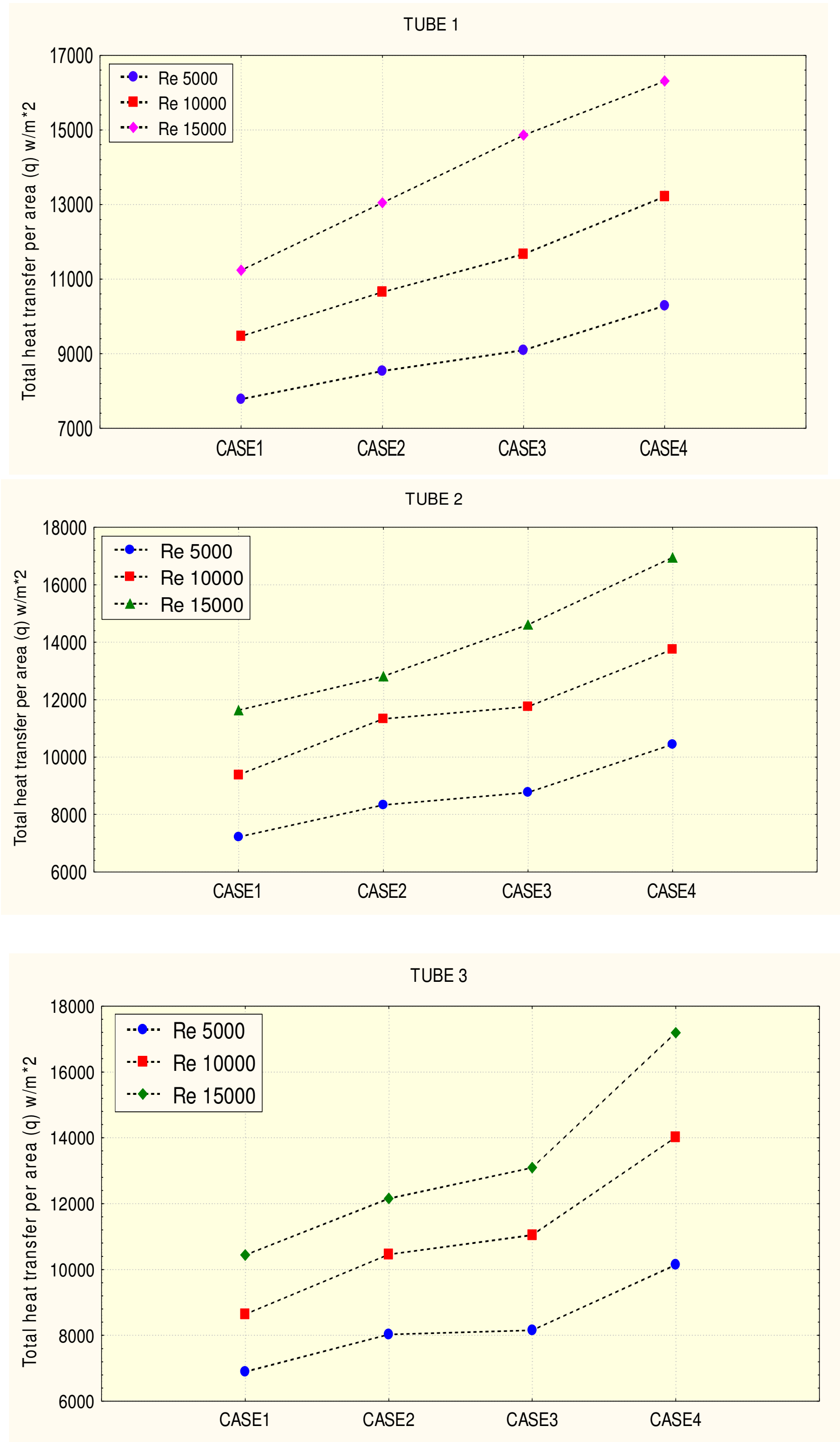

Figure (5): Total heat transfer per area (q) from tubes for all cases with Reynolds number

4.4 Flow Characteristics and Velocity Vector

From figure (6), (7) it can be clearly seen that the comparison of velocity streamlines and velocity vector for tube without splitter (baseline model) and tube with triangular wake splitter (modified model) of length 0.5D, 1D and 1.5D.fluent 6.3.26 has been used to obtain streamlines. For low Reynolds number, flow remains attached to surface. Flow behaves as a potential flow dominated by viscous force and doesn't separate. As Reynolds number increases, inertial forces become strong enough to overcome viscous force and flow separates. This can be seen from recirculation wakes formed for higher Reynolds number. With increase in Reynolds number, length of wake increases till attainment of critical Reynolds number which is in good agreement with linear stability theory. Beyond this, Von-Karman vortex shedding was observed. In this study velocity vector plotted at Re 10000 and streamline plotted at Reynolds number 10000 also, and that observed the wake region in baseline model and it is clear and vortex shedding occur also, at same Reynolds number with different chord length of triangular splitter also plot and that was clear there is reducing in wake region. The main purpose of using wake splitter is to delay flow separation and the onset von Karman vortex shedding and reduce flow induced vibration and this can be clearly seen in the plotted cases. 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 length of triangular wake splitter is more different from the case of flow over the Tubes without using wake splitter (free) especially near the rear of Tubes and the wake behind the Tubes. Splitter plates are used as passive means of controlling vortex formation and vortex shedding in the wake. It has been observed that with increase in Reynolds number, the length of wake increases for steady symmetric flow for the entire above mentioned configuration. It is found that wake length for tubes without splitter for all Reynolds number to be more when compared with triangular wake splitter. It is observed that the wake length for triangular wake splitter was reduced compared with baseline.

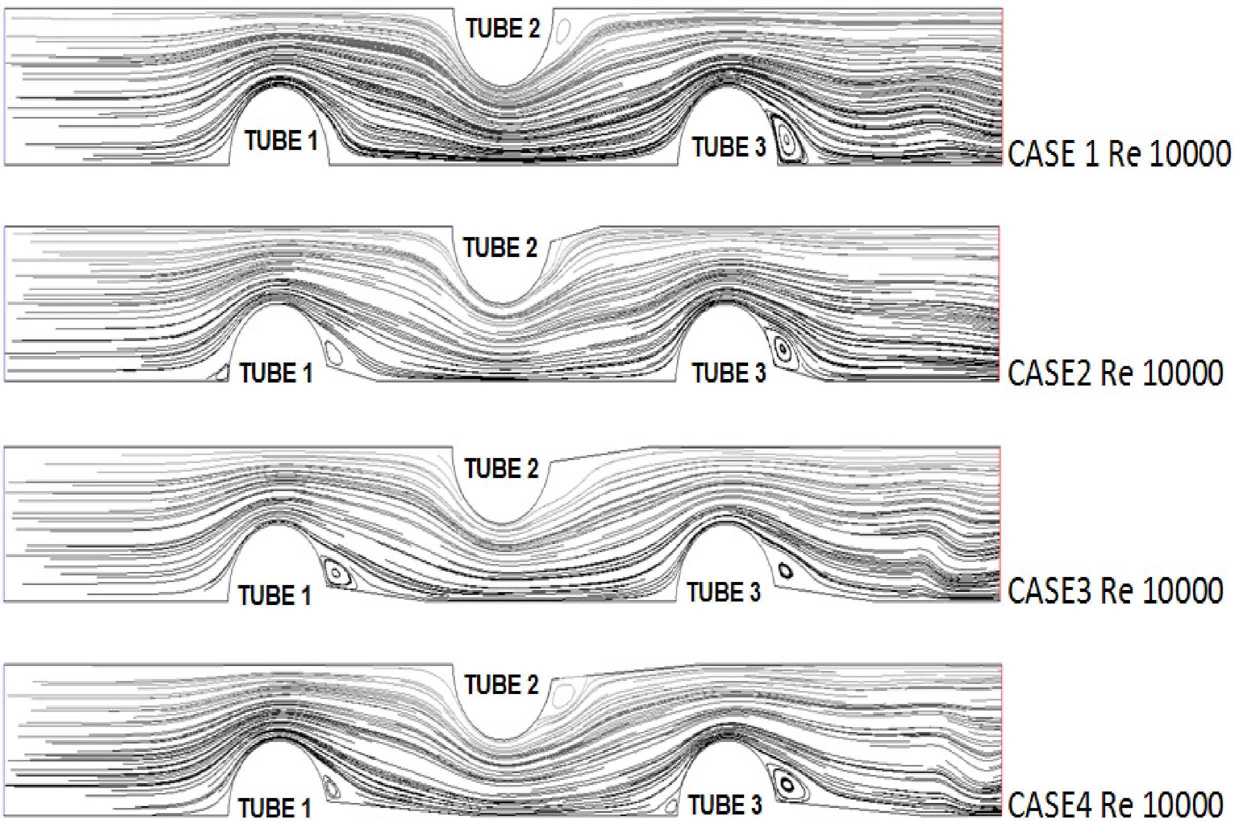

Figure (6): Streamlines on the three rows of tube bank with and without models for Re = 10000

International Journal of Mechanical Engineering and Technology (IJMET), ISSN 0976 – 6340(Print), ISSN 0976 – 6359(Online), Volume 5, Issue 12, December (2014), pp. 36-46 © IAEME

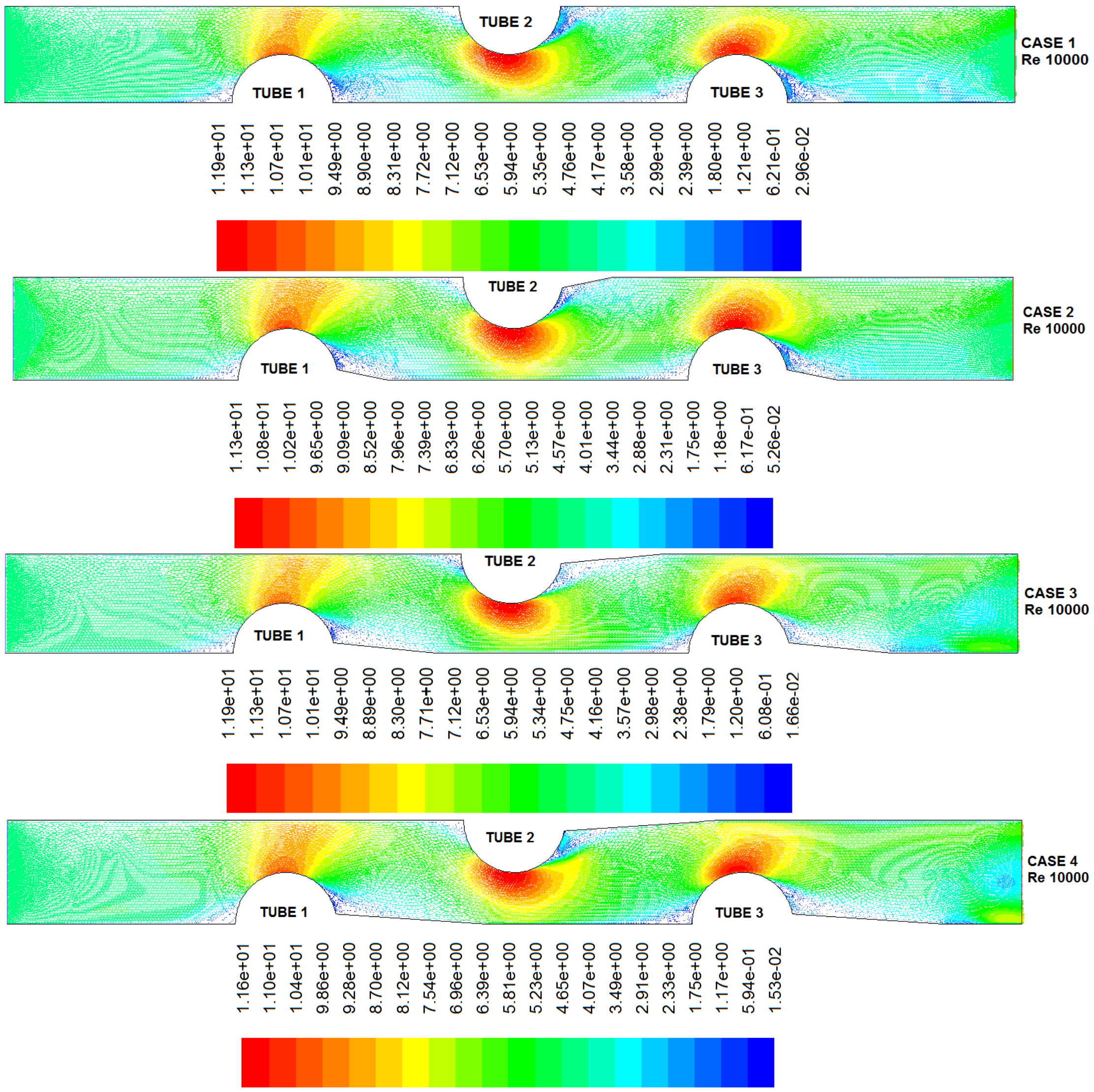

Figure (7): velocity vector colored by velocity magnitude at Re 10000 for all cases

5. CONCLUSIONS

Numerical analysis has been performed for flow over tube bank with and without wake splitter using (fluent) version 6.3.26. Results of total heat transfer per area have been consolidated and it can be concluded that total heat transfer per area for tube1, tube2 and tube3 with triangular splitter at length 1.5D are (45.14%, 45.67% and 64.65%) greater than tubes without splitter. This makes tubes with triangular splitter best possible configuration for optimum heat transfer also, it was observed that the reduction in total pressure drop of triangular wake splitter with length 1.5D at Reynolds 15000 about (9.79%) compared with tubes without splitter. Vortex shedding and pressure drop mostly depends on Reynolds number and tube spacing. Addition splitter plate on a tube can be attributed to the attenuation of vortex shedding in the wake and also reduced pressure drag significantly. Increase in total heat transfer can be also observed as a result of extra surface area generated by the splitter plate.

THE NOMENCLATURE

D : circular tube diameter (m)

S_L : longitudinal pitch
S_T : Transverse pitch
L : length of computational domain (m)
P : pressure (Nm-2)
Gk : Turbulent production term
Pr : Prandtl number
Re_D : Reynolds number based on the diameter of tube
T_∞ : air temperature k
T_w: wall temperature k
U_∞: air velocity at inlet (ms-1)
u, v : velocity components (ms-1)
U, V : non-dimensional velocity components
W : height of the inflow and outflow openings
x, y : Cartesian coordinates (m)
IWS : Integral wake splitter
H : height of channel (m)
L/D : Length to diameter ratio of splitter

Greek symbols

α: angle of attack (o)
v : 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

REFERENCES

[1] Vivek Shrivastava, Pavan Badami, Saravanan V, K N Seetharamu, "Effect of Triangular Wake Splitter on Flow and Heat Transfer over a Circular Cylinder for Various Chord Lengths", International Journal of Scientific & Engineering Research, Volume 5, Issue 4, April-2014 554 ISSN 2229-5518.
[2] Mansingh, Vivek and P.H Oosthuizen. "Effects of Splitter Plates on the Wake Flow behind a BluffBody," AIAA Journal: 778-783 (May 1990).
[3] S. Tiwari, D. Chakraborty, G. Biswas, P.K. Panigrahi ‘‘Numerical prediction of flow and heat transfer in a channel in the presence of a built-in circular tube with and without an integral splitter’’, international journal of heat and mass transfer, volume 48 issue 2, January 2005.
[4] Anderson, E. A. & szewczyk, A.A 1995 "vortex shedding from a straight and tapered circular cylinder in uniform and shear flow" proc sixth Intl conf. on flow induced vibration.
[5] Mahir A, and zekeriya A. "Numerical investigation of convective heat transfer in unsteady flow past two cylinders in tandem arrangements”, Int. J. of Heat and Fluid Flow 29 (2008) 1309-1318.
[6] Y. Sudhakar, S. Vengadesan, ‘‘vortex shedding characteristics of circular cylinder with oscillatingwake splitter plate’’, computers and fluid, volume 53, 15 January 2012.
[7] Panchal, Lakdawala, ‘‘Numerical investigation of thermal performance in cross flow around square array of circular cylinders’’, NUiCONE – 2011.
[8] B.S.V.P.Patnaik, K N Seetharamu, P.AaswathaNarayana, ‘‘simulation of laminar confined flow past, a circular cylinder with integral wake splitter involving heat transfer’’, Int.J.Numer.Method Heat flow Fluid Flow 6 (1996).
[9] Chandra A., Chhabra R.P., “Flow over and forced convection heat transfer in Newtonian fluids from asemi-circular cylinder”, Int. J. of H. and M. T. 54 (2011) 225-241.
[10] E.M. Sparrow, S.S. Kang, Longitudinally-finned cross flow tube banks and their heat transfer an pressure drop characteristics, Int. J. Heat Mass Transfer 28 (1985) 339–350.

[11] A. Roshko, on the drag and shedding frequency of two dimensional bluff bodies, Technical Report TN 3169, NACA, 1954.

[12] Vivek Shrivastava, Pavan Badami, Nandeesh Hiremath, Saravanan V, K N Seetharamu, 'Numerical analysis of flow past circular cylinder with triangular and rectangular wake splitter', ICMAE, WASET, 2012.

[13] Ferzigen, J. H. and Peric, M. "Computational methods for fluid dynamic". 2th edition, Springer. Berlin, (1999).