Heat Transfer and Pressure Drop Investigation for a Bank of Inclined Flat Tubes

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Abstract: A study was performed to investigate the average Nusselt number and pressure drop for a bank of inclined flat tubes. The tubes are arranged in rows, each one is inclined to a reversed direction with respect to its neighbor rows. The angle of inclination varied from zero to 90 for two values of R; 2.5/7 and 3/7. A numerical model was prepared, supported with the necessary defined functions, and verified to simulate the different studied cases. The results showed that, when increasing the inclination angle, almost within the first 20^o the nusselt number and the non-dimensional pressure drop increased slightly, then, a considerable increase is observed within the range from 20 to about 80, and, increasing the angle beyond that exhibited little increase in both values. Considering the ratio between the percentages of increase for both the Nusselt and the non-dimensional pressure drop, it is concluded that, the most economical conditions are included in two cases; the first is when R=2.5/7, Re = 12830, with inclination angle of 30o, and the second is when R=3/7, Re = 12830, with inclination angle of 50^o.

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Keywords: Nusselt number, flat tube, and aspect ratios

1. Introduction:

Energy and reduction of its losses is the field of interest for many researchers, because of its wide scope of application. To achieve the goal of energy economization, many industrial applications use heat exchangers that have to be sized according to space availability. One of the popular heat exchanger designs is that which contains a tube Bundle which carry one of the fluids and the other fluid moves in a direction perpendicular to the tube axes. Tubular heat exchangers are used in many energy conversion and chemical reaction systems ranging from nuclear reactors to refinery condensers. The most important design variables of tubular heat exchangers are the external surface heat transfer coefficient of the tube and the pressure drop of the fluid flowing externally. Based on previous studies reported in the literature, the effects of tube shape and arrangement have indicated that they could have a positive influence on heat transfer. Flat tube heat exchangers are expected to have lower air-side pressure drop and better air-side heat transfer coefficients compared to circular tube heat exchangers. The pressure drop is expected to be lower than that for circular tubes because of a smaller wake area. For the same reason, vibration and noise is expected to be less in flat tube. Many researchers investigated the flow and heat for circular and flat tube banks; Prediction of turbulent Flow in a Staggered Tube Bundle was introduced by Watterson et al. (1999). Modeling of heat transfer for flow across tube banks was presented by Wilson et al. (2000). Flow field characteristics for flow past a circular tube confined in a narrow channel was presented by Tiwari et al.(2006). A finite volume method on general surfaces and its error estimates was introduced by Ju Lili et al. (2009). Comparison of heat transfer conditions in tube bundle cross-flow for different tube shapes was investigated by Horvat et al. (2006). Bahaidarah (2004) used a finite volume based FORTRAN code to investigate the steady laminar two-dimensional incompressible flow over both, in-line and staggered flat tube bundles used in heat exchanger applications. Sundaresan (2003) developed a numerical model to predict the overall performance of an advanced high temperature heat exchanger. Analysis of laminar forced convection of air for cross flow in banks of staggered tubes is studied by Wang et al. (2000). Also, a steady flow over staggered tube bank, have been presented by Wang (2004). Zeinab S. A (2011), investigated the average Nusselt number and pressure drop for a circular and flat tube bank, with different aspect ratios. Sunil Lakshmipathy (2005), investigated the capability of PANS (Partially Averaged Navier-Stokes Simulation) model over a wide range of Reynolds numbers and flow physics for a turbulent flow past a circular cylinder, at ReD 140,000 and ReD 3900. In the present work, the average Nusselt number and pressure drop were investigated for a bank of flat tubes which are inclined. The inclination angle α ranged from zero to ninety degrees. Two aspect ratios, ((H/Dh) / (L/Dh)); 2.5/7 and 3/7, were chosen, each of them was tested at two Reynolds numbers; 7850 and 12830. The four cases are illustrated in table 1.

Table 1. Description of the four cases

Case number	Reynolds number	Aspect ratio	Angle
1	7850	2.5 / 7	From zero to ninety degrees
2	7850	3 / 7	
3	12830	2.5 / 7	
4	12830	3 / 7	

Numerical Model

The present simulation was performed using ANSIS - V13 – 64, to solve the following transport equations:

Continuity: $\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0$ (1)

The momentum equation,

$$\frac{\partial}{\partial t}\rho\vec{u}+\nabla(\rho\vec{u}\vec{u})=-\nabla p+\nabla.(\bar{\bar{\tau}})+\rho\vec{g}+\vec{F} \quad (2)$$

where, $\vec{g}$ is the gravity vector

The RNG-based *k*- ϵ, (renormalization group) with enhanced wall function:

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_i}(\alpha_k\mu_{eff}\frac{\partial k}{\partial x_j})+G_k+G_b-\rho\varepsilon+Y_M+S_k \quad (3)$$

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_i}(\alpha_\varepsilon\mu_{eff}\frac{\partial\varepsilon}{\partial x_j})+C_{2\varepsilon}\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon+R_\varepsilon \quad (4)$$

Energy:

$$\frac{\partial(\rho E)}{\partial t}+\nabla.(v(\rho E+p))=-\nabla(\sum_j h_j J_j)+S_h \quad (5)$$

Where, G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients, G_b is the generation of turbulence kinetic energy due to buoyancy, Y_M represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate, α_K and α_ε are the inverse effective Prandtl numbers for *k* and ϵ, respectively. S_k and α_ε are source terms, and J is mass flux component. Necessary user defined functions were written to calculate the following:

$$T_{bo} = \frac{\int \rho u_o C_p T_o dA}{\int \rho u_o C_p dA}, K \quad (6)$$

$$T_{av} = (T_{bo} + T_{in}) / 2 \quad (7)$$

$$P)_{\text{at any cross section}} = \frac{\int p\, dA}{\int dA} \quad (8)$$

$$Q_{tot} = \rho\, u\, A_i\, C_p\, (T_{bo} – T_i) \quad (9)$$

$$U = Q_{tot} / (n * A_s * (T_s – T_{av})) \quad (10)$$

$$Nu = U\, D_h / K \quad (11)$$

$$Re = \rho\, u\, D_h / \mu \quad (12)$$

$$dp_{av} = \left(\frac{\int p_o dA_o}{\int dA_o} - \frac{\int p_i dA_i}{\int dA_i} \right) / \rho u^2 \quad (13)$$

Preliminary runs were performed on both cases to estimate the optimum grid designs. For aspect ratio 2.5/7, a grid with 220 X 78 nodes was used, 34 nodes for each one of the flat tube sides, and, 28 nodes for each one of the curved tube sides. some researchers simulated the tube bank cases using a periodic domain technique. This technique assumes a similarity of pressure drop and non-dimensional temperature pattern across the periodic domain, depending on the repeated periodic geometry. This assumption is to some extent logic, but not accurate. So, in the present work, a group of four successive tube rows were simulated to approach the practical conditions. Figure.1 illustrates the domain grid for R=3/7 and α = 60 ^{O}

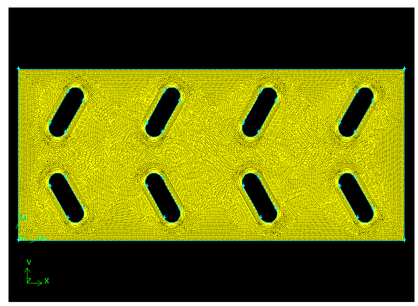


Figure 1. The grid for R=3/7 and α = 60^O.

Model Validation

For the purpose of model validation, the case of aspect ratio 2/5 was chosen and tested at Reynolds numbers which ranged from 2000 to 35000, and the results were compared with three previous research works. Figure.2 illustrates the comparison of the present work with the three previous research works.

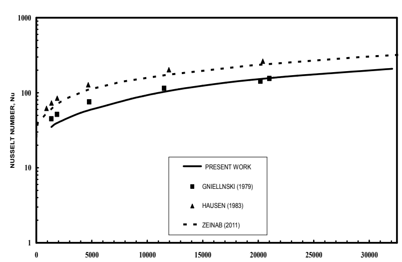


Figure.2 Variation of Average Nusselt Number With Reynolds Number.

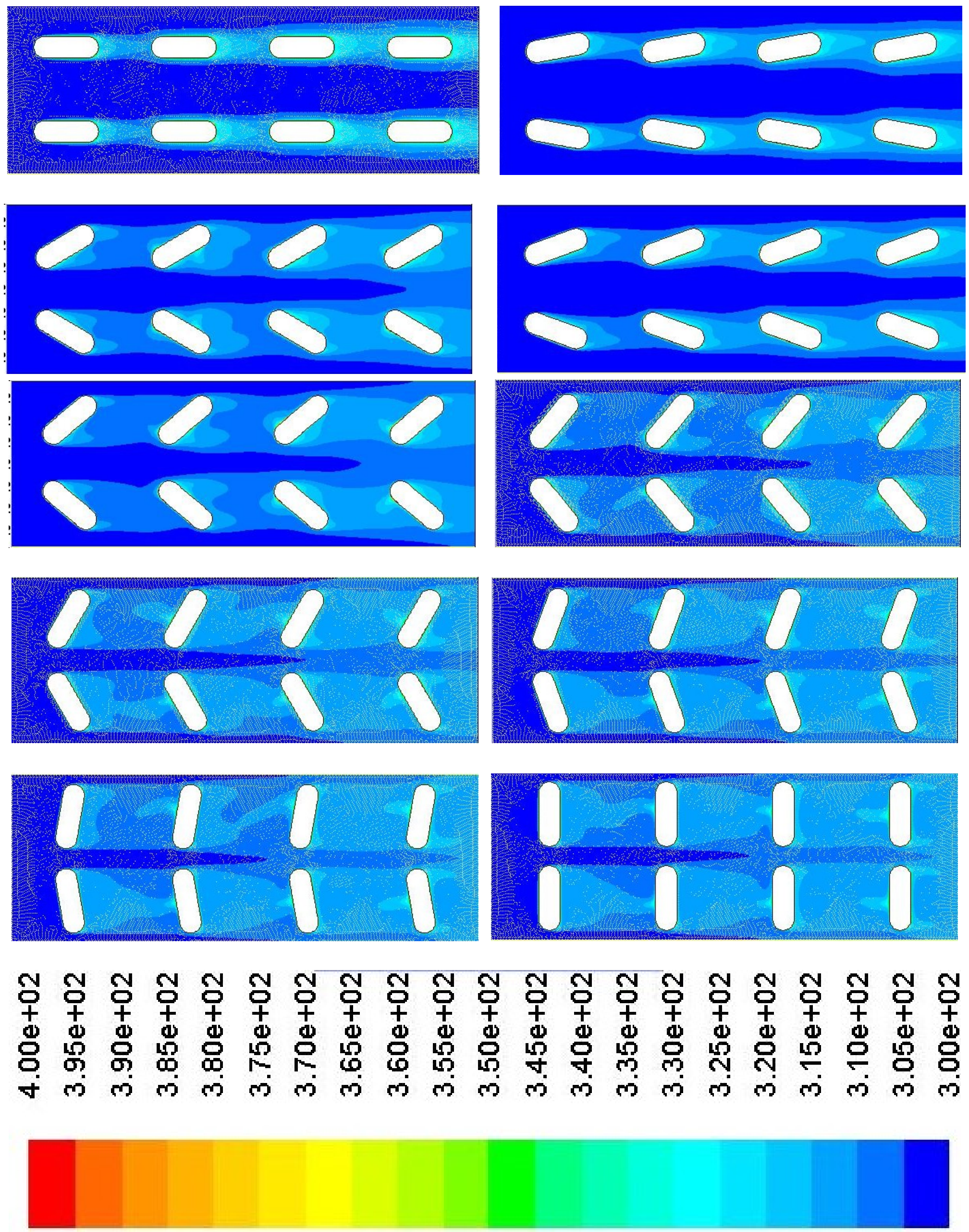
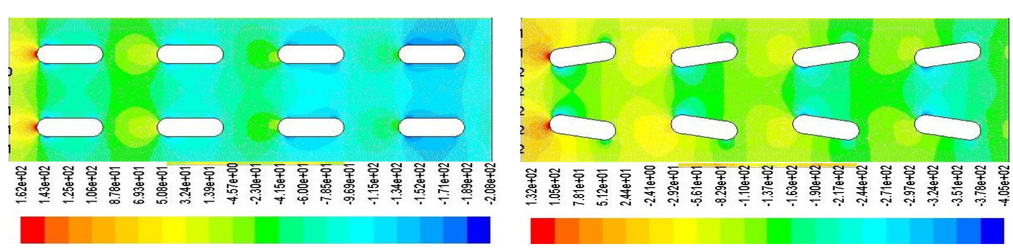


Figure 3. variation of static temperature for all values of α in case 3.



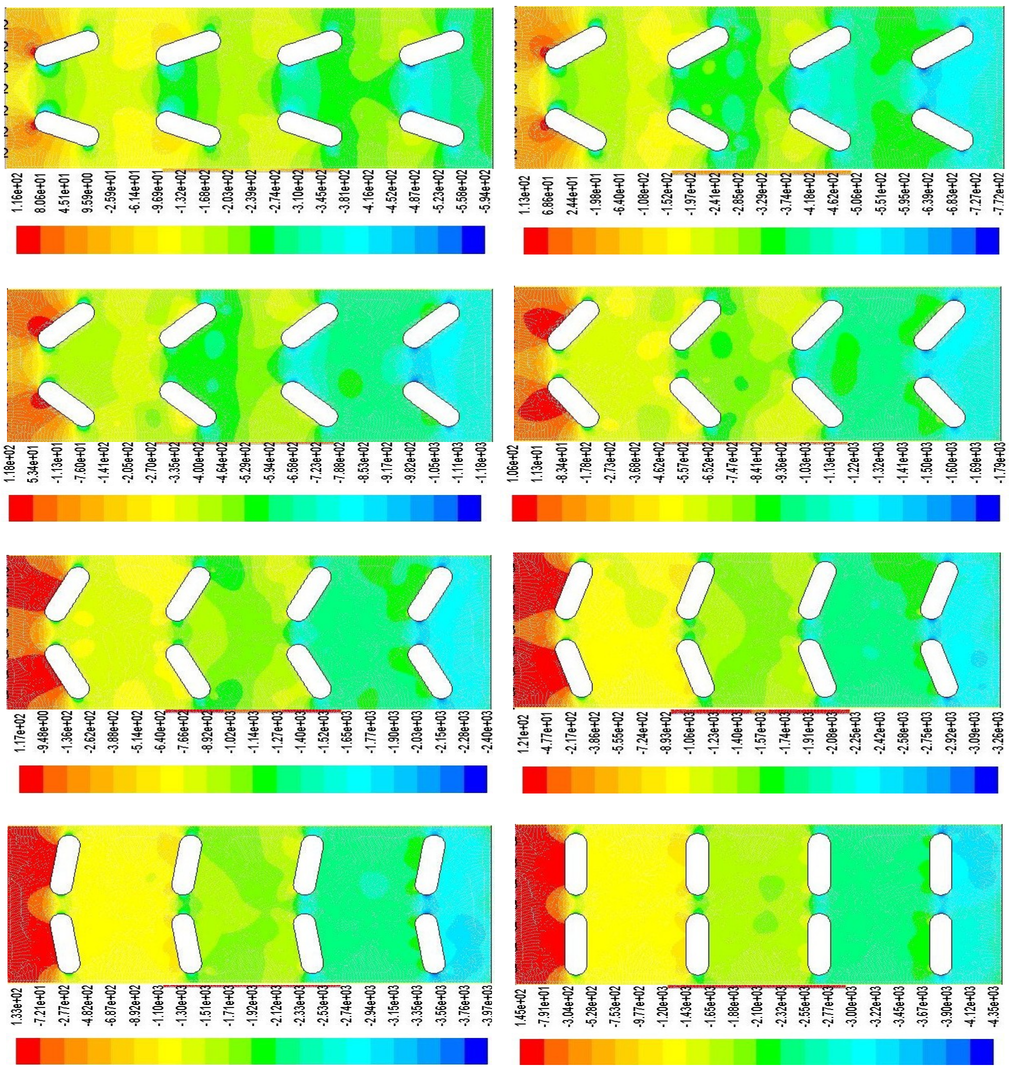


Figure 4. variation of static pressure for all values of α in case 3.

3. Results and Discussion

The present work it is intended to investigate the heat transfer and pressure drop for a bank of flat tubes which are inclined. The inclination angle α ranged from zero to ninety degrees for two aspect ratios; 2.5/7 and 3/7, each of them was tested at two Reynolds numbers; 7850 and 12830. The angle between the longer tube cross section diameter with the horizontal tube changed in the range from zero to ninety degrees for both aspect ratios. Many runs were performed to investigate the effect of changing α on Nusselt number, as a measure of heat transfer activity, and the non dimensional pressure drop, as a measure of power consumption.

For all cases, it was observed that both Nusselt number and pressure drop increased with α. The rate of increase for both pressure drop and Nusselt number was observed to increase rapidly first, and then, started to decrease while approaching the right α. That reflects the effect of the dynamic blockage which increases with at a rate that decreases while approaching the right angle. Blockage increases form drag and produces a narrower air passages, that result in higher velocity gradients and consequently higher turbulent fluctuations. For the heat transfer activities, the increase of the projection of the tube on a plane normal to the flow direction increases the scale of large eddies in the wake of the tube, which contain more kinetic energy, and stretch more smaller size eddies which promote for more turbulent fluctuations. These fluctuations carry more thermal heat from the tube surfaces, and, consequently, increase the Nusselt number with α. Figures 3 illustrates the distribution of static temperature, for all studied values of α, in case 3. It is obvious in the figure that, the mixing effect and turbulence intensity increase with α , which, in turn, help the regions of higher temperature to stretch in the backword direction. Figures 4 illustrates the distribution of static pressure, for all studied values of α , in case 3. The effect of tube blockage appear behind the first tube row, but behind the next row, this effect is reduced by the lower pressure in the wake of the previous tube row, and so on. The same trend repeats with a scale which increases with α .

Figure 5 illustrates the variation of average Nusselt number with α for cases 1 to 4. The Nusselt number reflects the convection activities with respect to the conductivity, and, is observed to increase with the Reynolds number. It exhibited higher values with the smaller aspect ratio, 2.5/7. That may be interpreted by the narrower passages with smaller R, which increases the interaction between the large scale eddies that resulted in more turbulent fluctuations production.

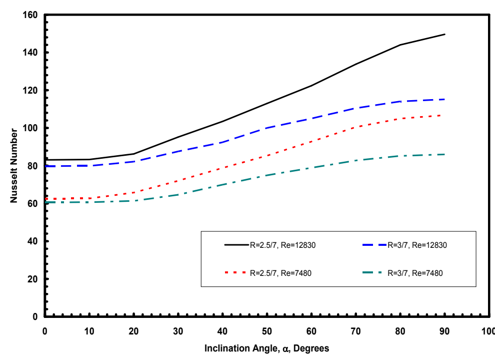


Figure 5. average Nusselt number with α .

Figure 6 illustrates the variation of non-dimensional pressure drop with α for the same cases. It is obvious that, the rate of pressure drop increase with α has the same trend as that of the tube blockage, which is proportional to sin α .

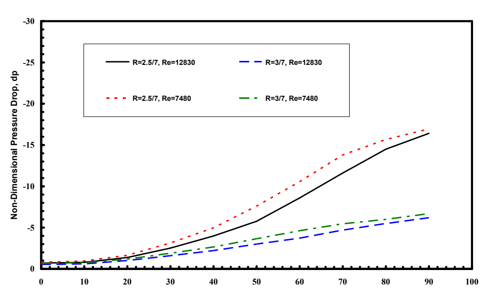


Figure 6. Variation of non-dimensional pressure drop with α

Considering the efficiency for the tube bank device, it was suggested to indicate the gain of heat transfer with respect to the pressure losses, for the purpose of economical assessment. The Nusselt number could be considered as indication of the restored part of dissipated thermal energy, whereas, the pressure drop is considered as the additional energy cost. So, the percentage of increase of Nusselt number for an inclined tube with respect to that of the horizontal one was divided by the same percentage for the non-dimensional pressure losses. The resulting ratios are illustrated for the four cases in Figure 7. The figure indicates that, in all cases, the ratio increased rapidly till almost α =30^o, and then, a little variation till α =40^o, followed by mild decrease to the end. It is observed that, the highest ratios occur in two cases; the first is when R=2.5/7, Re = 12830, with inclination angle of 30^o, and the second is when R=3/7, Re = 12830, with inclination angle of 50^o.

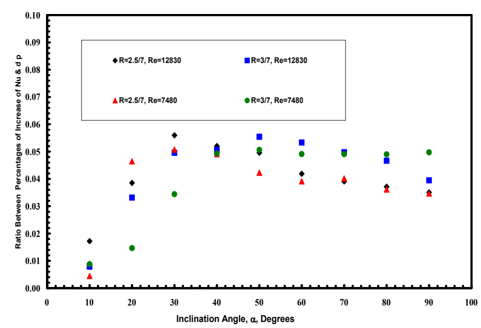


Figure 7. Ratios of Nusselt to pressure drop percentages

Conclusion

The present work includes an investigation of the heat transfer and pressure drop for a bank of inclined flat tubes, that are arranged with two aspect ratios; 2.5/7 and 3/7, each of them was tested at two Reynolds numbers; 7850 and 12830. both Nusselt number and pressure drop increased with α . The most economical cases are noticed when R=2.5/7, Re = 12830, with inclination angle of 30^o, and when R=3/7, Re = 12830, with inclination angle of 50^o.

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Nomenclature and Abbreviations

A	area
C_p	specific heat of air, [kJ/kg K].
dp_{av}	Nondimensional average pressure drop
D	Diameter
dp	non-dimensional pressure drop
E	energy
h	inthalpy
H	vertical distance, (between cross section centroids)
K	thermal conductivity, (W/m.K)
L	horizontal distance, (between cross section centroids)
Nu	Nusselt number, ($h D_h / k$)
n	number of tubes in the domain
P	static pressure
Q	total heat transfer, W
R	Aspect ratio, ($(H/D_h) / (L/D_h)$)
Re	Reynolds number, ($\rho u D_h / \mu$)
S	source term
$\vec{u}$	velocity vector
U	average heat transfer coefficient, ($W/m^2.K$)
u	velocity horizontal component
v	velocity vertical component
t	time
T	static temperature

Subscripts

a	air.
av	inlet-outlet average
b	bulk
h	hydraulic.
i	index for one of the three coordinates
j	index for one of the three coordinates
in	inlet to domain
o	outlet from domain
s	tube outer surface area
t	tube
tot	total

Greek Symbols

α	inclination angle
Γ	diffusion coefficient
κ	turbulent kinetic energy
ε	turbulent energy dissipation rate
ρ	density, (kg/m^3).
μ	viscosity
ν	kinematic viscosity