

Heat Transfer Characteristics and Fluid Flow past Staggered Flat-Tube Bank Using CFD

Zeinab Sayed Abdel-Rehim

Abstract—A computational fluid dynamic (CFD-Fluent 6.2) for two-dimensional fluid flow is applied to predict the pressure drop and heat transfer characteristics of laminar and turbulent flow past staggered flat-tube bank. Effect of aspect ratio (H/D) on pressure drop, temperature, and velocity contour for laminar and turbulent flow over staggered flat-tube bank is studied. The theoretical results of the present models are compared with previously published experimental data of different authors. Satisfactory agreement is demonstrated. Also, the comparison between the present study and others analytical methods for the Re number with Nu number is done. The results show as the Reynolds number increases the maximum velocity in the passage between the upper and lower tubes increases. The comparisons show a fair agreement especially in the turbulent flow region. The good agreement of the data of this work with these recommended analytical methods validates the current study.

Keywords—Aspect ratio (H/D), CFD, fluid flow, heat transfer, staggered arrangement, tube bank, and turbulent flow.

I. INTRODUCTION

A steady flow over staggered tube bank, have been presented by Wang [1]. Convective heat transfer in cross flow is illustrated by Zhukauskas [2].

Numerical laminar and turbulent fluid flow and heat transfer predictions in tube banks are presented by reference [3]. Analysis of laminar forced convection of air for cross flow in banks of staggered tubes is studied, [4].

Heat Transfer from Tubes in Counter Flow, Parallel Flow and Cross Flow is investigated, [5].

This work presents numerical computation using the commercial code (FLUENT 6.0) to predict heat transfer characteristic and fluid flow of laminar and turbulent flow over staggered flat-tube bank for different aspect ratio (H/D).

II. GOVERNING EQUATIONS OF THE PRESENT WORK

A. Laminar Model

For steady, two-dimensional, incompressible, laminar flow of a fluid, with constant property fluid, the differential equations governing the conservation of mass, momentum, and energy can be expressed into a general form as:

$$\text{Continuity: } \frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0.0 \quad (1)$$

$$\text{Momentum: } \rho (\vec{V} \cdot \nabla) \vec{V} = -\nabla P + \mu \nabla^2 \vec{V} \quad (2)$$

Momentum Equation in Two Directions:

$$\frac{\partial(UU)}{\partial x} + \frac{\partial(UV)}{\partial y} = -\frac{\partial P}{\partial x} + \frac{1}{\text{Re}} \left(\frac{\partial^2 U}{\partial^2 x} + \frac{\partial^2 V}{\partial^2 y} \right) \quad (3)$$

$$\frac{\partial(UV)}{\partial x} + \frac{\partial(VV)}{\partial y} = -\frac{\partial P}{\partial y} + \frac{1}{\text{Re}} \left(\frac{\partial^2 U}{\partial^2 x} + \frac{\partial^2 V}{\partial^2 y} \right) \quad (4)$$

$$\text{Energy: } \rho C_p (\vec{V} \cdot \nabla) T = k \nabla^2 T \quad (5)$$

$$\frac{\partial(UT)}{\partial x} + \frac{\partial(VT)}{\partial y} = \frac{1}{\text{Re Pr}} \left(\frac{\partial^2 U}{\partial^2 x} + \frac{\partial^2 V}{\partial^2 y} \right) \quad (6)$$

Equations (1)-(5) above are referred to as the Navier-Stokes equation for the general constant property incompressible flow of a Newtonian fluid. The Navier-Stokes equation in the above form constitute a system of two equations (in two-dimensional flow) with two unknowns, P and \vec{V} .

Correlation

The correlations have been developed for the predication of the local Nu, the importance lies in the overall average heat transfer coefficient. The empirical correlation proposed by Hilpert [6] for the overall Nu is widely used as:

$$\bar{N} u_D = \frac{\bar{h} D}{k} = C \cdot \text{Re}_D^m \cdot \text{Pr}^{\frac{1}{3}} \quad (7)$$

where; C and m are the model constants and Pr=0.71 (air).

B. Turbulence Model

2d segregated, k-ε solver is used to solve the governing differential equations for the conservation of the mass, momentum and energy equations.

For incompressible flow, the Navier-Stokes equations, the continuity, momentum and energy equations can be written as:

$$\frac{\partial V_i}{\partial X_i} = 0.0 \quad (8)$$

Z. S. Abdel-Rehim is with Mechanical Engineering Department, National Research Center, (phone: 01222174362, e-mail: abdelrehimz@yahoo.com).

$$\frac{\partial V}{\partial t} + V_j \frac{\partial V_i}{\partial X_j} = -\frac{1}{\rho} \frac{\partial P}{\partial X_i} + \nu \nabla^2 V_i \quad (9)$$

$$\frac{\partial T}{\partial t} + V_j \frac{\partial T}{\partial X_j} = K \nabla^2 T \quad (10)$$

where $\nu = \frac{\mu}{\rho}$ being the kinematic viscosity coefficient

and ∇^2 is denoting the Laplace operator. The turbulence modeling was performed using the standard k- ϵ model, an inbuilt module in the commercial code (FLUENT 6.0). The standard k- ϵ model is an empirical model based on model transport equations for the turbulence kinetic energy (k) and the specific dissipation rate (ϵ). The model transport equations for k and ϵ are developed as explained in Pope [7]. They are:

$$\frac{\partial k}{\partial t} + (U) \times \nabla k = P - \epsilon + \vec{\nabla} \times \left(\frac{v_T}{\sigma_k} \nabla k \right) \quad (11)$$

$$\frac{D \epsilon}{D t} = \nabla \times \left(\frac{v_T}{\sigma_\epsilon} \nabla \epsilon \right) + C_{\epsilon 1} \frac{P \epsilon}{k} - C_{\epsilon 2} \frac{\epsilon^2}{k} \quad (12)$$

where

$$v_T = \frac{C_\mu k^2}{\epsilon} \quad (13)$$

where the standard values for all the model constants due to Launder and Sharma [8], are $C_\mu = 0.09$, $C_{\epsilon 1} = 1.44$, $C_{\epsilon 2} = 1.92$, $\sigma_k = 1.0$, and $\sigma_\epsilon = 1.3$.

C. Boundary Conditions and Near Wall Treatments

At inlet, uniform profiles for all dependent variables. At wall, a non-slip boundary condition is imposed on the wall of the tube. The velocity used in the simulation is 1.0m/s. This is typical air velocity value for 0.05m tube diameter heat recovery application. In heat recovery applications the gas flow over the tube should be a hot stream. The approach air flow temperature was set at 300°K and the tube's inner wall set to 400°K. Very close to the walls, viscous damping reduces the tangential velocity fluctuations, while kinematic blocking reduces normal fluctuations.

However, towards the outer part of the near-wall region, the turbulence is rapidly augmented by the production of turbulence kinetic energy, due to the large gradients in the mean velocity.

D. Correlation

Correlations for overall Nusselt number and dimensionless pressure drop have been developed as functions of Reynolds number and aspect ratio ($\frac{H/D}{L/D}$) for the entire flat tube-bank for staggered configurations with isothermal boundary condition,

using multiple linear regressions. The general forms of the correlations, [9], are:

$$Nu = a \text{Re}^b \left(\frac{H}{D} \right)^c \left(\frac{L}{D} \right)^d \quad (14)$$

$$P_d = e \text{Re}^f \left(\frac{H}{D} \right)^g \left(\frac{L}{D} \right)^h \quad (15)$$

where the coefficients and constants for proposed correlations for dimensionless Nusselt number and pressure drop are given as: a=0.0073, b=0.9156, c=1.4263, d=0.0211, e=23659.72, f=0.8030, g=-1.8734, and h=-1.0407. The excellent R² values confirm the statistical goodness of the fit, which R²=0.92, [10], [11]. Also, in the form of Nusselt numbers so that a comparison of the present data to other correlations is facilitated as:

$$Nu = 0.044 \left(\frac{S_T}{S_L} \right)^{0.2} \left(\frac{s}{D} \right)^{0.18} \left(\frac{h}{D} \right)^{-0.14} \text{Re}^{0.8} \quad [2], (16)$$

$$Nu = 0.45 \left(\frac{A}{A_i} \right)^{-0.375} \text{Re}^{0.625} \text{Pr}^{\frac{1}{3}} \quad [12], (17)$$

$$Nu = 0.23 K \phi^{0.2} \left(\frac{D}{s} \right)^{-0.54} \left(\frac{h}{s} \right)^{0.14} \text{Re}^{0.625} \quad [13], (18)$$

III. RESULTS AND DISSECTIONS

Fig. 1 shows grid of laminar flow for Re =400 for the selected different aspect ratio ($\frac{H/D}{L/D}$). Pressure drop of laminar flow with Re number for different aspect ratio ($\frac{H/D}{L/D}$) is illustrated in Fig. 2. As the aspect ratio ($\frac{H/D}{L/D}$) decreases the normalized pressure drop decreases. Pressure drop contour for turbulent flow with Re number for different aspect ratio ($\frac{H/D}{L/D}$) is shown in Fig. 3. It is noticed that as the aspect ratio ($\frac{H/D}{L/D}$) increases the normalized pressure drop decreases. The higher values of pressure drop distribution usually recorded near the initial of the external tube surface in all cases. Fig. 4 shows static temperature contour of laminar flow with Re number for different aspect ratio ($\frac{H/D}{L/D}$). This behavior is a function of the ratio of the pressure drop to velocity squared terms in the calculation of the temperature distribution. Static temperature contour for turbulent flow with Re number for different aspect ratio ($\frac{H/D}{L/D}$) is depicted in Fig. 5. The contours expose the temperature increase in the fluid due to heat transfer from the tubes. The fluid is heated in the near wall and

wake regions. Figs. 6 and 7 show velocity vector contour of laminar and turbulent flows with Re number for different aspect ratio $(\frac{H/D}{L/D})$. As the Reynolds number increases the maximum velocity in the passage between the upper and lower tubes increases. The comparisons between the present work and different empirical correlations for Nu number with Re, [5] and [14], for laminar and turbulent flow and for the case $(\frac{H/D}{L/D}) = \frac{2}{5}$ are carried out in Fig. 8. The figure shows a fair agreement especially in the turbulent flow region. The theoretical work indicates the different flow zones namely: laminar, transient and turbulent with different slopes while the correlations have a fixed form for the whole regions. The comparison between the present study and others analytical methods for the Re number with Nu number is shown in Fig. 9. It was believed that the good agreement of the data of this work with these recommended analytical methods validates the current study.

IV. CONCLUSIONS

The higher values of pressure drop distribution usually recorded near the initial of the external tube surface in all cases. The contours expose the temperature increase in the fluid due to heat transfer from the tubes. The fluid is heated in the near wall and wake regions. As the Reynolds number increases the maximum velocity in the passage between the upper and lower tubes increases. The comparisons show a fair agreement especially in the turbulent flow region. The theoretical work indicates the different flow zones namely: laminar, transient and turbulent with different slopes while the correlations have a fixed form for the whole regions.

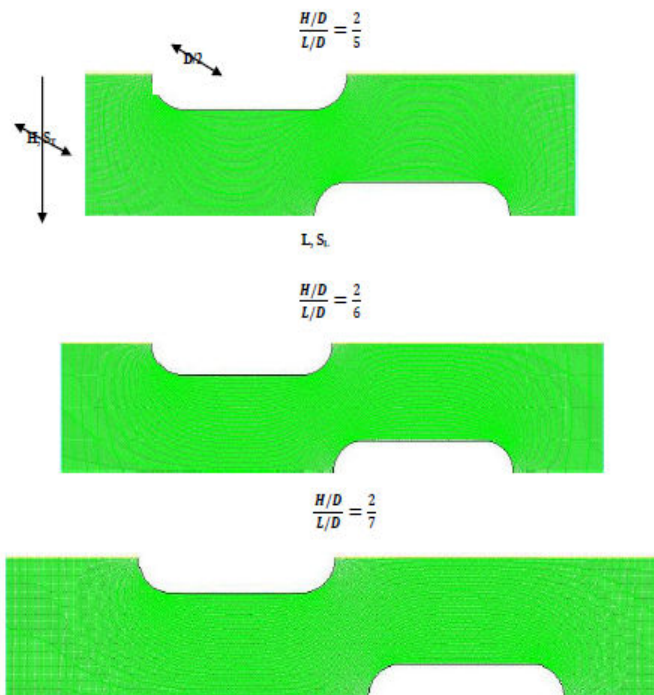


Fig. 1 Grid of laminar flow for Re =400

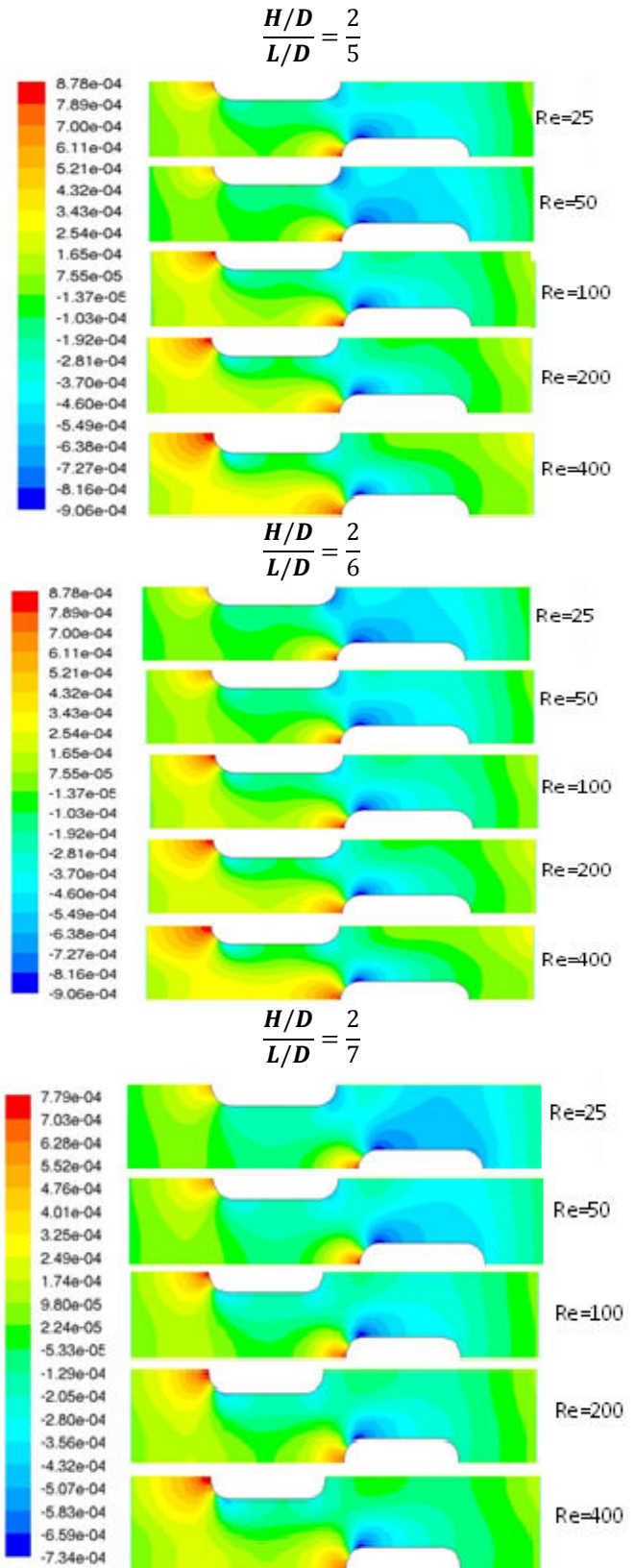


Fig. 2 Pressure drop contour of laminar flow with Re number for different aspect ratio $(\frac{H/D}{L/D})$

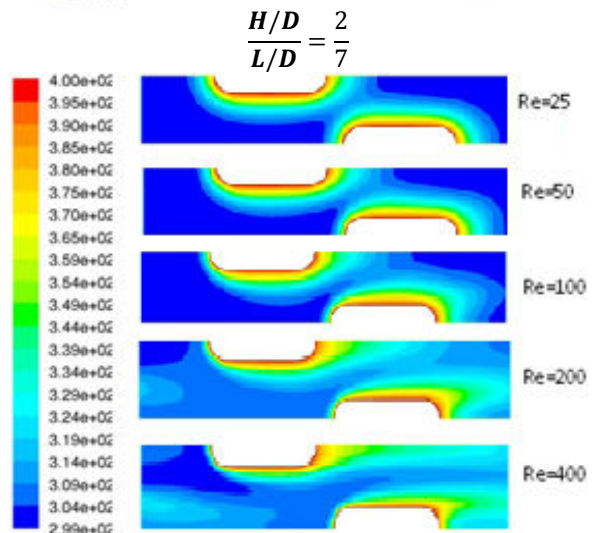
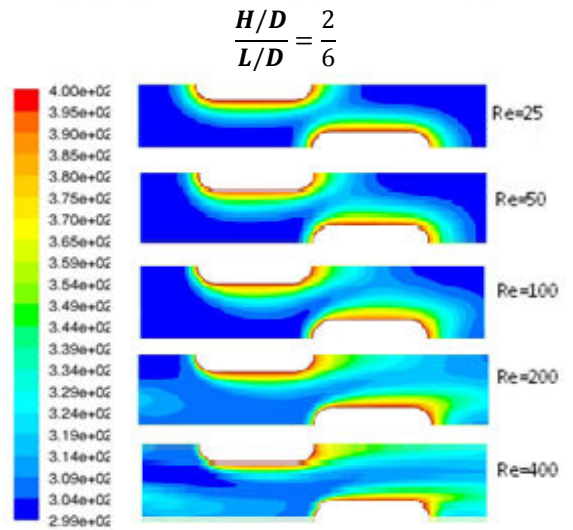
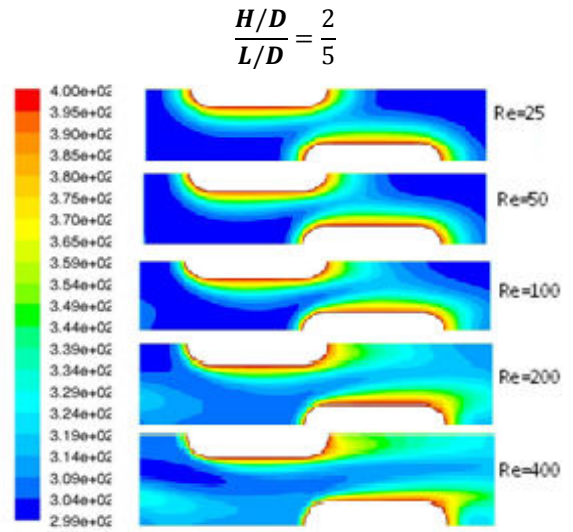
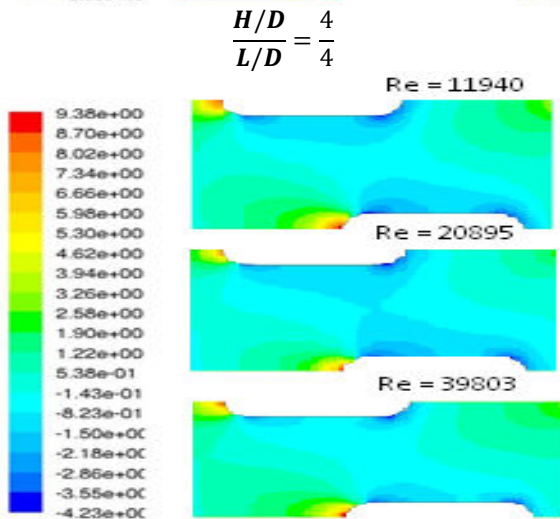
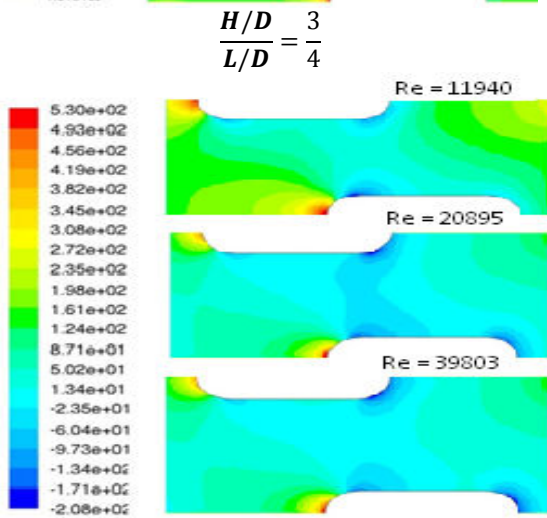
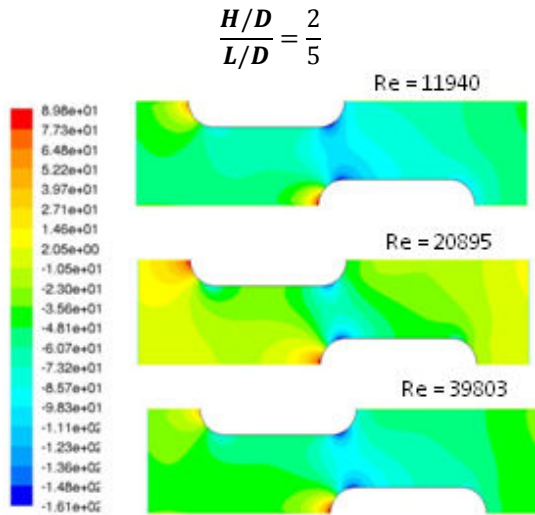


Fig. 3 Pressure drop contour for turbulent flow with Re number for different aspect ratio ($\frac{H/D}{L/D}$)

Fig. 4 Static temperature contour of laminar flow with Re number for different aspect ratio ($\frac{H/D}{L/D}$)

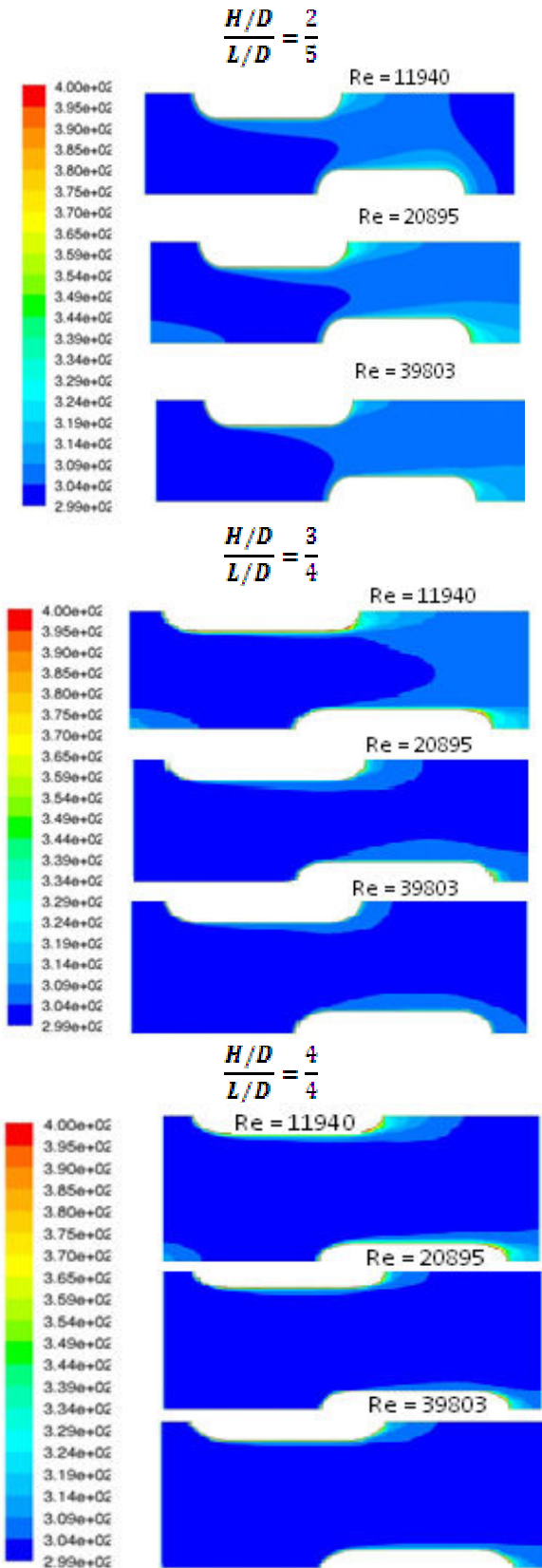


Fig. 5 Static temperature contour for turbulent flow with Re number for different aspect ratio ($\frac{H/D}{L/D}$)

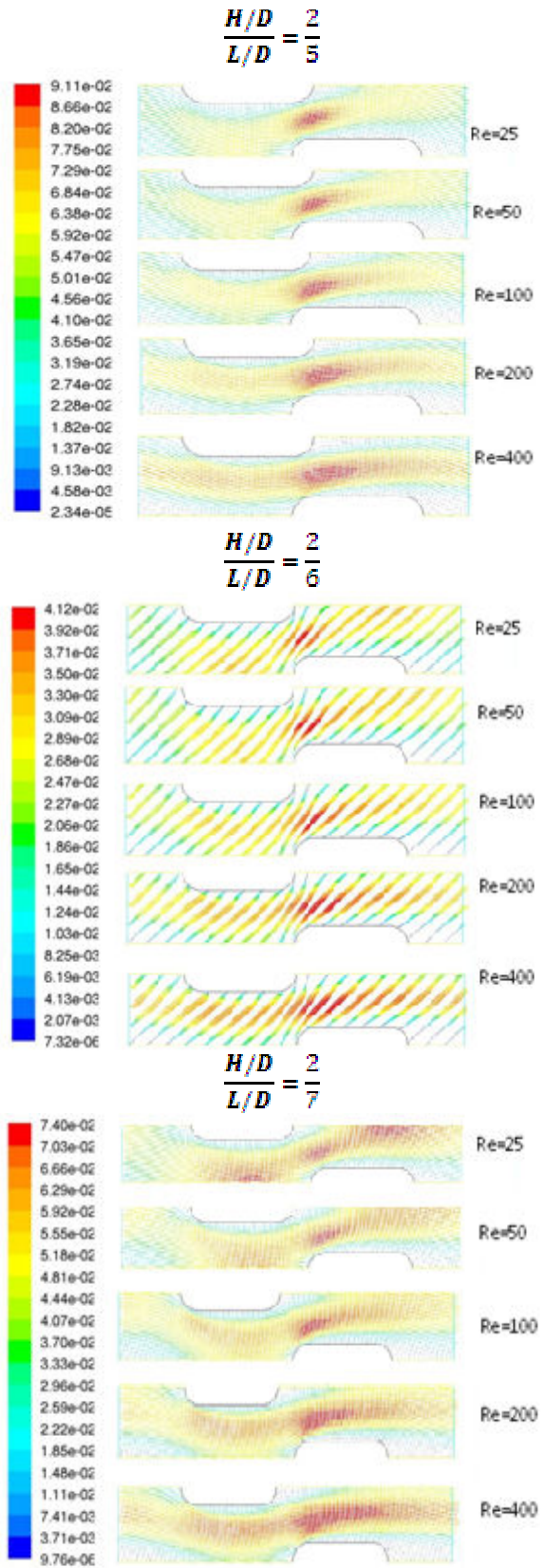


Fig. 6 Velocity vector contour of laminar flow with Re number for different aspect ratio ($\frac{H/D}{L/D}$)

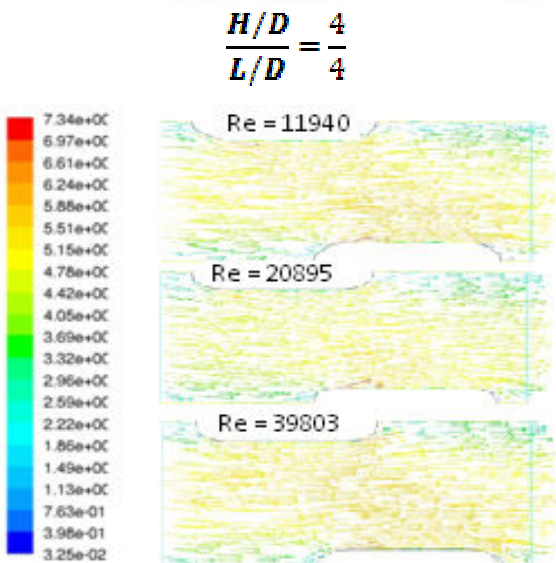
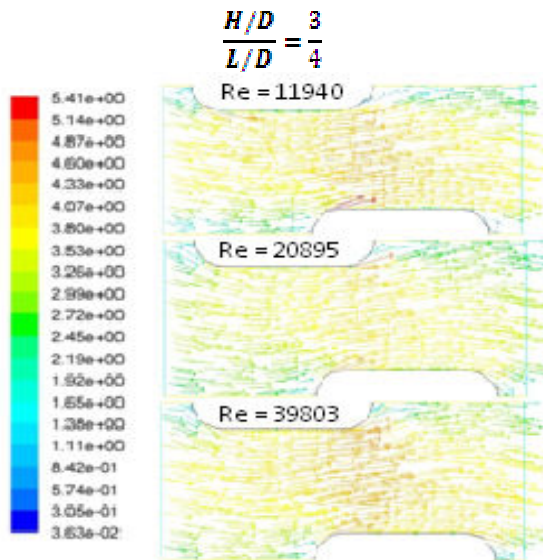
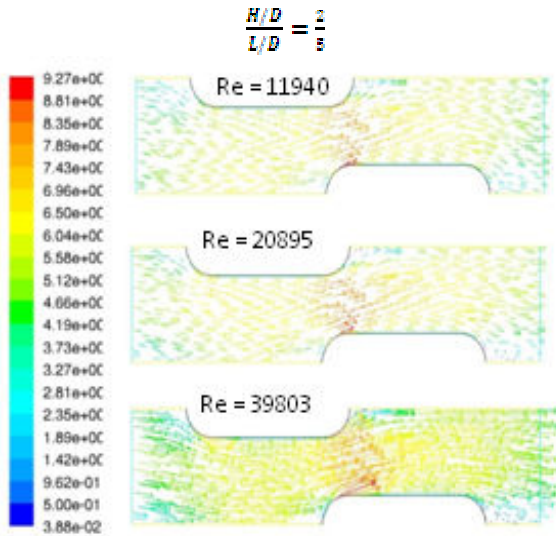


Fig. 7 Velocity vector contour for turbulent flow with Re number for different aspect ratio $\left(\frac{H/D}{L/D}\right)$

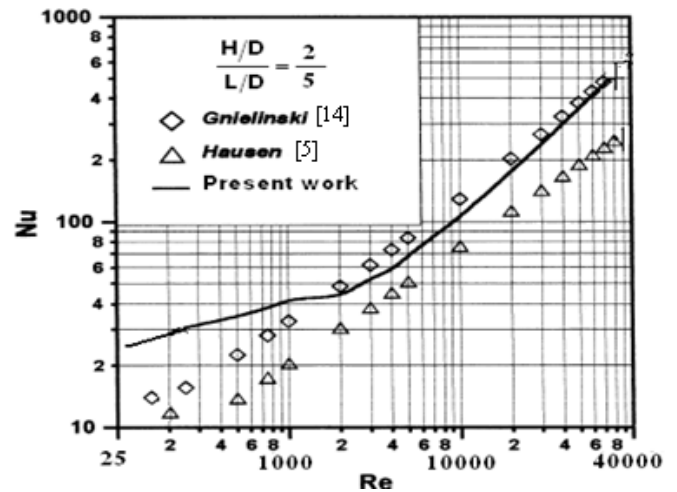


Fig. 8 Comparison between the present work and different empirical correlations for Nu number with Re for laminar and turbulent flow and for the case $\frac{H/D}{L/D} = \frac{2}{5}$

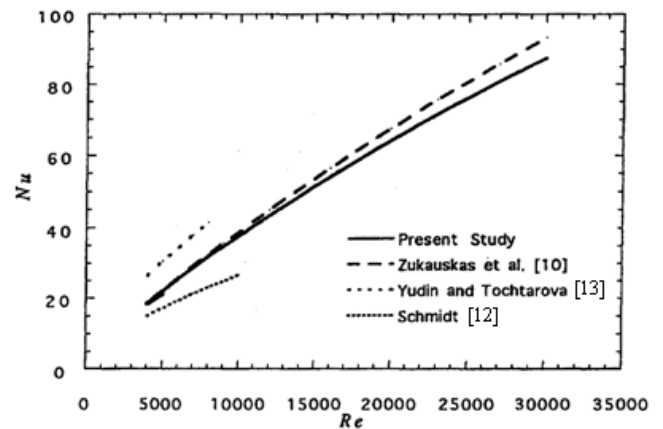


Fig. 9 Comparison between the present study and others of analytical methods for different Re number with Nu number

V. NOMENCLATURE

- A, A_t total heat transfer surface and tube surface area, m
 - C_p specific heat, J.kg⁻¹.K⁻¹
 - D diameter, m
 - H height of control volume, m
 - h heat transfer coefficient, W.m⁻². K⁻¹
 - K thermal conductivity, W. m⁻¹. K⁻¹
 - L length, m
 - Nu Nusselt number ($=h Dh / k$)
 - P Pressure drop, pa
 - Pr Prandtl number ($= \mu C_p / k$) = 0.71 (air)
 - Re Reynolds number ($= \rho V D_h / \mu$)
 - s tube spacing, m
 - t time, sec
 - T temperature, K
 - U, V Cartesian components of fluid velocity, m.s⁻¹
- Greek symbols**
- ϕ surface extension factor, (A/A_t)

μ Dynamic viscosity, N.s.m⁻²

ρ Mass density, kg.m⁻³

Subscript: h hydraulic,

Superscript: - Average value

REFERENCES

- [1] Wang, Y. Q., "Laminar flow through a staggered tube bank", *Journal of Thermo-physics and Heat Transfer*, 18: 4, 2004, pp. 557–559.
- [2] Zhukauskas A., "Convective heat transfer in cross flow", *Handbook of single-phase convective heat transfer*, Wiley & Sons, New York, 1987.
- [3] Zdravistich F., C. Fletcher, M. Behnia, "Numerical laminar and turbulent fluid flow and heat transfer predictions in tube banks", *Int. J. Num. Meth. Heat Fluid Flow* 5: 8, 1995, pp. 717–733.
- [4] Wang, Y. Q., Penner, L. A., and Ormiston, S. J., "Analysis of laminar forced convection of air for cross flow in banks of staggered tubes", *Numerical Heat Transfer, Part A, Applications*, 38: 8, 2000, pp. 819–845.
- [5] Hausen H., "Heat Transfer from Tubes in Counter Flow, Parallel Flow and Cross Flow", McGraw H. -Hill, USA, 1983, p. 54.
- [6] Hilpert, R., *Wärmeabgabe Von geheizten Dröhnten und Röhren*, *Forsch. Geb. Ingenieurwes*, 4, 1933, pp. 215-224.
- [7] Pope, S. B., "Turbulent flows", Cambridge University Press: Cambridge, 2000.
- [8] Launder B. E. and Sharma B. I., "Application of the energy-dissipation model of turbulence to the calculation of flow near a spinning disc", *Letters in Heat and Mass Transfer*, 1, 1974, pp. 131-138.
- [9] Benarji N., C. Balaji, S. P. Venkateshan, "Unsteady fluid flow and heat transfer over a bank of flat tubes", *Heat Mass Transfer*, 2008.
- [10] Zhukauskas A. "Heat transfer from tubes in cross flow", *Advances in Heat Transfer*, 8, 1972, pp. 93-160.
- [11] Bahaidarah HMS, Anand NK, Chen HC, "A numerical study of fluid flow and heat transfer over a bank of flat tubes", *Heat Transfer Part A* 48, 2005, pp. 359–385.
- [12] Schmidt Th., "In Heat transfer of finned tube bundles in cross flow", eds. Zukauskas A., and Hewitt, G., p. 172, Hemisphere Publishing, Washington, D.C., 1988.
- [13] Yudin V. F., and Tochtarova, L. S., "In Heat transfer of finned tube bundles in cross flow", eds. Zukauskas A., and Hewitt, G., p. 173, Hemisphere Publishing, Washington, D.C., 1988.
- [14] Gnielinski V., *Wärmeübergang bei Querströmung durch einzelne Rohrreihen und Rohrbindel*, VDI-Wärmeatlas, 2. Auflage, Abschnitt Ge, 1974.