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Heat Transfer and Fluid Flow Characteristic in banks Flat Tubes

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Abstract

In this research a study effect of the length ratio (L/Da) and the height ratio (H/Da) for banks flat tube heat exchanger In-Line and staggered arrangement on force convection heat transfer and friction coefficient by (Fluent-CFD) numerical program.

The governing equations (mass, momentum and energy) are solving by using Finite Volume (Fluent-CFD) software for considering steady state, two dimensional, at constant heat flux with Reynold's number ($100 \leq Re \leq 8000$).

The results show that increasing (H/Da), (L/Da) lead to decreasing friction coefficient and enhancement of (Nu) is at (H/Da=2) for all (L/Da) values In-line arrangement and at (H/Da =2, L/Da =5) for staggered arrangement.

Key words: banks flat tubes, convection heat transfer, and fluid flow.

انتقال الحرارة وجريان مائع لحزمة من الأنابيب المستوية

الخلاصة

تضمن البحث دراسة تأثير النسبة الباعية الطولية (L/Da) والشاقولية (H/Da) لمبادل حراري ذو الانابيب المسطحة في ترتيب خطي ومتخالف لانتقال الحرارة بالحمل القسري ومعامل الاحتكال السطحي بأستخدام البرنامج العددي(Fluent-CFD).

تم حل المعادلات الحاكمة (معادلة الاستمرارية والزخم والطاقة) بطريقة الحجوم المحددة بواسطة البرنامج المذكوراعلاه مع الاخذ بنظر الاعتبار ان الجريان ثنائي البعد للحالة المستقرة عند ثبوت الفيض الحراري لانابيب المبادل لمدى عدد رينولد($100 \leq Re \leq 8000$).

اظهرت النتائج ان زيادة (H/Da)و(L/Da) يؤدي الى انخفاض في قيمة معامل الاحتكاك السطحي، كما ان افضل قيمة لعدد نسلت عند(H/Da=2) لجميع قيم (L/Da) للترتيب الخطي واما للترتيب المتخالف (H/Da =2 ,L/Da =5).

الكلمات الدالة: حزمة الانابيب المستوية، انتقال الحرارة بالحمل القصري، جريان الموائع

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Symbol

C_f	Skin friction coefficient	-	P	Pressure	Pa
Da	Small diameter	m	Pr_{eff}	Turbulent Prandtl number	-
			Re_{Dh}	Reynolds number	-
Db	Long diameter	m	T	Temperature	^oC
D_h	Hydraulic diameter	m	WU,WL	Surface upper, lower flat tube	-
H	Height	m	u_x,u_y	Cartesian velocity components in x & y axis	
h	Heat transfer coefficient	$W/m^2.^oC$	ρ	Air density	/s kg/m^3
K	Thermal conductivity	$W/m^2.^oC$	μ	Viscosity	kg/s.m
L	Length	m	μ_{eff}	Effected Viscosity	kg/s.m
Nu ave.	Average Nusselt Number	-	q_H	Constant heat flux	W/m^2

Introduction

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 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 [1]. 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. A brief preview of different studies involving flow over a variety of shapes with various types of flow conditions is worth mentioning at this time for the reader.

Merker and Hanke (1986)[2] found experimentally the heat transfer and pressure drop performance of staggered oval tube banks with different transversal and longitudinal spacing. They showed that an exchanger with oval-shaped tubes had smaller frontal areas on the shell-side compared to those with circular tubes. Chang, et al. (1989)[3] has developed a numerical (by finite element technique) to predict the heat transfer and pressure drop coefficient in cross flow through rigid tube bundles. Results for heat transfer and pressure drop coefficients are obtained for tube arrays of pitch ratios of 1.5 and 2 are very good agreement of the predicted numerical results and experimental data obtained. Chen, et al. (1990)[4] numerically studied flow and thermal fields in forced convection over a heated cylinder for both incompressible and compressible flow, the governing equations included two-dimensional Navier-Stokes momentum, energy, and continuity equations in body-fitted coordinates. The finite difference approximation for the transformed conservation equation was obtained by the integration over the control volume. The velocity and pressure fields were linked by the Semi-Implicit Method for Pressure Linked Equation (SIMPLE) algorithm. Kundu, et al. (1991a)[5] numerically studied heat transfer and fluid flow over a row of in-line cylinders placed between two parallel plates. Incompressible, two dimensional, and laminar flow was considered. The spacing between cylinders causes three different separation patterns, when the spacing is small; the separated flow between cylinders is stable. As the spacing increases, flow in the separated zone becomes periodic. At higher values of spacing, the separated flow is local and does not extend to the next cylinder. The heat transfer data for different aspect ratios and Reynolds numbers are reduced to form a single formula for ease of interpolation. Grannis and Sparrow (1991)[6], for the fluid flow in a heat exchanger consisting of an array of diamond shaped pin-fins. The model that underlines the solutions was based on the concept of the periodically fully developed regime, where by the velocity field repeats itself from row to row and the pressure drop per module remains constant, the result included representative streamline maps and isobar and an in depth display of pressure drop information. Yu, et al. (1995)[7] applied the weighted residuals method to analyze mixed convection heat transfer in a 3x3 in-line horizontal tube bundles placed between two vertical parallel plates. The flow regimes of Reynolds numbers up to 500 and Grashof numbers up to 53000 were investigated and the local data of the different geometries were reported. The average Nusselt number for the array increases 20-30% when stream wise spacing was increased by 50%. Ertan buyruk (1999)[8] an experimental study was carried out to investigate heat transfer and flow characteristics from one tube within a staggered tube bundle and within a row of similar tubes. The tube spacing examined the longitudinal pitches (St) and transverse pitches (Sl) are (1.5-1.5), (1.5-1.25) respectively. The variation of local Nusselt number was predicted with Reynolds number ($4.8*10^4$). The aim of the second part of the investigation was to examine the influence of the blockage of a single tube in a duct and transverse pitch for a single tube row with Reynolds number range of (7960) to (47770). For single tube row experiments, if the blockage ratio is less than 0.5, the general shape of local Nusselt number distribution around the cylinder varies only slightly with blockage. Castiglia, et al (2001)[9] the subcritical flow over an array of elliptic cylinders with an axis ratio of (1:2) was studied both experimentally and numerically. The mean velocities, turbulence levels and the vortex dynamics of the array were determined experimentally by flow visualization and using a Laser Doppler Anemometer (LDA) and the flow was modeled using three-dimensional Large Eddy Simulation (LES). The experimental results were compared with results obtained previously using circular cylinders and with numerical predictions of the flow. The study indicated that the flow past such a widely spaced array is characterized by low turbulence levels, poor lateral mixing compared with conventional circular cylinder arrays, the predicted mean, and velocities, as well as the flow periodicity, were in good agreement with the experimental results. Vikas Kumar, et al (2003)[10] three dimensional numerical simulation study has been carried out to predict airflow and temperature distribution in the tube type heat exchanger. Due to symmetry in geometrical construction, a section of heat exchanger has been considered for CFD analysis by using PHOENICS software. The k - ε turbulence model has been used to solve the transport equations for turbulent flow energy and the dissipation rate. The simulated results predict the temperature distribution reasonably at different locations of heat exchanger. The CFD model may be used to optimize its thermal performance by varying the location of baffles & partition plate in the heat exchanger. Andrej , Borut (2006)[11] Transient numerical simulations of heat and fluid flow were performed for eight heat exchanger segments with cylindrical and wing-shaped tubes in staggered arrangement. Their hydraulic diameters were from (0.5824 to 3.899) cm for the cylindrical tubes, and from (0.5413 to 3.594) cm for the wing-shaped tubes. In general, the drag coefficient and the Stanton number are smaller for the wing shaped tubes than for the cylindrical tubes. However, with an increasing hydraulic diameter, these differences between both forms of tubes diminish. Liang, Papadakis (2007)[12]The Large Eddy Simulation (LES) technique is used to study the vortex shedding characteristics inside a staggered tube array consisting of six rows with intermediate spacing (S_L/D=1.6, S_T/D=3.6) at the subcritical Reynolds number of 8600 (based on the g velocity). The filtered equations a using the finite volume method in an unstructured, collocated grid arrangement with second-order accurate methods in space and time. The predictions of mean velocities and Reynolds stresses are in very good agreement with detailed LDA measurements performed in 17 stations along the depth of the array. The low frequency component was present behind all rows, the high component was detected behind the first and second rows only. Clearly, the fact that the flow was allowed to develop along multiple rows was instrumental for the successful prediction of these two shedding frequencies.

The main objective of this work is numerical study to predict heat transfer and friction coefficient characteristics of flows around or through rigid complex geometry (flat tube, Db=2Da), which two-dimensional model created and meshed in (Gambit) software. The effects of various independent parameters such as Reynolds number (Re=100-8000), length ratio (L/Da= 5, 6, 7), and height ratio (H/Da= 2, 3, 4) on friction coefficient and heat transfer were studied by (Fluent-CFD) software.

Mathematical model

Consider banks tubes ranging from a flat tube placed at in-lined and staggered arrangement with cross airflow. The wall of the flat tube is heated under constant heat flux q_H, and air inlet at variable velocity V_x. The physical model of the present problem is illustrated in Fig. (1). The force convection heat transfer between heated flat tube surface and inlet airflow in a horizontal x-y plane. The two-dimensional governing equations were summarized as follows under the following assumptions [13, 14 and 15] Steady state, the laminar, turbulent flow, the fluid is incompressible, and the viscous dissipation is negligible in the energy equation, constant heat flux at surface flat tube.

Key modeling equations in Fluent for fluid flow and heat transfer from force convection are the conservation equations of mass, momentum, and energy:

$$\left[\frac{\partial u_x}{\partial x}+\frac{\partial u_y}{\partial y}\right]=0 \text{ .... } (1)$$

$$\rho\left(u_x\frac{\partial u_x}{\partial x}+u_y\frac{\partial u_x}{\partial y}\right)=-\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left(2\mu_{eff}\frac{\partial u_x}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial u_y}{\partial y}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial u_y}{\partial x}\right) \text{ ....} (2)$$

$$\rho\left(u_x\frac{\partial u_y}{\partial x}+u_y\frac{\partial u_y}{\partial y}\right)=-\frac{\partial P}{\partial y}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u_y}{\partial x}\right)+\frac{\partial}{\partial y}\left(2\mu_{eff}\frac{\partial u_y}{\partial y}\right)+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u_x}{\partial y}\right) \text{.......... } (3)$$

$$\rho\left(u_x\frac{\partial T}{\partial x}+u_y\frac{\partial T}{\partial y}\right)=\frac{\partial}{\partial x}\left(\frac{\mu_{eff}}{\Pr_{eff}}\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\mu_{eff}}{\Pr_{eff}}\frac{\partial T}{\partial y}\right) \text{ .... } (4)$$

$$Cf=4f \text{ .... } (5)$$

$$\mathrm{Re}_{Dh}=\rho\times u\times D_h/\mu \text{ .... } (6)$$

$$Nu=\frac{h*D_h}{k} \text{ .... } (7)$$

$$D_h=(4Ac/P)=2H \text{ .... } (8)$$

There are four boundary condition which needed to solve this problem: at entrance region, inlet velocity at Re = (100 - 8000). At outlet region, outlet flow rate. For all wall pipes of banks flat tubes are constant heat flux and symmetry boundary conditions as a show in fig. (1).

Numerical simulation

By using (Fluent-CFD) software solves equations for conservation of mass, momentum, and energy using a finite volume technique to show dynamic flow and heat transfer around flat plat tubes, The model geometry and mesh generation are build by (Gambit) software [13] as a show in fig.(2). The grid is made up of triangular elements to improve the quality of the numerical prediction near the curved surfaces. The continuity is satisfied using a semi-implicit method for pressure linked equations, which is referred to as the SIMPLE procedure. To reduce numerical errors, second order upwind discrimination 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 1500 iterations are generally needed for convergence.

Results and Discussion

The temperature profile

Fig.(3, 4) shows the effect of Reynolds number (100-8000) on the temperature profile for In-Line and staggered arrangement. As the Reynolds number increases, the lower value of temperature to pass through deeper, which means the colder fluid is getting closer to the hot surface that because of the heat transfer would be increased, the Symmetric condition is preserved because of symmetric geometry. The increase in height ratio (H/Da) would make the cold temperature to pass through further downstream as clearly shown in Fig. (5). the effect of varying length ratio (L/Da) is minimal on temperature profile and thus does not have much of an effect on heat transfer. This behavior is more clear at higher Reynolds numbers (Re = 8000). For the staggered arrangement as shown in Figs. (5, 6) It is marked that the lower value of temperature to pass through deeper compared with In-Line arrangement.

Skin friction coefficient (C_f)

Fig. (7, 8) show the effect of Reynolds number (100-8000) flow on Skin friction coefficient (C_f) for In-line and staggered arrangement (H/Da=4, L/Da=6). A higher (C_f) at first column flat tube (WU1, WL1) and a lower (C_f) at fourth column flat tube (WU4, WL4). When increase (Re), Skin friction coefficient is increased reaching turbulent flow the value of (C_f) approximation is the same for all columns.

When increase height ratio (H/Da) and length ratio (L/Da), the value of (C_f) decrease because increasing in distance between tubes of heat exchanger that lead to fluid flow easily and than low-pressure drop.

Fig. (9) indicate velocity vector profile and made vortexes at end of tubes are increase with increase (Re).

Average Nusselt Number

Fig. (10) Comparison between tl present numerical predictions with M A. Mehrabian, (2007) [16] and then obtained a good agreement with this previous work.

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Fig. (11 a, b)(12 a, b) shows the average Nusselt number (Nu) as a function of Reynolds number (Re=100-8000) for In-Line and staggered arrangements, Fig. (11a)(12a) show the effect of ratio length on the average Nusselt number for fixed height ratios, Fig. (11b)(12b) show the effect of ratio height on the average Nusselt number for fixed length ratios.

In general, the average Nusselt number increases with an increase in Reynolds number. The overall performance of an in-line arrangement with a lower height ratio (H/Da = 2) is preferable since it provides higher heat transfer rate for all length ratios and Reynolds numbers as shown in Fig.(11 a, b).

When increase of the length ratio would result in a slight increase in average Nusselt number at a lower height ratio (H/Da = 2).

In staggered arrangement, the one with minimum spacing between the upper and lower tubes appears to give a higher average Nusselt number if the height and length ratios are maintained at the lowest value (H/Da = 2 and L/Da = 5) and at higher height and length ratios (H/Da = 4 and L/Da = 6) the staggered arrangement with the higher spacing gives also a higher module average Nusselt number as shown in Fig. (12 a, b).

Conclusions

1- When increasing ratio (H/Da), the cold temperature to pass through further downstream but effect (L/Da) is minimal on temperature profile.

2- For In-line arrangement, enhancement of (Nu) is at (H/Da=2) for all length ratios and Reynolds numbers; as increase of (L/Da), a slight increase ´Nu).

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For staggered arrangement, ennancement of (Nu) is at (H/Da=2, L/Da = 4).

4- Skin friction coefficient is decreasing when increasing ratios (H/Da), (L/Da).

References

1- Ota T., Nishiyama H., Kominami J., Sato K., 1986, "Heat Transfer Two Elliptic Cylinder in Tandem Arrangement" J. Heat Transfer,Vol.108,PP. 525-531.
2- G. P. Merker, H. Hanke, 1986, "Heat transfer and Pressure Drop on the Shell-Side of Tube-Banks Having Oval-Shaped Tubes" International Journal of Heat and Mass Transfer, Vol .29 (12), PP. 1903–1909.
3- Chang Y., Beris A.N., Michaelides E.E., 1989, "A Numerical Study of Heat and Momentum Transfer for Tube Bundles in Cross-Flow” Int. J. Num. Methods in Fluids Vol. 9, PP.1381-1394.
4- Chen, Chiun-Hsun, Weng, Fang-Bor, 1990, "Heat Transfer for Incompressible and Compressible Fluid Flows Over a Heated Cylinder" Num. Heat Transfer, Part A Vol. 18, PP.325-349.
5- Kundu D., Haji-Sheikh A., Lou D.Y.S., 1991a, "Heat Transfer Predictions in Cross Flow Cver Cylinders Between Two Parallel Plates". Num. Heat Transfer, Part A Vol. 19, PP. 361-377.
6- Grannis V.B., Sparrow E.M., 1991, "Numerical Simulation of Fluid Flow Through an Array of Diamond-Shaped Pin Fins" Num. Heat Transfer Part A Vol. 19, PP. 381-403.
7- Yu, D., Barron, R.F., Ameel, T.A., Warrington, R.O., 1995, "Mixed Convection From Horizontal Tube Banks Between Two Vertical Parallel Plates" Num. Heat Transfer, Part A Vol. 27, PP.473-486.
8- Ertan Buyruk, 1997, "Heat Transfer and Flow Structures Around Circular Cylinders in Cross-Flow" Tr. J. of Engineering and Environmental Science Vol.23, PP.299 - 315.
9- D Castiglia, S Balabani, G Papadakis and M Yianneskis, 2001, "An Experimental and Numerical Study of the Flow Past Elliptic Cylinder Arrays" IMechE 2001 Proc Instn Mech Engrs Part C Vol. 215.
10- Vikas Kumar1, D. Gangacharyulu, Parlapalli MS Rao and R. S. Barve, 2003, "CFD Analysis of Cross Flow Air to Air Tube Type Heat Exchanger".
11- Andrej Horvat and Borut Mavko, 2006, "Drag Coefficient and Stanton Number Behavior in Fluid Flow a Cross a Bundle of Wing-Shaped Tubes" Journal of Heat Transfer Vol. by ASME 128 / 969.
12- C. Liang, G. Papadakis, 2007, "Large Eddy Simulation of Cross-Flow Through a Staggered Tube Bundle at Subcritical Reynolds Number" Journal of Fluids and Structures Vol. 23, PP.1215–1230
13- Fluent 6.2, "FLUENT user’s guide", Lebanon, Fluent Inc., USA, 2005.
14- Pantankar SV., 1980, "Numerical Heat Transfer and Fluid Flow" Hemisphere McGraw-Hill.
15- Launder B. E. and Spalding D.B., 1974, "The Numerical Computation of Turbulent Flows" Computer Methods in Applied Mechanics Engineering Vol. 3 PP. 269-289.
16- M. A. Mehrabian, 2007, "Heat Transfer and Pressure Drop Characteristic of Cross Flow of Air Over a Circular Tube in Isolation and/or in a Tube Bank" The Arabian Journal for Science and Engineering, Vol. 32, Number 2B.

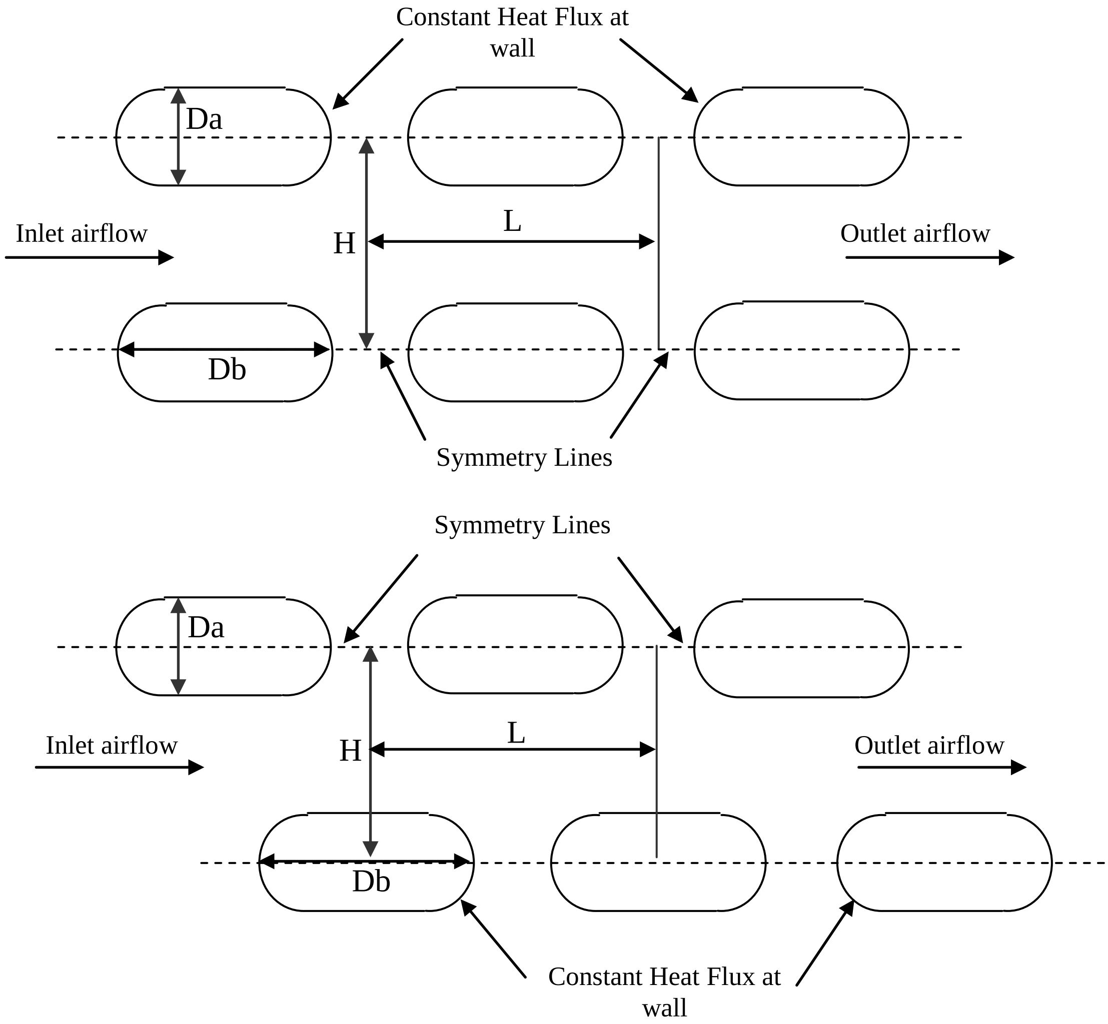

Figure.(1) Physical model and boundary conditions

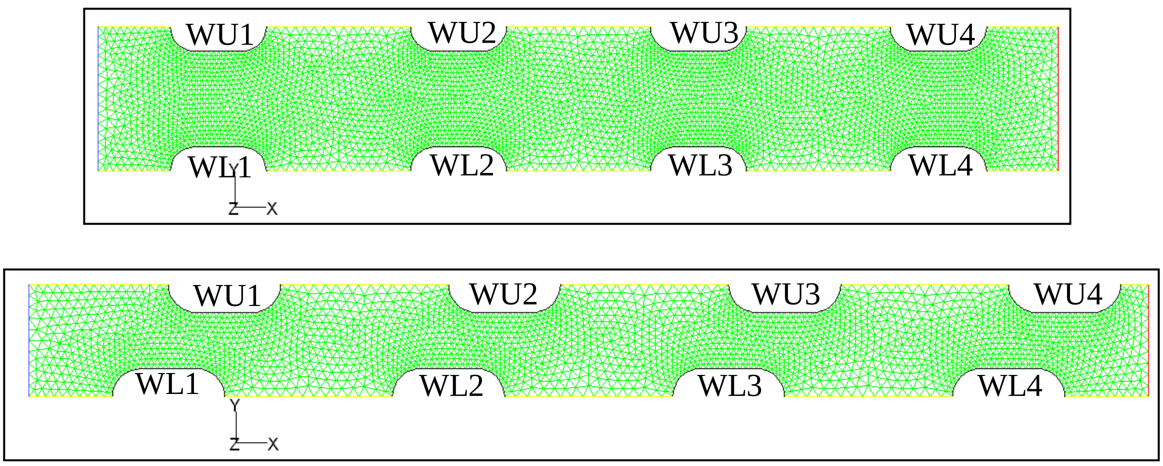

Figure.(2)Two-dimensional model created and meshed in GAMBIT

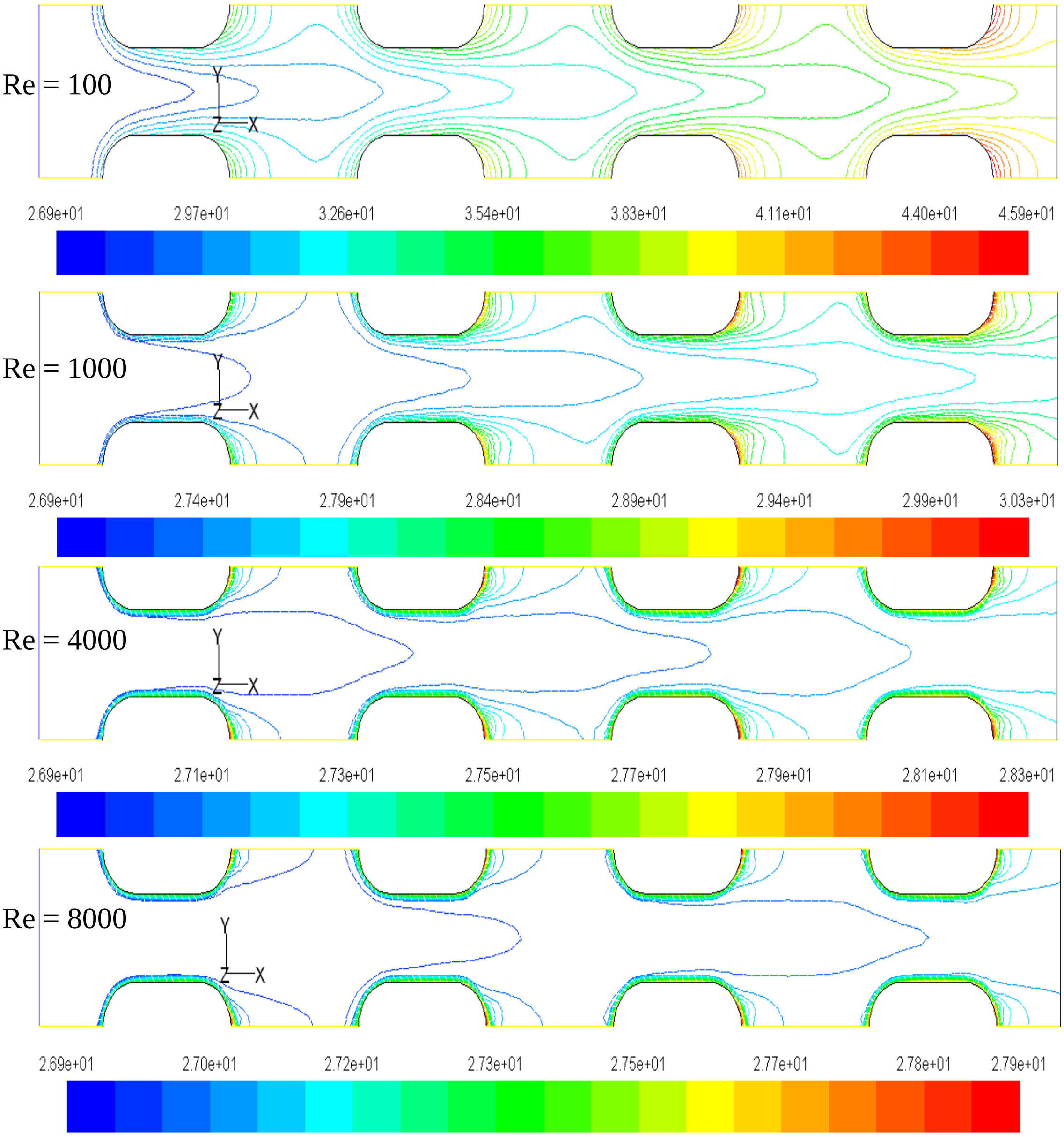

Figure.(3) Effect of Reynolds number on temperature profile for a In-Line arrangement, H/Da = 2, L/Da =4, (100≤Re≤8000)

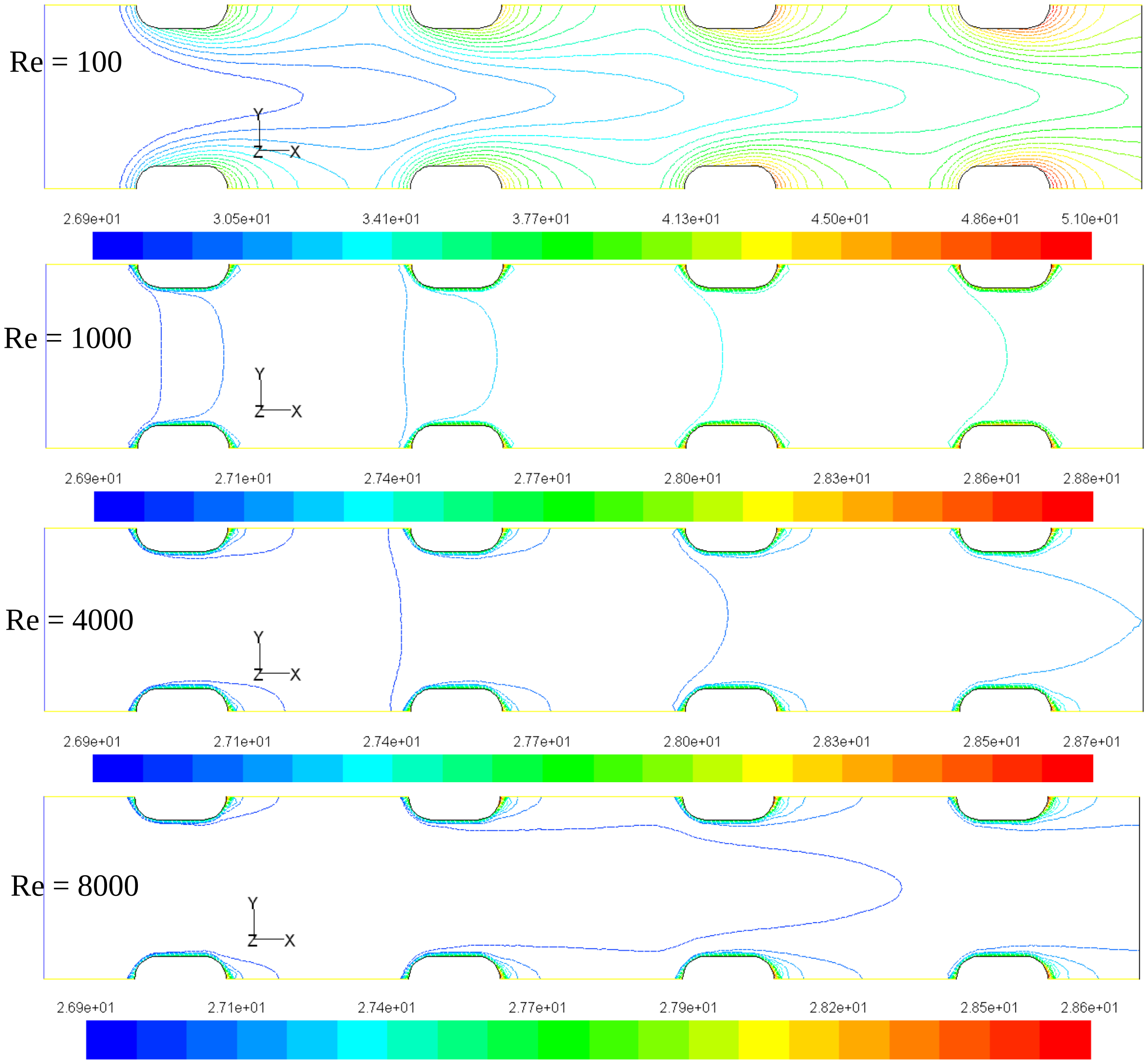

Fig.(4) Effect of Reynolds number on temperature profile for a In-Line arrangement, H/Da =4, L/Da =6, (100≤Re≤8000)

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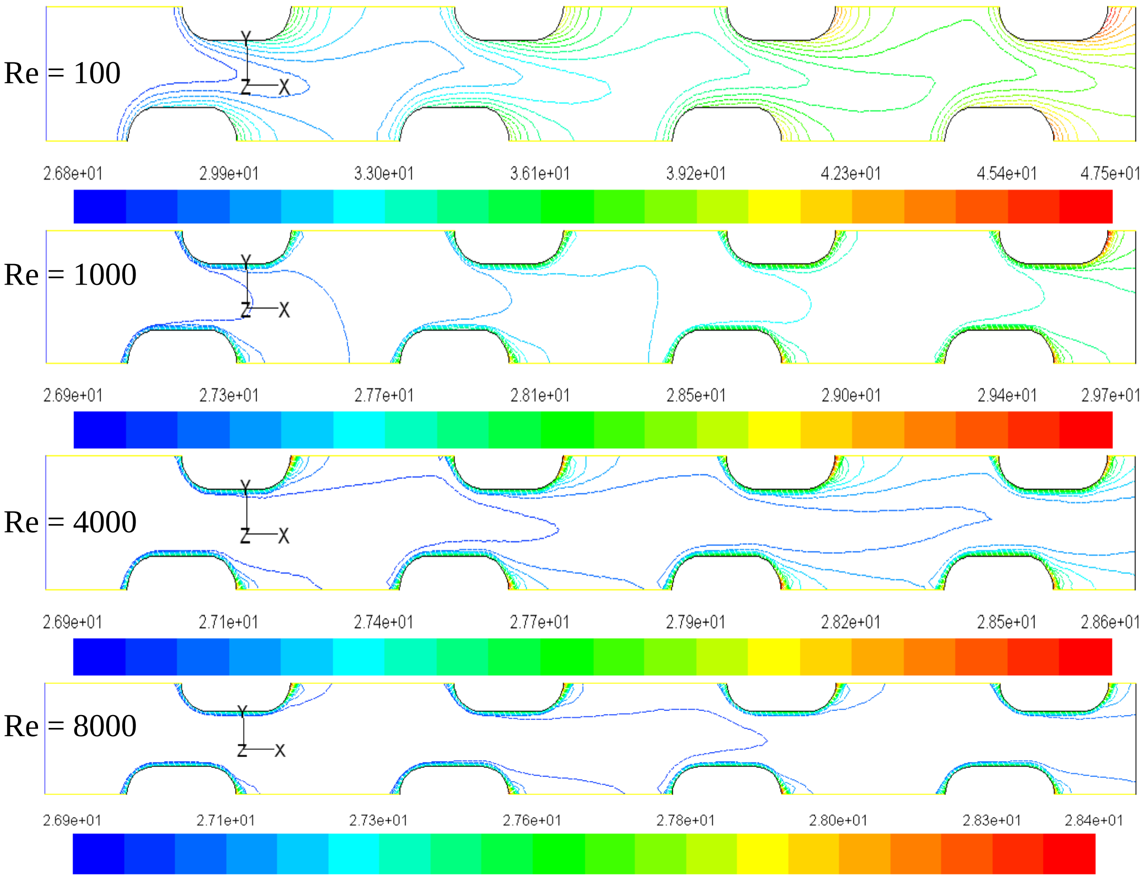

Fig.(5) Effect of Reynolds number on temperature profile for a Staggered arrangement, H/Da =2, L/Da =4, (100≤Re≤8000)

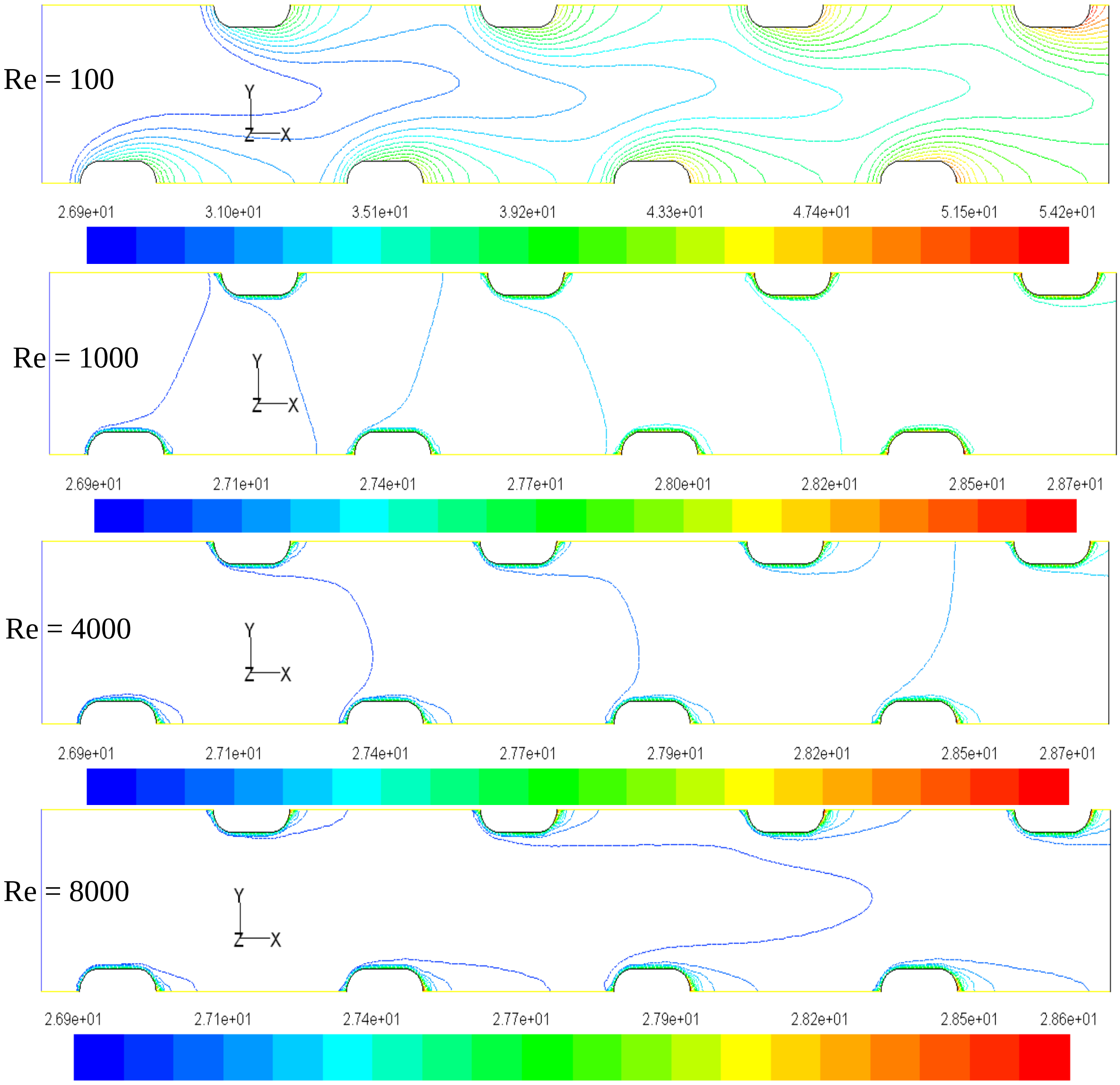

Fig.(6) Effect of Reynolds number on temperature profile for a Staggered arrangement, H/Da =4, L/Da =6, (100≤Re≤8000)

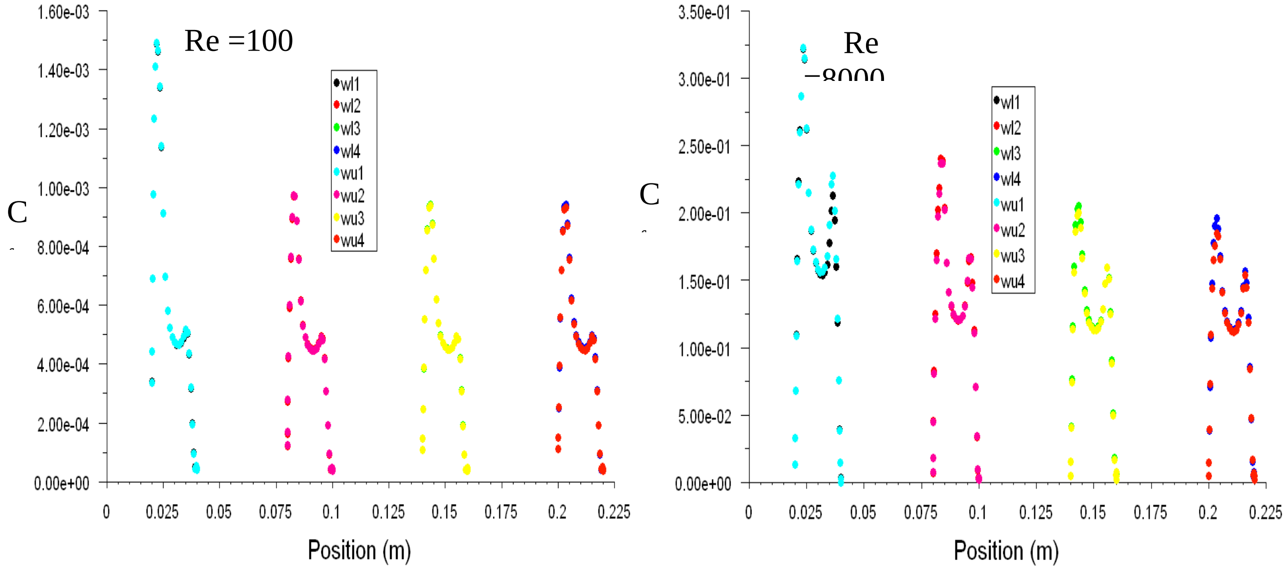

Fig.(7)Effect of Reynolds number on skin friction coefficient along cross section of banks tubes for a In-Line arrangement H/Da =4, L/Da =6 , (100≤Re≤8000)

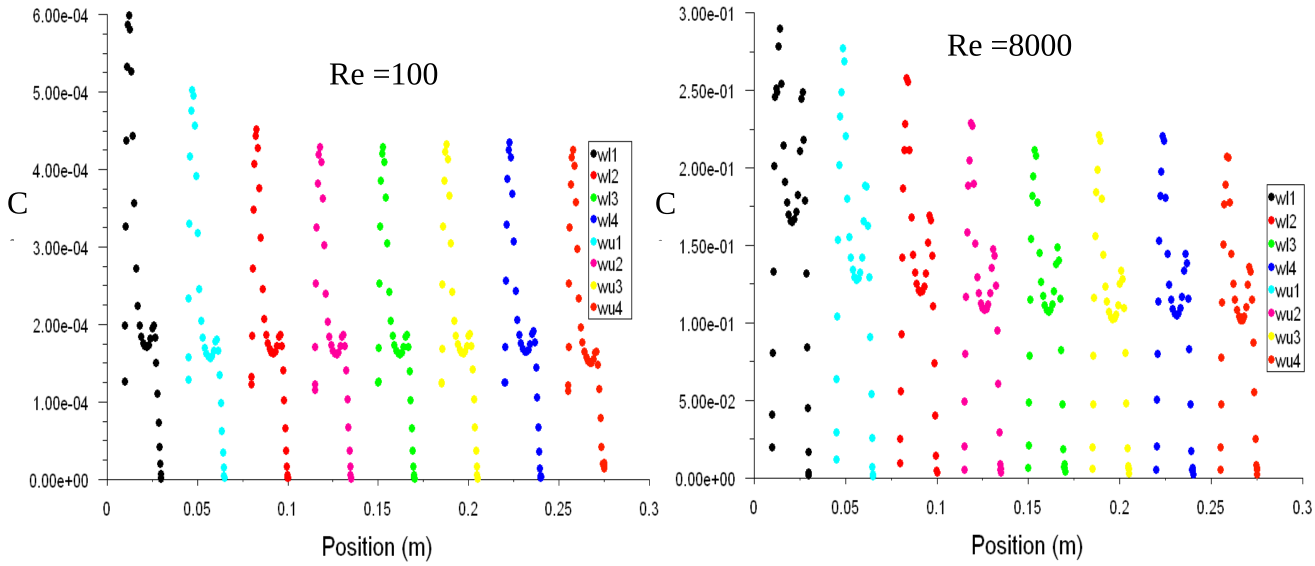

Fig.(8)Effect of Reynolds number on skin friction coefficient along cross section of banks tubes for a Staggered arrangement H/Da =4, L/Da =6 , (100≤Re≤8000)

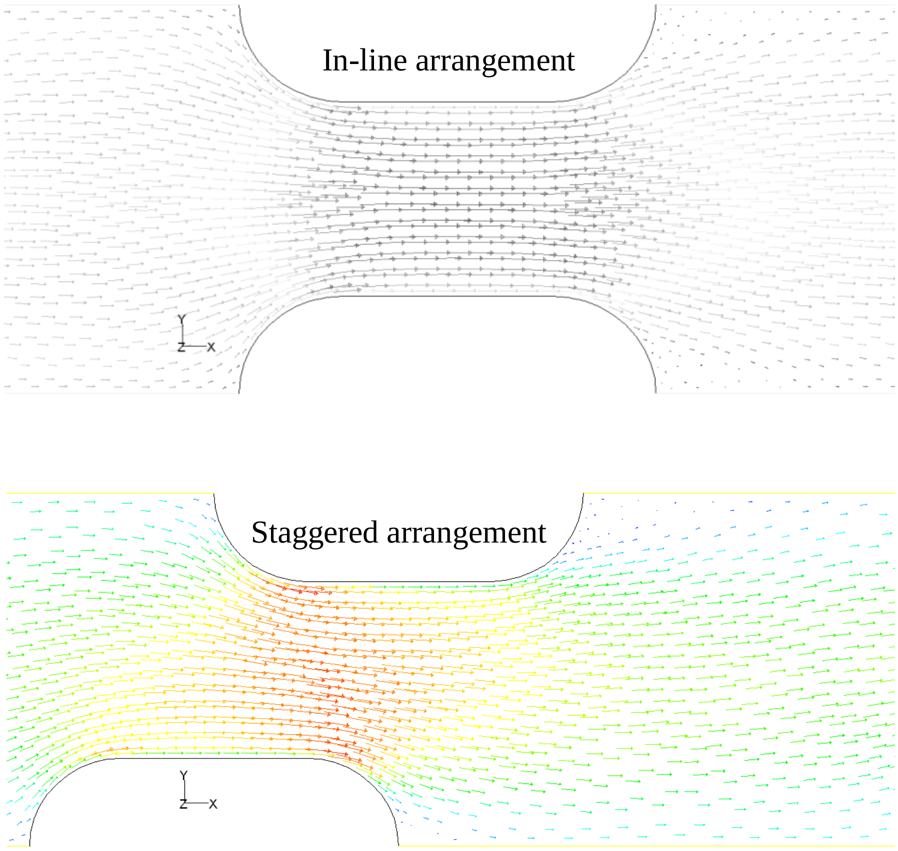

Fig.(9) Velocity vectors profile In-line, staggered arrangement at (Re 8000)

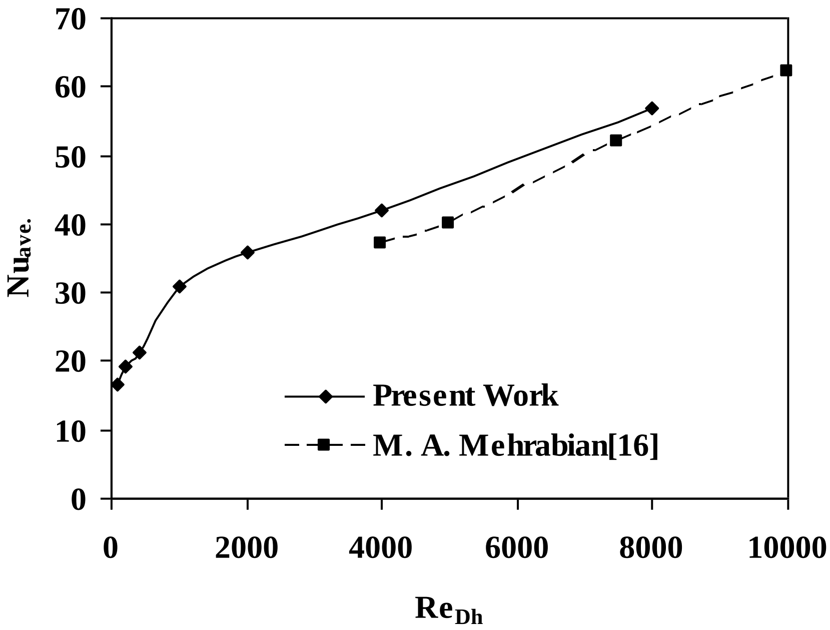

Fig.10 Comparison between the present numerical predictions with the previous work.

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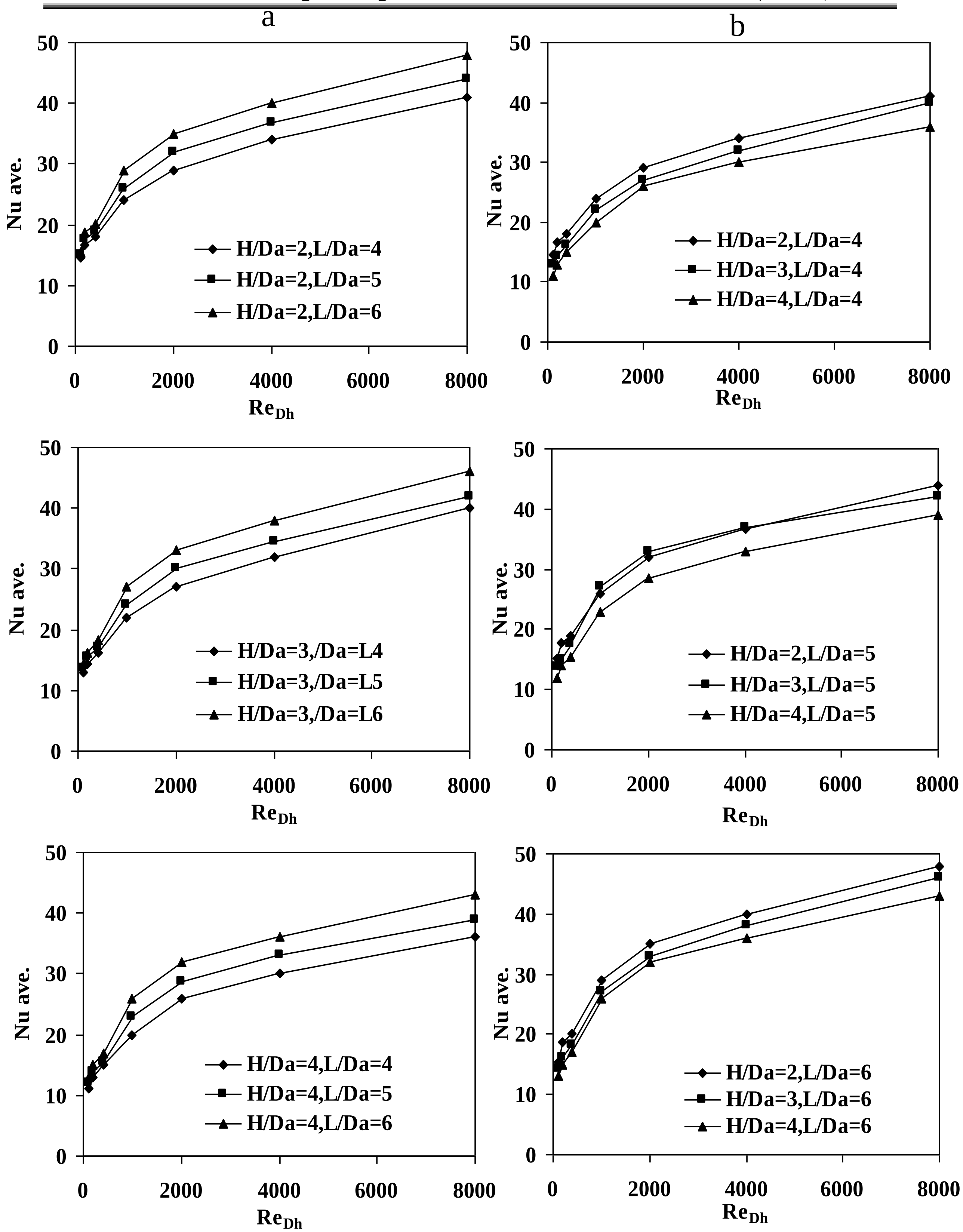

Fig. (11 a, b) Average Nusselt number for in-line arrangements

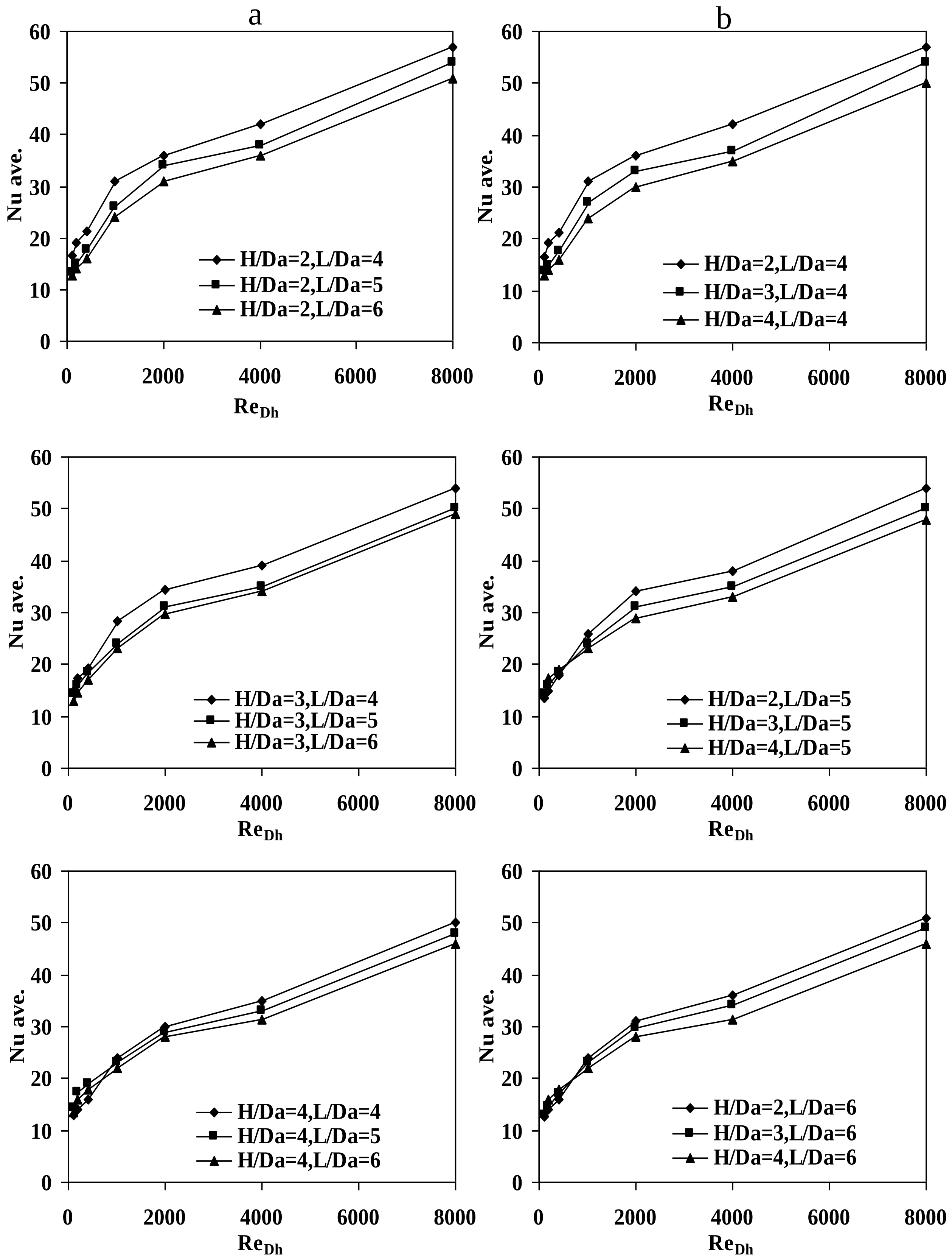

Fig. (12 a, b) Average Nusselt number for staggered arrangements