Numerical Investigation of Heat Transfer and Fluid Flow a cross Circular and Elliptical Tube Bank

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Abstract

In this study, a numerical investigation of heat transfer and fluid flow a cross circular and elliptical tube bank has been made then comparisons made between these two types. The study focus on the effect of Reynolds number and ellipse eccentricity on the Nusselt number and pressure drop through the tube bank for two arrangements (in-line and staggered). The results are reported for air and for Reynolds number range of $20 \leq Re \leq 500$. The results of the study shows that Nusselt number increased as Reynolds number increased for all cases. The values of Nusselt number for elliptical tube are higher than for circular tube for Reynolds number range of $100 \leq Re \leq 500$.The pressure drop for the two arrangements increased as the Reynolds number increases. The elliptical tube array has batter thermal performance and lower pressure drop than circular tube array .

المستخلص

تم في هذا البحث دراسة انتقال الحرارة وجريان السائل خلال نوعين من مجمعات الأنابيب عدديا. مجمع ذا أنابيب دائرية المقطع ومجمع ذا أنابيب بيضوية المقطع ثم تم المقارنة بينهما . تم دراسة تأثير عدد رينولد و درجة تحدب المقطع البيضوي على عدد نسلت وانخفاض الضغط خلال مجمع الأنابيب لنوعين من الترتيب هما (الخطي والمتعرج). النتائج المستخلصة كانت للهواء ولعدد رينولد بحدود $20 \leq Re \leq 500$. النتائج التي تم الحصول عليها تشير إلى إن عدد نسلت يزداد مع زيادة عدد رينولد لكل الحالات . كذلك فان قيمة عدد نسلت للأنابيب البيضوية اكبر منها للأنابيب الدائرية لقيم عدد رينولد $100 \leq Re \leq 500$. هبوط الضغط لكلا الترتيبين يزداد بزيادة عدد رينولد . كذلك وجد إن الأنابيب البيضوية لها أفضل أداء حراري من الأنابيب الدائرية.

1. Introduction

Heat exchangers have been playing a huge role in energy applications. Many studies have been done to improve the thermal and hydraulic performance of heat exchanger. Austin et al [1] studied the effect of the entering water jet on the heat transfer coefficient and pressure drops across of 10-row depth having staggered and in-line tube arrangement. For there study the heat transfer coefficient was found strongly dependent on the transverse and longitudinal position of the tube in the first rows.

Fujii et al[2] studies numerical analysis of laminar flow and heat transfer in tube bank. They solved numerically the two-dimensional Navier-Stokes and enegy equations. The calculation are carried out for in-line square tube bank up to five rows deep,with pitch –to diameter ratios 1.5×1.5 under the condition of uniform tube wall temperature for Re= 60,120 and 300. An analogous relation between an in-line tube bank and parallel plates with heat transfer characteristics is presented.

Ota et al[3] studied the flow in the neighborhood of an elliptical cylinder with an axis ratio of 0.33 in the Reynolds number range (based on major axis length) from 3.5×10^4 to 1.25×10^5, where a discontinuous variation of both the drag and lift were observed. It is evidenced that at cross-flow the elliptical tube perform better drag than the circular tube .

Zdravistch et al[4] studied numerically the predication of the laminar and the turbulent fluid flow and heat transfer in the tube banks .Two tube arrangement investigated staggered and in-line tube banks. The cell-centred finit-volume algorithm is used to solve the steady state Reynolds-averaged Navier-Stokes equations. Two-dimensional results include velocity vectors and streamline, surfaces shear stresses, pressure coefficient distributions, temperature contours, local Nusselt number distributions and average convective heat transfer coefficient. These results indicate good agreement with experimental data.

Nishiyama et al[5] investigated the effects of cylinder spacing and angle of attack on heat transfer for elliptical tube. They found that the angle of attack as wall as the cylinder spacing influence the local heat transfer coefficient. They concluded that the cylinder spacing and angle of attack, should be arranged as small as possible to minimize the drag and to achieve higher heat transfer rate.

Badr[6] conducted a numerical study with a single elliptical cylinder and investigated the effects of Reynolds number, cylinder axis ratio and the angle of attack on heat transfer. The Reynolds number (based on the focal distance) varied from 20 to 500 for a constant

Prandtl number of 0.7. The axis ratio was varied between 0.4 and 0.9 for zero angle of attack and for a fixed Reynolds numberof 100. In addition, the angle of attack was altered from 0° to 90° for a fixed axis ratio of 0.5 for Reynolds number between 20 to 500.He found that the maximum Nusselt number occurred at zero angle of attack and the maximum heat transfer at small axis ratio.

Nasibi and Shicani [7] investigated the laminar flow of air around three isothermal horizontal cylinders in an in-line tube bank. They used a body fitted curvilinear system with Beam-Wrming numerical method to solve full Navier-Stokes and energy equations. They study the effects of longitudinal and transverse pitches and Reynolds number ranged 25-2500 on the flow parameters such as streamlines, surface presser, total drag , pressure drag and friction drag coefficient . They found that the change of longitudinal and transverse pitches and Reynolds number effects the flow parameters, As longitudinal pitch is decrease, the wakes behind the cylinder are spread between two roes. The drag coefficient as Reynolds number increased as the transverse pitch increased.

Harris and Goldschmidt [8] investigated the effect of axis ratio and angle of attack on overall heat transfer in the Reynolds number range from 7.4×10^{3} to 7.4×10^{4}.Both the Reynolds number and Nusselt number were based on the length of the major axis. They concluded that an axis ratio of 0.3 or less must be achieved to realize any appreciable change in heat transfer coefficient greater than 10% over the elliptical tube.

The objective of the present study is to numerically study the heat transfer and fluid flow across circular and elliptical tube bank then a comparison between the two types.

2. Mathematical formulation

Both in-line and staggered arrangement are considered for analysis. A total of six longitudinal rows have been considered. The working media is air of (Pr=0.7). The arrangements considered in this study are shown in Figures (1) and (2). The comparison is done by comparing circular array of tube diameter 1 cm with elliptical array. The ellipsoidal tubes were designed with the ratio of 0.8, 0.6 and 0.4 between the minimum and the maximum ellipse radius (b and a).This ratio is ellipse eccentricity (e = b/a). To give the same heat exchange area for all cases. To simplify the numerical solution and reducing the run time and due to similarity a heat exchanger module (HEM) consist of two tubes as shown in figures(1)and (2).

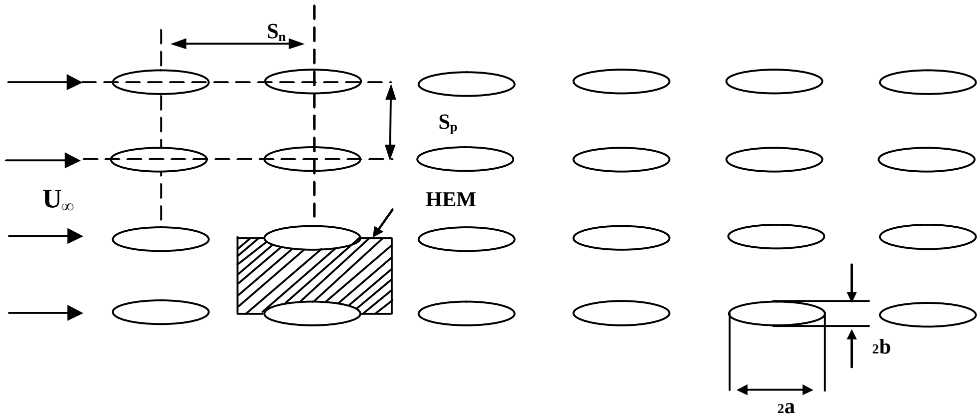


Figure (1). In-line arrangement of tube array .

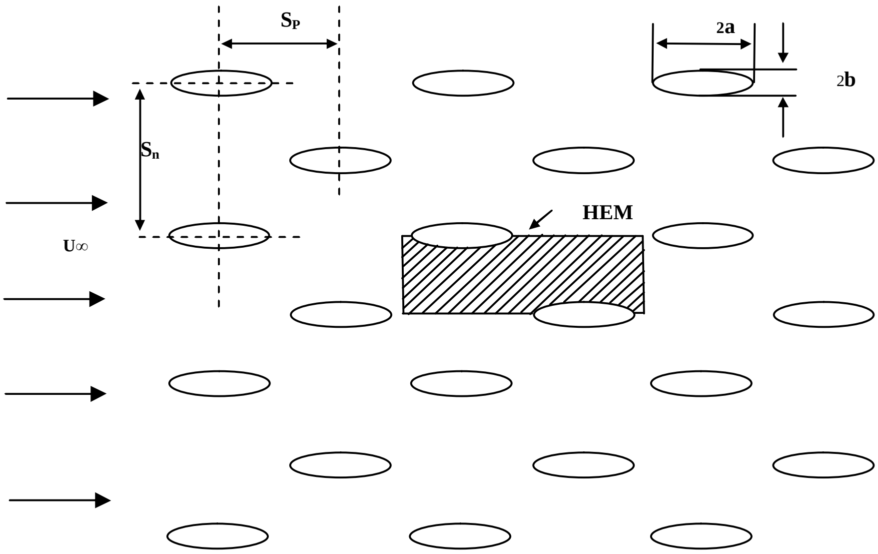


Figure (2). Staggered arrangement of tube array.

The parameters considered for numerical simulation are 20≤ Re ≤ 500, Pr = 0.7 .Reynolds number is define as

$$Re=\frac{U_{max}D_H}{n} \qquad (1)$$

Where:

$$D_H = \frac{4A_e}{P} \tag{2}$$

The flow is fully developed hydrodynamically and thermally. The thermo physical properties assumed constant and taken at the inlet temperature of the fluid.

2.1 Governing equations

For steady state, two-dimensional, incompressible flow, and constant fluid property, the governing equations are:

Continuity equation:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \tag{3}$$

X- momentum equation:

$$r\left[U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y}\right] = -\frac{\partial P}{\partial X} + m\left[\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right] \tag{4}$$

Y-momentum equation:

$$r\left[U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y}\right] = -\frac{\partial P}{\partial Y} + m\left[\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right] \tag{5}$$

Energy equation:

$$rCp\left[U\frac{\partial T}{\partial X} + V\frac{\partial T}{\partial Y}\right] = K\left[\frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2}\right] \tag{6}$$

2.2.Boundary condition:

2.2.1 Inlet condition

The fluid is assumed to enter with a uniform horizontal velocity, U_{in} and temperature of T_{in}.

$U=U_{in}$, $T=T_{in}$, $V=0$

2.2.2.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 is

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

2.2.3.Wall condition

At the tube surfaces no slip condition is applied as far as energy equation is consider the tube is at isothermal condition:

$U = V = 0$, $T=T_w$

2.2.2 Outlet condition

Zero diffusion flux implemented for all variables at the outlet boundary.

$$\frac{\partial U}{\partial X}=\frac{\partial V}{\partial X}=\frac{\partial T}{\partial X}$$

The calculation of Nusselt number for the isothermal boundary condition:

$$Nu=\frac{hD_H}{K} \qquad (7)$$

Where h is given by

$$h = \frac{Q}{A(T_w - T_b)} \tag{8}$$

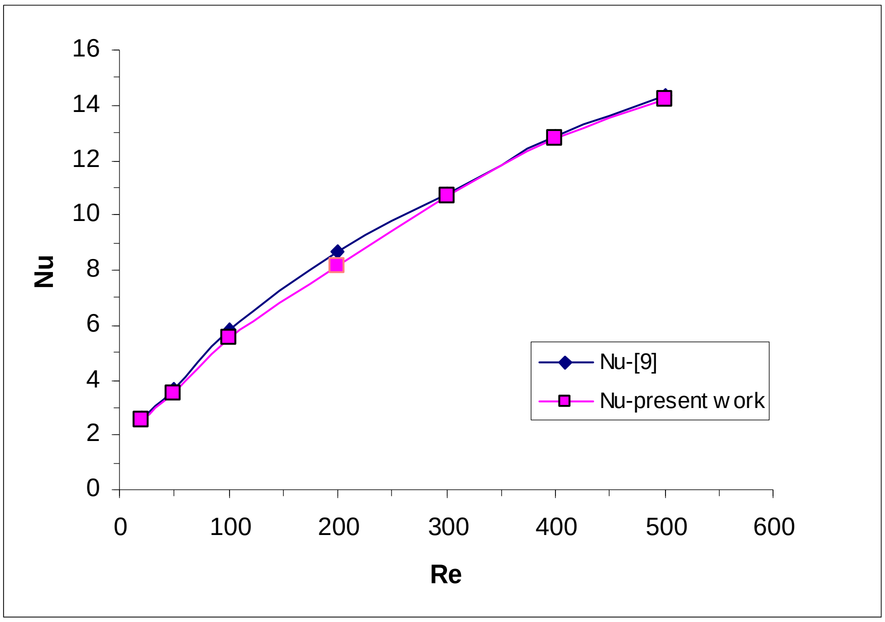
And

$$Q = m^{\circ} Cp[T_b(out) - T_b(in)] \tag{9}$$

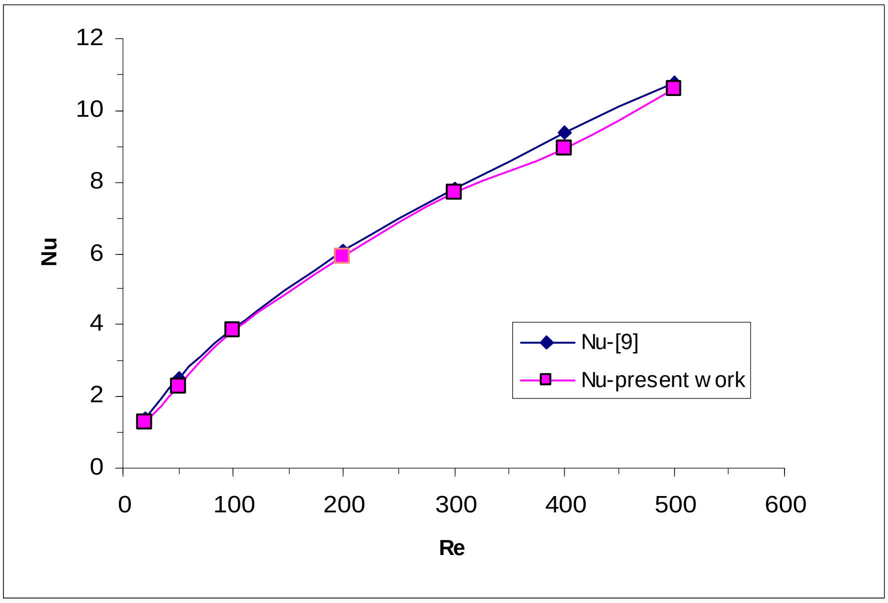
The governing continuity, fluid flow and energy equations (3-6) along with boundary conditions are solved simultaneously by using a finite volume method. Steady segregated solver was used with second order upwinding scheme for the convective terms in the momentum and energy equations. For pressure-velocity coupling, pressure implicit with splitting of operators (PISO) scheme was used. A convergence criterion of 1×10^{-6} was applied to the residual of the continuity and the momentum equations and 1×10^{-9} to the residual of the energy equation.

3. Validation

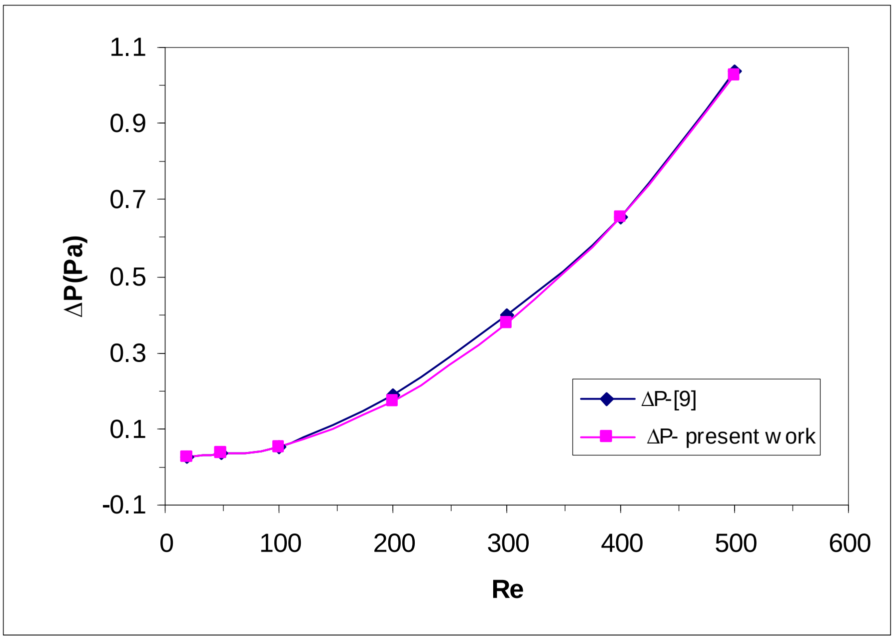
In order to check the validity of the present work, a comparison is made with [9]. A comparison of the Nusselt number and the pressure drop for the two arrangement in-line and staggered are made. Circular tube array of six longitudinal rows have been considered with tube diameter d=1 cm , $S_{n=}2$ cm and S_p=2 cm. Figure (3) shows the comparison between Nusselt number obtained from [9] with Nusselt number obtained from present model for staggered arrangement ,from this Figure it can be seen that ,the agreement is accepted and the average error is 2.6%. From figure (4), it can be observed that the comparison of the Nusselt number computed from (9) with Nusselt number from present model for in-line arrangement also with accepted agreement, The average error is 3.6%. In Figure (5) the comparison of pressure drop calculated from [9] with pressure drop obtained from present model for staggered arrangement. The average error is 3.8 %. The comparison of pressure drop calculated from [9] with pressure drop obtained from present model for in-line arrangement is shown in Figure(6). The average error is 2.8 %.From the above results ,it can be concluded that the present model have good accuracy.



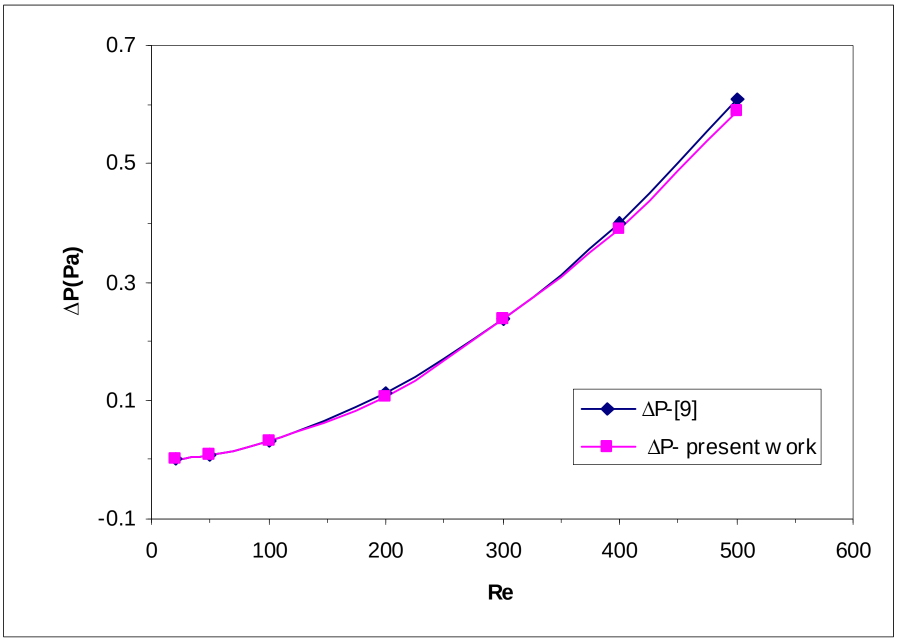
Figure(3). Comparison of Nu of [9] with Nu of present work for staggered arrangement.



Figure(4). Comparison of Nu of [9] with Nu of present work for in-line arrangement .



Figure(5). Comparison pressure drop of [9] with pressure drop of present work for staggered arrangement .



Figure(6). Comparison pressure drop of [9] with pressure drop of present work for in-line arrangement.

4. Results and discussion

The effect of a constant temperature boundary condition on the steady fluid flow and heat transfer is analyzed here Figure(7) shows the variation of Nusselt number with Reynolds number for staggered arrangement for circular and elliptical tube that have ellipse eccentricity e=0.8, e=0.6 and e=0.4 . It can be seen that, Nusselt number increase with increasing of Reynolds number for all cases . The Nusselt number for elliptical tube is less than that of the circular tube for $20\leq Re\leq 100$. However for $100\leq Re\leq 500$ this relationship is inverted due to decreases in flow turbulence in case of elliptical tube. The effect of decreases ellipse eccentricity from e=0.8 to e=0.4, it gave high Nusselt number for the same Reynolds number. The maximum heat gain at ellipse eccentricitye=0.4.

Figure(8) shows the variation of Nusselt number with Reynolds number for in-line arrangement for circular and elliptical tube that have ellipse eccentricity e=0.8, e=0.6 and e=0.4 . The Nusselt number exhibit the same behavior as that observed in staggered arrangement .The Nusselt number value have lowest than in staggered arrangement. From the above Figures, it can be observe that from a heat transfer point of view staggered arrangement performs better than in-line arrangement.

Figure(9) shows variation of pressure drop with Reynolds number for staggered arrangement for circular and elliptical tube that have ellipse eccentricity e=0.8, e=0.6 and e=0.4 . It can be seen that, the pressure drop increases as the Reynolds number increases. The effect of ellipse eccentricity on the pressure drop are as ellipse eccentricity decreases from e= 0.8, to e= 0.4, the pressure drop value decreases for the same Reynolds number mainly due to its more streamline flow pattern. The differences between the pressure drop circular and elliptical tube increase with increase of Reynolds number.

Figure(10) illustrate variation of pressure drop with Reynolds number for in-line arrangement for circular and elliptical tube that have ellipse eccentricity e=0.8, to e=0.4. It can be observed that pressure drop exhibit the same behavior as was observed in staggered arrangement .The pressure drop value have lowest than in staggered arrangement. From the above Figures, it can be observe that from a pressure drop point of view in-line arrangement performs better than staggered arrangement.

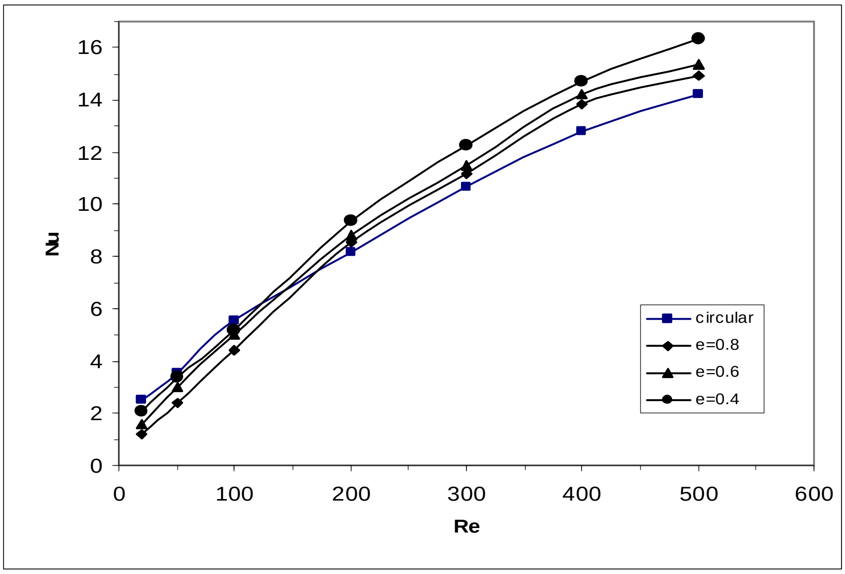
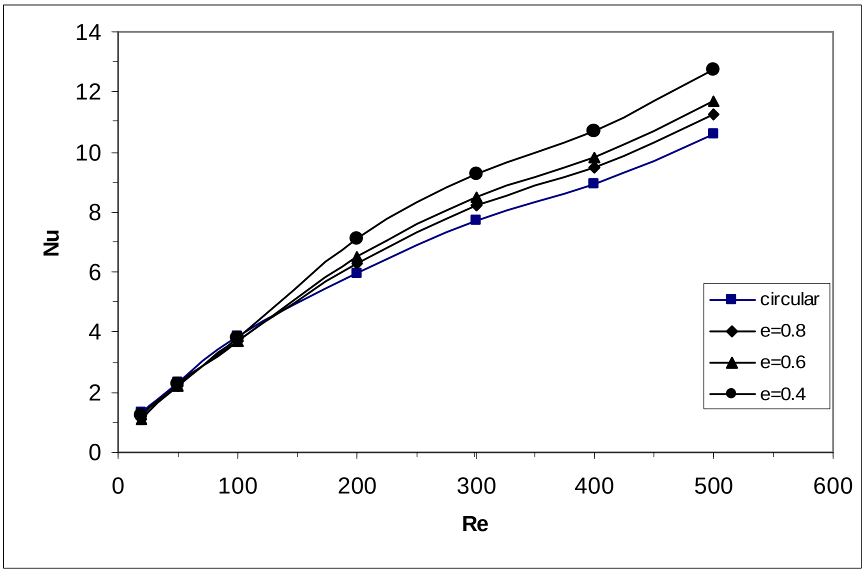


Figure (7). Variation of Nusselt number with Reynolds number for staggered configuration for circular and eliptical tube with ellipse eccentricity e=0.8, e=0.6 and e=0.4



Figure(8). Variation of Nusselt number with Reynolds number for in-line configuration for circular and elliptical tube with ellipse eccentricity e=0.8, e=0.6 and e=0.4

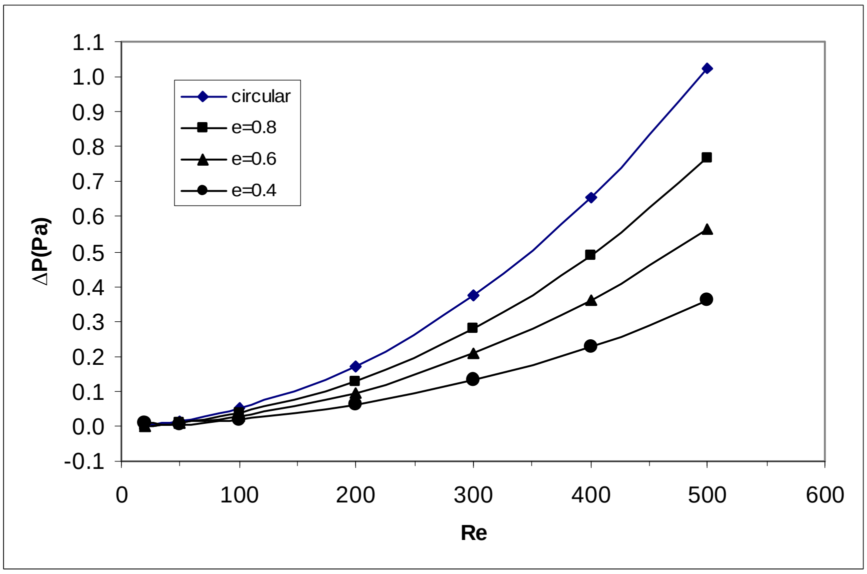


Figure (9). Variation of pressure drop with Reynolds number for staggered configuration for circular and elliptical tube with ellipse eccentricity e=0.8, e=0.6 and e=0.4.

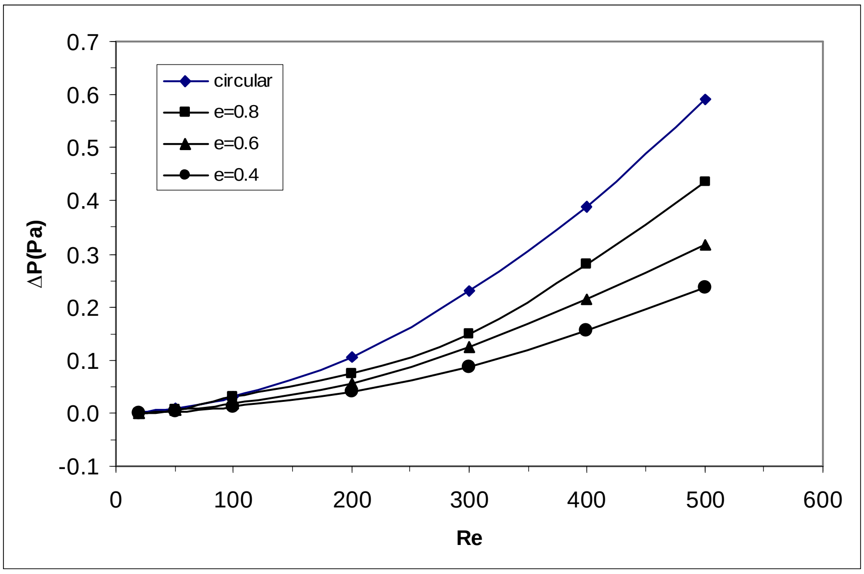


Figure (10). Variation of pressure drop with Reynolds number for in-line configuration for circular and elliptical tube with ellipse eccentricity e=0.8, e=0.6 and e=0.4.

5. Conclusions

Heat transfer and fluid flow a cross circular and elliptical tubes bank was numerically investigated. From the result obtained it can be concluded:

1- Nusselt number increased as Reynolds number increased for all cases.
2- The effect of decreases ellipse eccentricity from e=0.8 to e=0.4, it gave high Nusselt number for the same Reynolds number.
3- The pressure drop for the two arrangements increased as the Reynolds number increases.
4- The effect of decreases ellipse eccentricity from e=0.8 to e=0.4, it gave lower pressure drop for the same Reynolds number.
5- The elliptical tube array has batter thermal performance and lower pressure drop than circular tube array .

6. References

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7. Nomenclature

Symbol	Description
a	Larger ellipse semi-axis,m.
A	Total area of all tubes ,$m^{2.}$
A_e	Ellipse area
b	Smaller ellipse semi-axis,m.
C_p	Specific heat , J/kg. k
d	Tube Diameter,m.
D_H	Hydraulic Diameter ,m.
h	Heat transfer Coefficient ,$W/m^2.k$
HEM	Heat Exchanger Model
K	Thermal conductivity, W/m.k
Nu	Nusselt Number
P	Ellipse circumference
Q	overall heat transfer rate ,W.
Re	Reynolds number
S_n	longitudinal spacing between tubes ,m.
S_p	transverse spacing between tubes ,m.
T	Temperature, K.
T_b	Bulk Temperature, K.
T_{in}	Inlet Temperature, K.
T_w	Wall Temperature, K.
U_∞	Upstream Velocity ,m/s.
U_{max}	maximum Velocity ,m/s.
ν	Kinematic Viscosity ,m^2/s.
ΔP	Pressure drop ,Pa.