Jurnal
Teknologi

Full Paper

NUMERICAL EVALUATION OF THERMO-HYDRAULIC PERFORMANCE IN FIN-AND-TUBE COMPACT HEAT EXCHANGERS WITH DIFFERENT TUBE CROSS-SECTIONS

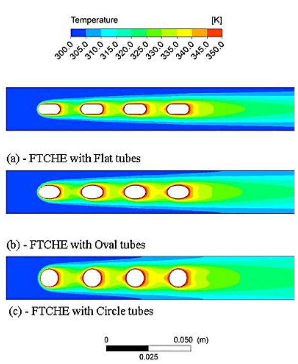
Ahmadali Gholami*, Mazlan Abdul Wahid, Hussein A. Mohammed, A. Saat, M.F. Mohd Yasin, M.M. Sies, N. Kamaruzaman, Md. Mizanur Rahman

High Speed Reacting Flow Laboratory, Department of Thermofluids, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, 81310 UTM Johor Bahru, Johor, Malaysia

Article history
Received
1 January 2016
Received in revised form
18 May 2016
Accepted
15 June 2016

*Corresponding author
agahmad2@live.utm.my

Graphical abstract



(a) - FTCHE with Flat tubes
(b) - FTCHE with Oval tubes
(c) - FTCHE with Circle tubes

Abstract

This study examined numericallythe Thermal-hydrodynamic properties of airflow in the fin-and-tube compact heat exchangers (FTCHEs) with considering different shapes of tubes in lowReynoldsnumbers. The influence of applying flat, oval and circular tube adjustments on the thermal and hydraulic characteristics of air flow were analyzed on the in-line tube arrangements. Establishing standard conditions, the study compared different geometries based on circular tubes of 10.459 mm diameter tubes with 25.4 mm longitudinal pitches and 25.4 mm transverse pitches. The other geometries of tubes were assumed in a stable and constant state preparing the same heat transfer surface area per unit volume as that of the nominal case. The results showed that the FTCHE with flat tubes gives the best area goodness factor (j/f) with in a certainrange of Reynoldsnumbers. In addition, FTCHE with flat tubes shown the best thermo-hydraulic performance and a significant augmentation of up to 10.83% and 35.63% in the average area goodness factor achieved accompanied by a decrease in the average friction factor of 17.02% and 43.41% in the flat tube case compared to the oval and circle tube shapes, respectively. It is concluded that the average area goodness factorfor the oval tube is about 25.04% higher than that of the circular tube, while the average friction factor for the oval tube is about 26.9% lower than that of the circular tube. This means that the flat tube has a better-combined thermal–hydraulic performance than the oval and circle tube.

Keywords: Fin-and-tube compact heat exchanger, Oval, Flat, Heat transfer enhancement, thermo-hydraulic performance, Area goodness factor

Abstrak

Penyelidikan ini mengkaji sifat-sifat termo-hidrodinamik aliran udara di penukar haba padat tiub-dan-sirip (FTCHEs) secara kaedah berangka dengan mengambil kira pelbagai bentuk tiub pada angka Reynolds yang rendah. Kesan daripada pelarasan pada tiub rata, bujur dan bulat ke atas ciri-ciri haba dan hidraulik aliran udara dianalisis pada susunan tiub selari. Perbandingan geometri tiub yang berbeza dilakukan dengan keadaan piawai pada tiub bulat bergaris pusat 10.459 mm dengan anggul membujur 25.4 mm dan anggul melintang 25.4 mm. Geometri lain dianggap dalam keadaan stabil dan mantap dengan permukaan pemindahan haba se unit isipadu seperti mana kes namaan. Hasil kajian menunjukkan bahawa FTCHE dengan tiub rata memberikan faktor kebaikan kawasan (j/f) yang terbaik dalam beberapa julat angka Reynolds. Di samping itu, FTCHE dengan tiub rata menunjukkan prestasi termo-hidraulik yang terbaik dan peningkatan yang ketara sehingga mencapai 10.83% dan 35.63% bagi faktor kebaikan kawasan purata diikuti dengan pengurangan dalam faktor geseran purata 17.02% dan 43.41% dalam kes tiub rata berbanding dengan kes tiub bujur dan bulat. Kesimpulannya ialah faktor kebaikan kawasan purata untuk tiub bujur adalah sekitaran 25.04% lebih tinggi daripada tiub bulat, manakala purata faktor geseran untuk tiub bujur adalah sekitaran 26.9% lebih rendah daripada tiub bulat. Ini bermakna tiub rata mempunyai prestasi gabungan termo-hidraulik lebih baik daripada tiub bujur dan tiub bulat.

Kata kunci: Penukar haba padat tiub-dan-sirip, bujur, rata, peningkatan pemindahan haba, prestasi termo-hidraulik, faktor kebaikan kawasan

1.0 INTRODUCTION

Configuring and adjusting Flow in tube banks has gained momentum in different engineering Applications,such as cooling, heating and ventilation. The global objective of the FTCHE design is a significant augmentation of convective heat transfer with minimal increases in the streamwise pressure drop penalties.This is related to significance of consideringthe air side convection resistance while liquid fluid flow turns into air flow in heat exchangers. The resistance is related to the thermo-physical properties of air leading to cause the heat transfer coefficients being remarkably lower incase of the gas side compared to those of liquid or two phase flow cases.Such limitation accompanied with rising need for enhancing the efficiency of energy utilization for reducing the manufacturing coststo face the turbulentenvironment have caused theresearch in gas side heat transfer enhancement to receive growing attention from researchers.Inattempts done to meet this need,various forms of fin patterns and different tube geometries have been proved to function as a way of improving the air side heat transfer performance[1–3].

Hasan and Siren [4] investigated the performance of two various evaporative cooled heat exchangers with circular and oval tubes which had an equivalent perimeter. Consideringsimilar operating conditions of airflow rates and inlet hot water temperatures, andintegrating the thermal–hydraulic properties of the tubes, they approved that the oval tube shows better results for the ratios of (j/f)1.930–1.960 times compared to that of the circular tube, considered as a superiorcharacteristic for the pressure drop.

In the other study conducted by Horvat et al.[5]fluid flow and heat transfer have beennumerically examined with cylindrical, elliptical and wing-shaped tubesarrangement in the tube bundle crossflow for the almost 100 analyzed cases. The comparison of the collected results indicated that the large flow fluctuations and stochastic motion of the flow in the spanwise direction was observed as the flow regime changes. This was particularly more dominant in case of the wing-shaped tubes.

In other similar vein of numerical investigation conducted by ,Sayed Ahmed et al.[6]the thermal heat transfer properties and fluid flow of a cross flow heat exchanger was examined through incorporating the staggered wingshaped tubes treated by different angles of attack. The outcomes revealed thatthe best thermal performance and efficiency appeared at zero angle of attack. The recent stream of research in this area has addressed application of the numerical techniquesinorder to investigate the fluid flow and heat transfer under periodic circumstances , in which the appropriate and robust design for evaluation of various geometries have been proposed[7–9].

The preceding literature review shows that only the limited numerical and experimental analyses of the FTCHEs with various forms of tubes have been published with the same conditions. This paper presents a two-dimensional numerical investigation of laminar flow and heat transfer characteristics with constant wall heat flux conditionover a four rows of the tube bank with in-line arrangement of the flat, oval, and circular shapes of tubes in different Reynolds number ranging from Re = 100 to 500. In this article, the effects of three different of tube shapes on heat transfer and pressure drop of anoriginalFTCHE are examined using computational fluid dynamics. For a comparison of FTCHE with various tube geometries, equal wetted perimeter of all tubes per base area will be used to provide the same heat transfer surface area per unit volume.

To the authors' best knowledge the problem of flow across a tube bank of oval, flat, and circular tubes in the FTCHE has not been solved with the same conditions. In this study, the thermo-hydraulic performance of FTCHE is analyzed by numerical simulation and augmentation mechanism.

2.0 METHODOLOGY

2.1 Governing Equations

The fluid is considered to be incompressible with constant thermo-physical properties. The governing equations with negligible viscous dissipation, for numerical approach procedure in this research can be written as equations from Ref[2,3].

Continuity equation:

$$\frac{\partial(\rho u_i)}{\partial x_i}=0. \tag{1}$$

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j)=\frac{\partial}{\partial x_i}\left(\mu\frac{\partial u_j}{\partial x_i}\right)-\frac{\partial p}{\partial x_j}. \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T)=\frac{\partial}{\partial x_i}\left(\frac{k}{C_p}\frac{\partial u_j}{\partial x_i}\right). \tag{3}$$

General transport equation:(for scalars):

$$\frac{\partial(\rho u_i \phi)}{\partial x_i}=\frac{\partial}{\partial x_i}\left[\Gamma_\phi\frac{\partial \phi}{\partial x_i}\right]+S_\phi \tag{4}$$

Eq. (1)-(3) as general equations are used in CFD procedure of calculations to analyze the flow field for both fluid (air) dynamics and thermal characteristics, solving for pressure drop and thermal heat transfer.

2.2 Physical Model and Computational Domain

In this study, three different geometries design of the FTCHE with flat, oval, and circular forms of tubes is shown in Figure 1.

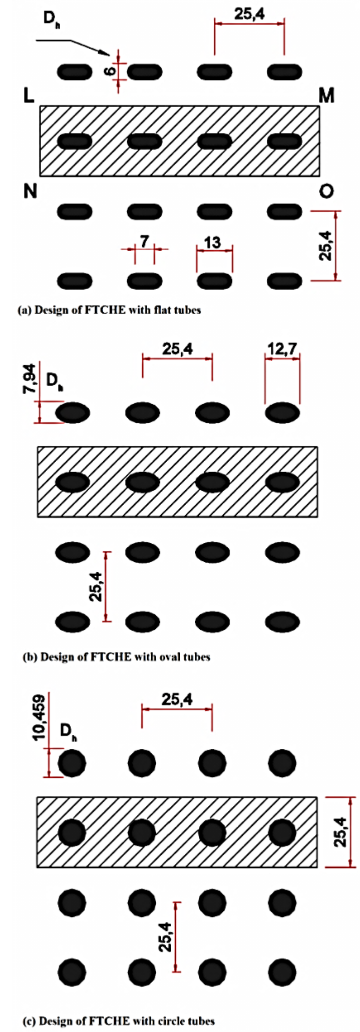


Figure 1 Schematic design in 2D model of FTCHE with different tube geometry

The hydraulic diameter D_h of the computational domain is 10.459 mm, 7.940 mm, and 6.0 mm for circular, oval, and flat tube shapes, respectively. In this study, the perimeter of the three different shapes of tubes were chosen fixed to provide the same heat transfer surface area per unit volume as that of the nominal case.

2.3 Boundary Conditions

Fluid is considered as a substance showing fixedproperties, which cannot be compressed. The same assumption can be taken granted for flow with a laminar form, which derives from the low air inlet velocity and the small inlet space condition making thethe flow to move in the fin channel of FTCHE in a steady state[10]. In this study, the thermal contactresistance between the tube and fin collar was disregarded, and the tube surface is taken into account as a fixed steady temperature. An inlet air flow with the velocity ranging from 0.820 to 0.930 m/s is incorporated to the inlet boundary of the periodic module. The fluid is injected with constant temperature of T_{in} = 300 K While various inlet constant velocities V_{in} are used. At the outlet domain as exit flow region of the computational model,a zero value was assumed for average pressure. A constant temperature T_w = 350 K is specified for the tube wall. At the side boundaries symmetry conditions considered. Thecomputational domainof FTCHE model consists in boundary conditions as shown in Figure 2. In the airflow direction the computational region is extended accordingly to ensure the uniform incoming flow condition at inlet and outflow condition at the exit[11].

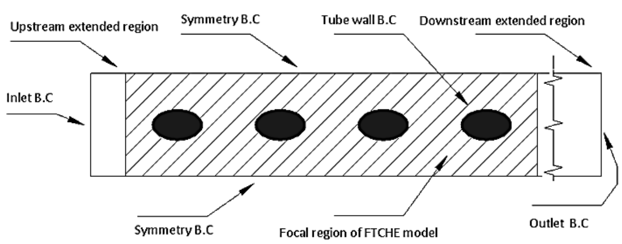


Figure 2 Computational domain including boundary conditions (BC) and extended flow areas

A summary of the outer and inner boundaryconditions practical in the computational domain can be defined for the three sections as follows:

(1) In the upstream side as inlet extended region (inlet domain):

- At outer side of the inlet boundary:

$u = u_{in}$ = const., $v = 0$,
$T = T_{in}$ = 300 K

- At the side boundaries:

Symmetry conditions,
v=0

(2) In the downstream side as outlet extended region

- At the side boundaries:

Symmetry conditions,
v=0

- At the outlet B.C (One-way):$(\partial u/\partial x)=0$,

$(\partial v/\partial x)=0$,
$(\partial T/\partial x)=0$,
$(\partial P/\partial x)=0$

(3) In the main region of model

- At the side boundaries:

Symmetry conditions,
v=0

(4) In the tube wall surfaces:

Velocity state and condition: $u = v = w = 0$
Temperature state and condition: $T = T_{wall}$ = 350 K

2.4 Numerical Method

Applying a computational fluid dynamics approach in ANSYS Workbench [12]., the researcher solved the Navier-Stokes and energy Eq. (1)-(3) with the boundary conditions. In order to solve the governing equations, a segregated, implicit solver option is applied .Besides, this study has incorporated the first order upwind discrimination pattern for the terms in energy, momentum, and laminar flow parameters. Finally, a standard pressure interpolation scheme and SIMPLE algorithm with pressure velocity coupling are applied. The convergence measures and procedure for the velocities with the maximum values of mass residual in the cells divided by the maximum residual of the first 5 iterations is less than 1.0×10^{-5}, and the convergence norm for the energy is that the maximum values of temperature residual in the cells divided by the maximum residual of the first 5 iterations is less than 1.0×10^{-8}.

2.5 Validation of numerical results

This study applied three different grid systems with nodes of 53870, 87378, and 180864 to validate if the solution is independent. The ultimately proved grid number used in the study is 87378 in term of flat form of tubes, in which the Reynolds numbersrange from 100 to 500. Figure 3 demonstrates the various Nusseltvalue of the three grid systems.

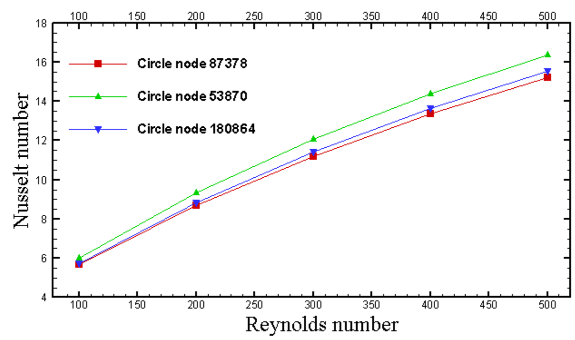


Figure 3 Grid independency criteria

Furthermore, to confirm the dependability of the numerical technique being used, the simulation is ledby a FTCHE with the average of convective heat transfer coefficient at constant surface temperature of solid surface tubes, analytical outcomes accepted by as presented in Ref[8].

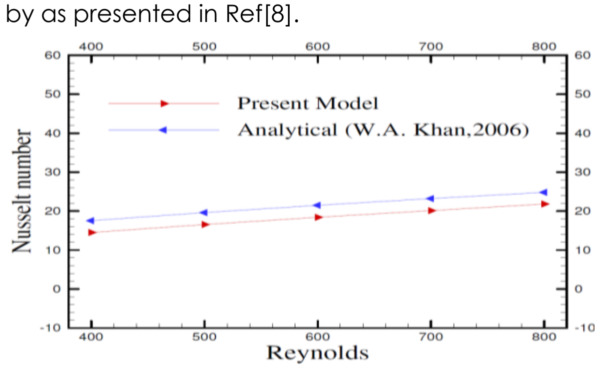


Figure 4 Comparison between numerical and experimental results

The relationship between average Nusselt number and Reynolds number for an in-lined tube bank is shown in Figure4. As can be seen from the figure, the numerical result is in a good agreement with the existing correlations.

3.0 RESULTS AND DISCUSSION

Conducting a numerical analysis of heat transfer performance with Reynolds value of Re = 400 to 800, the study investigated the effect of different forms of tube on the heat transfer properties and flow structure for the FTCHEs. The conductedanalysis aims to clarify the understanding about the local heat transfer and fluid flow circumstances in a FTCHE leading to establishmentof acommonly used friction factor, Nusselt number and area goodness factor capable of being applied for different shapes of tubes in a FTCHE model.

3.1 Streamlines, Temperature Contours

Evolution of numerical modellingconsists in streamlinesand temperatures contours for flow across a four-row tube bank arrangement with various forms of tubes are showed in Figure 5 and Figure 6. Illustrationof streamline indicated that the tube shape influence definitely on the flow pattern and streamlines path.

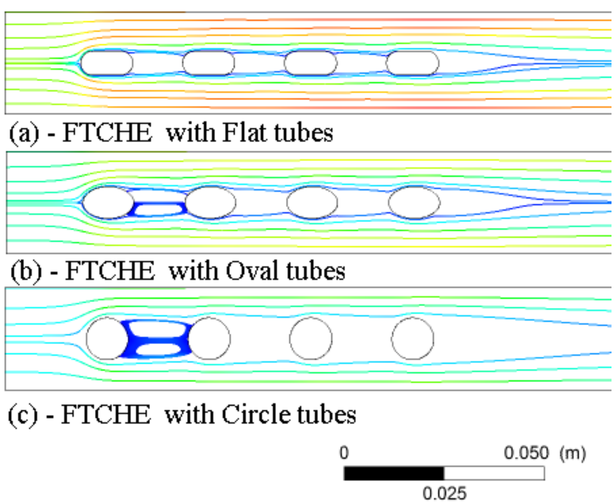


Figure 5 The streamline patterns in the various tube shapes of FTCHEs in Re = 100

The streamline patterns in the flow direction for baseline configuration and enhanced tube shapes configurations are shown in Figure 5.There is a recirculation zone behind of each circular tube for the in-line array of tube bank with comparing to the oval and flat tube shapes at the Reynolds number equal 100. Figure 5 (a-b) show relatively a steady decrease in the wake region behind the tubes as a better wake region controlling and reducing the form drag by introducing high momentum fluid to the wake and delaying the fluid flow separation on the tube surface. The streamlines are qualitatively similar for a different Reynolds number in the range of 100-500.

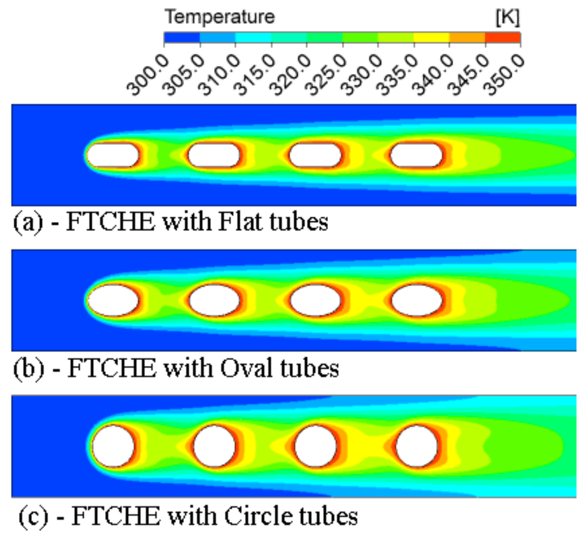


Figure 6 Temperature distribution in the three different forms of the FTCHEs in Re = 100

Figure 6 demonstrates the isotherms contour for the FTCHEs with three various tube shapes. As the air approaches over tube bank, the heattransfer is significantly enhanced. The oval and flat shapes of tubes obviously change the temperature distribution in the FTCHE. In addition, enhancements of local heat transfer across the tube bank achieved by oval and flat tubes. The Temperature Contours qualitatively similar for various Reynolds from 100 to 500.The outcomes of simulation indicated that the average values of temperature modification of air between inlet and outlet flow for flat tube is smaller than that oval and circle tube shapes in the FTCHEs, whereas the heat area goodness factor (J/f factor) improved when using a flat and oval forms of the tubes. Figure 7 and Figure 8 shows the variations in the Nusselt number and the J/f factor of FTCHE against Reynolds number, respectively.

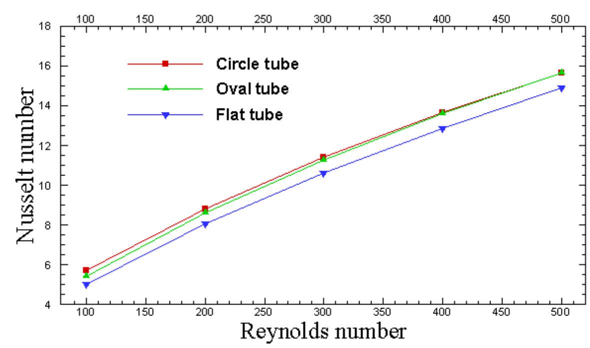


Figure 7 Variations of the Nusselt number vs. Reynolds number

The J/f factor increases for the flat forms and oval tubes compared to the conventional profiles of the tube in the Reynolds number range considered. The calculated friction factor for various Reynolds numbers ranged from 100 to 500 is shown in Figure 9 for a FTCHE with a four-row deep bundle of different tube shapes. It is seen that the friction factor of circular tube is higher than that of flat and oval configuration of tubes for Reynolds number ranged from 100 to 500. In additions, Figure 10 presents the variation pressure drop in the computational domain versus Reynolds number that shown flat tube has the greater pressure drop in the Reynolds number ranged from 100 to 500

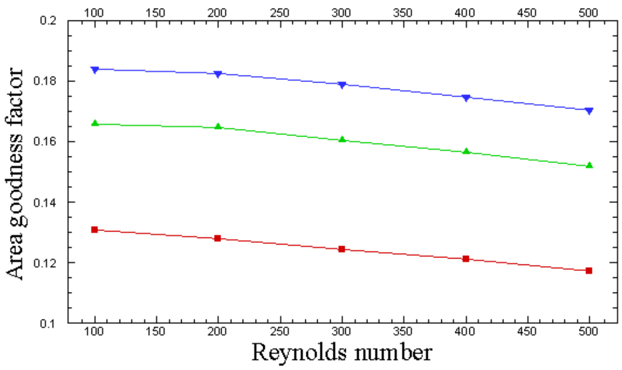


Figure 8 The Area goodness factor (J/f factor) vs. Reynolds number

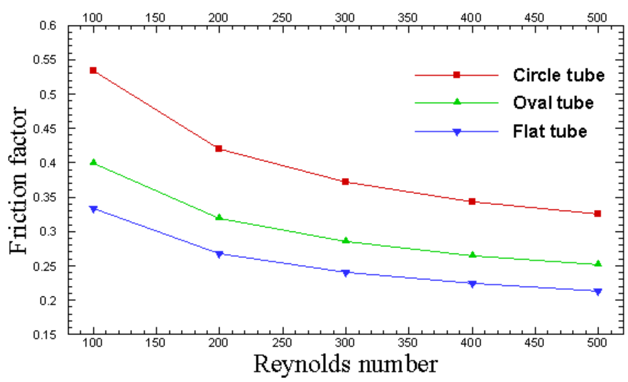


Figure 9 Variations of the friction factor vs. Reynolds number

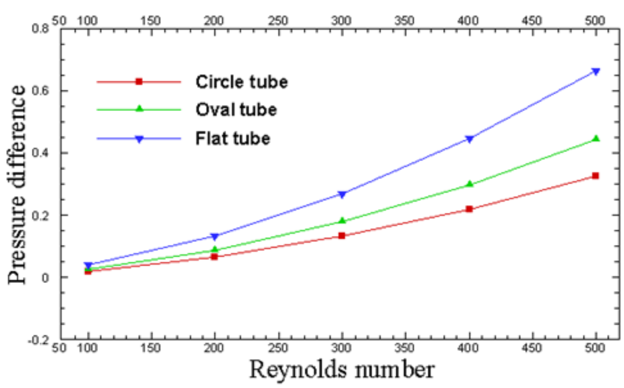


Figure 10 The pressure drop across computational domain vs. Reynolds number

4.0 CONCLUSION

Heat transfer characteristics and thermal performance criteria with various tube geometries placed in FTCHE have been numerically investigated for laminar flow regimes. The following conclusions are preferred:

1. The assessment indicates that the oval and flat form of tubes play a more important role on the thermo-hydraulic performance behavior along the computational model of FTCHEs. The area goodness factors (J/f) of the FTCHEs with oval and flat tubes are improved compared with the circular form of tubes. It is found that the area goodness factor a steady rise in the collection of Reynolds number used in this paper for all cases. In addition, the area goodness factor of the FTCHE with circular tubes has the lowest values in comparison to the other cases.

2. For a FTCHE with the usage of oval and circular tubes causes a steady growth in the Nusselt number and friction factor in proportion to use of the flat form of tubes in the collection of Reynolds number from 100 to 500. However, the flat form of the tube reasonsarise in the pressure difference in comparison to other cases.

References

[1] Wahid, M.A, A.A. Gholami, and H. A. Mohammed. 2013. Numerical Study of Fluid Flow and Heat Transfer Enhancement of Nanofluids Over Tube Bank. *Appl. Mech. Mater*. 388 :149–155.

[2] Gholami, A.A., M.A. Wahid, and H.A. Mohammed.2014. Heat Transfer Enhancement and Pressure Drop for Fin-And-Tube Compact Heat Exchangers with Wavy Rectangular Winglet-Type Vortex Generators, Int. Commun. *Heat Mass Transf*. 54 :132–140.

[3] Gholami, A., M.A. Wahid, and H.A. Mohammed.2015. Numerical Investigation of the Air-Side Performance in the Corrugated Fin-And-Oval Tube Compact Heat Exchangers, in: Recent Adv. Mech. Mech. Eng., Kuala Lumpur. 134–140.

[4] Hasan, A., and K. Siren. 2004 .Performance Investigation of Plain Circular and Oval Tube Evaporatively Cooled Heat Exchangers. *Appl. Therm. Eng*. 24 :777–790.

[5] Horvat, A., M. Leskovar, and B. Mavko, 2006 .Comparison of Heat Transfer Conditions in Tube Bundle Cross-Flow for Different Tube Shapes. *Int. J. Heat Mass Transf*. 49 :1027–1038.

[6] Sayed Ahmed, S.A.E., E.Z. Ibrahiem, O.M. Mesalhy, and M. a. Abdelatief.2014 .Heat Transfer Characteristics of Staggered Wing-Shaped Tubes Bundle at Different Angles of Attack. *Heat Mass Transf*. 50 :1091–1102.

[7] Comini, G., and G. Croce, 2003 .Numerical Simulation of Convective Heat and Mass Transfer in Banks of Tubes. *Int. J. Numer. Methods Eng*. 57 :1755–1773.

[8] Khan, W.A., J.R. Culham, and M.M. Yovanovich.2006 .Convection Heat Transfer from Tube Banks in Crossflow: Analytical Approach. *Int. J. Heat Mass Transf*. 49 :4831–4838.

[9] V.K. Mandhani, R.P. Chhabra, V. Eswaran, 2002 .Forced Convection Heat Transfer In Tube Banks In Cross Flow. *Chem. Eng. Sci*. 57 :379–391.

[10] He, Y.-L. , P. Chu, W.-Q. Tao, Y.-W. Zhang, and T. Xie, 2012 .Analysis of Heat Transfer and Pressure Drop for Fin-and-Tube Heat Exchangers with Rectangular Winglet-Type Vortex Generators. *Appl. Therm. Eng*. 61 :770–783.

[11] Han, H. , Y.L. He, Y.S. Li, Y. Wang, and M. Wu, 2013 .A Numerical Study on Compact Enhanced Fin-and-Tube Heat Exchangers with Oval and Circular Tube Configurations. *Int. J. Heat Mass Transf*. 65 :686–695.

[12] Canonsburg, PA, 2013. ANSYS Workbench User's Guide, ANSYS Inc.