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INVESTIGATION ON HEAT TRANSFER ENHANCEMENT IN A CORRUGATED FIN-AND-TUBE COMPACT HEAT EXCHANGER

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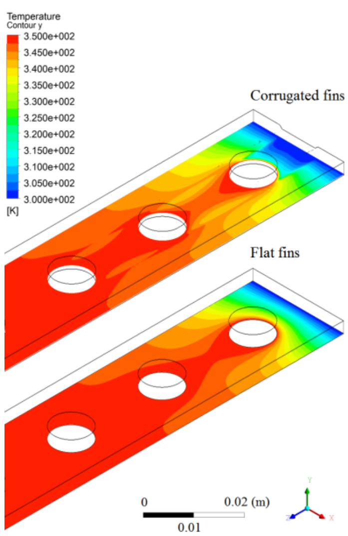
Graphical abstract



Abstract

Heat transfer augmentation and pressure loss penalty in the fin-and-tube compact heat exchangers (FTCHEs) with the corrugated shape as a special form of the fin are numerically investigated to improve heat transfer performance criteria in low Reynolds numbers. The corrugated fin as the newly design of fin pattern is presented in this study. The influence of applying corrugated design adjustments on the thermal and hydraulic characteristics of air flow are analyzed on the in-line tube arrangements. The performance of air-side heat transfer and fluid flow is investigated by numerical simulation for Reynolds number ranging from Re = 400 to 800 based on the tube collar diameter, with the corresponding frontal air velocity ranging from 0.35 to 0.72 m/s. The outcomes of simulation revealed that the corrugated fin could significantly improve the heat transfer augmentation of the FTCHEs with a moderate pressure loss penalty. The computational results indicated that some eddies were developed behind the fluted domain of corrugated finwhich produce some disruptions to fluid flow and enhance heat transfer compared with plain fin. The corrugated form of fins could enhance the thermal mixing of the fluid, delay the boundary layer separation, and reduce the size of the wake and the recirculation region behind tubes compared with the conventional form of the fin at the range of Reynolds number used in this study. In addition, the results showed that the average Nusselt number for the FTCHE with corrugated fin increased by 7.05–10.0% over the baseline case and the corresponding pressure loss decreased by 5.0–6.2%.

Keywords: Fin-and-tube compact heat exchanger, Corrugated fin, Heat transfer enhancement, Performance evaluation, Numerical simulation

1.0 INTRODUCTION

Flow over tube banks with different shapes and arrangements have comprehensive practical applications in different engineering fields, such as cooling, heating, ventilation equipment, aerospace, and automobile etc. due to their high effectiveness and compactness. The heat transfer coefficient on the air-side is typically low due to the thermo-physical properties of air. Thus the air-side thermal resistance is the dominant part of the overall heat transfer process and efforts to improve the performance of these heat exchangers should focus on the air side surfaces [1]. The request to advance energy performance, with reduced volume and manufacturing costs continues to encourage research in gas-side heat transfer enhancement. Thus, many efforts have been constructed to enhance the air-side heat transfer performance and alternatives of fin patterns like wavy, louver and slit fin have been approved [2].

The foregoing literature review shows that there are numerous types of fin surfaces, such as plain fin [3], offset strip fin [4], wavy fin [5], composite fin [6], louvered fin, and different fin surfaces with vortex generators [7, 8]. The main objective behind these researches were to investigate the thermal-hydrodynamic characteristics of systems with augmenting performance criteria. For each type of fin surface, the range of design parameters is considerable. Different fin geometries have been investigated under dry surface conditions, including the slit, herring bone wavy fins with different geometrical parameters, including longitudinal tube pitch, transverse tube pitch, tube diameter, and fin pitch, were used to develop the correlations by employing a Fanning friction factor and j-Colburn factor[9–13].

Čarija *et al*. [14] analyzed a numerical examination of thermal-hydrodynamic characteristics of air-side Fin-and-tube compact heat exchangers (FTCHEs) with conventional flat and louvered shapes of fins in a range of Reynolds number based on fin pitch from 70 to 350. The outcomes showed that the heat transfer characteristics in the FTCHE with louvered fins were considerably better than conventional case of FTCHE. Furthermore, the pressure drop for louvered FTCHE showed a slightly greater values compare to standard FTCHE for all rang of Reynolds number.

Gong *et al*. [12], Lin *et al*.[15,16] presented different unique fin patterns with curved delta-winglet vortex generators, interrupted half annular groove, and punching curve rectangular vortex generators, respectively. A modification in surface of fins purposed as an active technique to develop streamline pattern when fluid flows through the fin spacing in the FTCHEs. The outcomes of these researches shown that in the FTCHEs with interrupted half annular groove vortex generators for the lower Reynolds numbers, the interrupted annular groove fin surface could not efficiently enhance heat transfer under identical pumping power criteria, and the admirable performance of the interrupted annular groove fin can be reached at higher Reynolds numbers. Moreover, for the FTCHEs with curved delta-winglet vortex generators the outcomes indicated that the heat transfers efficiently improved under either identical pumping power or identical mass flow rate constrains.

In general, many techniques are used to enhance the heat transfer in fin-and-tube compact heat exchangers. Among them is the periodic interruption of the growth of the thermal boundary layers close to the heat transfer surfaces. Another technique is the increase of fluid mixing, fluid vortices and turbulence intensity[8].

In this article, a new form of the fin as air-side heat transfer enhancement is presented. Numerical analysis of FTCHE with corrugated forms and a conventional configuration of fins are implemented. To the authors' best knowledge the problem of flow across a tube bank of cylinders in FTCHE has not been solved by the corrugated form of the fins. The performance of heat transfer and fluid flow is investigated by numerical simulation and enhancement mechanism is analyzed.

2.0 PHYSICAL AND MATHEMATICAL MODEL

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 simulation in this paper can be written as equations from Ref[6,8,17].

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):

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

The general Eq. (1)-(3) are used in CFD calculations to calculate the flow field for both thermal and fluid (air) dynamics, solving for heat transfer and pressure drop.

2.2 Computational Model

In this study, a fin-and-tube compact heat exchanger (FTCHE) with corrugated fin profile is used to evaluate heat transfer and pressure drop in air conditioning system. Typical core region of FTCHEs with four in-line tubes arrangement is presented in Figure 1 as physical model. Figure 2 shows the computational domain of FTCHE consist in dimensions of corrugated fins and their situation with respect to the fin configuration.

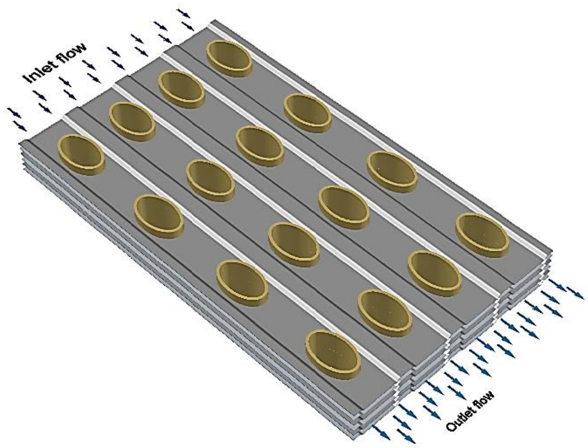
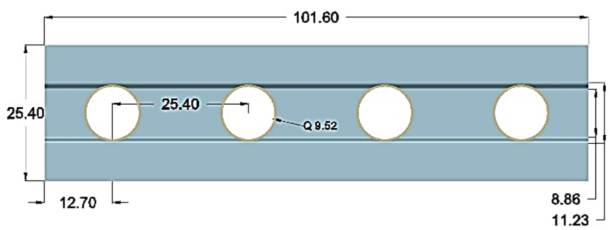
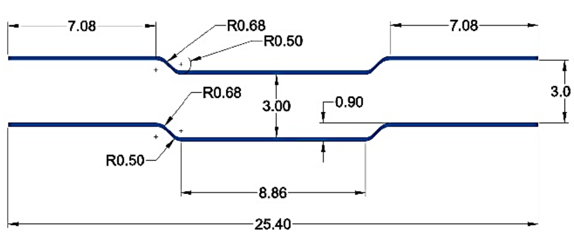


Figure 1 Schematic diagram of core region of a FTCHE.

The geometric characteristics and coordinates of a corrugated FTCHE are shown in Figure 2.



(a) Top view of fin and four tube with in-line arrangement



(b) Front view of corrugated fin patterns

All dimensions are in mm

Figure 2 Geometric characteristics of corrugated fin patterns: a) Top view of model; b) Front view of corrugated fin patterns

2.3 Boundary Conditions

The fluid is assumed to be laminar and in steady state condition. Due to the low air inlet velocity and the small inlet space, the flow in the fin channel of FTCHE is assumed to be laminar and steady state [7]. The thermal contact resistance between the tube and fin collar is ignored, and the tube surface is set as constant temperature. Fin thickness and heat conduction in the fins are taken into account. The temperature distribution for the fins can be determined by solving the conjugate heat transfer problem in the computational domain. An inlet air flow velocity rate of 0.23 to 2.068 m/s is applied to the inlet boundary of the periodic module. The fluid enters with uniform temperature of T_{in} = 300 K and different inlet uniform velocities ($0.35 \leq u_{in} \leq 0.72$ m/s) are applied. At the outlet of the computational model a relative average pressure equaling zero was defined. A constant temperature T_{wall} = 350 K is specified for the wall (tube). At the side boundaries symmetry conditions considered.

A summary of the boundary conditions applied in the computational domain can be described for the three regions as follows[17]:

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

- At the inlet boundary:

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

- At the up and bottom boundaries:

$(\partial u/\partial y) = 0$, $(\partial w/\partial y) = 0$, $v = 0$,
$(\partial T/\partial y) = 0$,
$(\partial P/\partial y) = 0$

- At the side boundaries:

$(\partial u/\partial z) = 0$,
$(\partial w/\partial z) = 0$, $w = 0$,
$(\partial T/\partial y) = 0$,

- In the downstream extended region At the top and bottom boundaries:

$(\partial u/\partial y) = 0$,
$(\partial w/\partial y) = 0$, $v = 0$,
$(\partial T/\partial y) = 0$,

- At the side boundaries:

$(\partial u/\partial z) = (\partial v/\partial z) = 0$, $w = 0$,
$(\partial T/\partial z) = 0$,

- At the outlet area (One-way):

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

(2) In the fin coil region or main and center region of model

- At the up and bottom boundaries:

Velocity condition: $u = v = w = 0$
Temperature condition: periodic conditions

- At the side boundaries:

Fluid region:
$(\partial u/\partial z) = (\partial v/\partial z) = 0$, $w = 0$,
$(\partial T/\partial z) = 0$

Fin surface region:
$u = v = w = 0$,
$(\partial T/\partial z) = 0$

(3) In the tube wall surfaces:
Velocity condition: $u = v = w = 0$
Temperature condition: $T = T_{wall} = 350$ K

3.0 NUMERICAL METHODOLOGY

The geometry for the three-dimensional corrugated FTCHE is complicated and it can be accepted that the velocity and temperature fields are complicated in the computational domain. The Navier-Stokes and energy Eq. (1)-(3) with the boundary condition equations are solved by using a computational fluid dynamics approach in ANSYS Workbench [18]. A segregated, implicit solver option is used to solve the governing equations. The first order upwind discrimination scheme is employed for the terms in energy, momentum, and laminar flow parameters. A standard pressure interpolation scheme and SIMPLE algorithm with pressure velocity coupling are employed. The convergence criterion for the velocities is that the maximum mass residual of the cells divided by the maximum residual of the first 5 iterations is less than 1.0×10^{-5}, and the convergence criterion for the energy is that the maximum temperature residual of the cells divided by the maximum residual of the first 5 iterations is less than 1.0×10^{-8}.

3.1 Parameter Definition

In order to present the simulation results, some characteristic and non-dimensional parameters defined as follows [8]:

$$Re = \frac{\rho U_m D_h}{\mu} \tag{5}$$

$$Q = m_f C_p (T_{out} - T_{in}) \tag{6}$$

$$\Delta T_{lm} = \frac{((T_{wall} - T_{in}) - (T_{wall} - T_{out}))}{Ln \frac{(T_{wall} - T_{in})}{(T_{wall} - T_{out})}} \tag{7}$$

$$h = m_f C_p (T_{out} - T_{in})/(A_t \Delta T_{lm}) \tag{8}$$

$$Nu = \frac{hD_h}{k} \tag{9}$$

$$\Delta P = (P_{in} - P_{out}) \tag{10}$$

$$f = \frac{\Delta P}{0.5\, \rho\, U_m^2} \frac{A_c}{A_t} \tag{11}$$

Where ρ is air density, U_m is the mean velocity at the minimum flow cross-sectional area A_c, A_t is the total heat transfer surface area, D_h is the hydraulic diameter for flow channel, P_{in} is inlet pressure, P_{out} outlet pressure, ΔP is the pressure drop across the computational domain, T_{wall} is the wall temperature and T_{in} is the bulk inlet temperature taken as constant, k is the thermal conductivity,h is the heat transfer coefficient.

3.2 Validation of Model

In order to confirm the independency of solution on the grid, various grid generation are assumed, which contain 746139, 1278420, and 1641344 nodes separately for the FTCHE. The final accepted grid number is 1278420 for the case of study at Reynolds number equal to 500. The difference in averaged Nusselt number for the grid systems is revealed in Table 1. Similar validations are also completed for other suitcases. Furthermore, to validate the reliability of the numerical simulation being used, the numerical model is conducted for a FTCHE with the average convective heat transfer coefficient at uniform surface temperature of tubes, experimental results carried by as presented in Ref. [9].

Table 1 Outcomes of various grid numbers.

Grid number	746139	1278420	1641344
Nu	4.0325	4.9115	4.9143
f	0.0234	0.0273	0.0284

The relationships between average Nusselt number and Reynolds number for an in-lined tube bank are shown in Figure 3. As can be seen from the figure, the numerical outcomes are in fairly good agreement with the existing correlations.

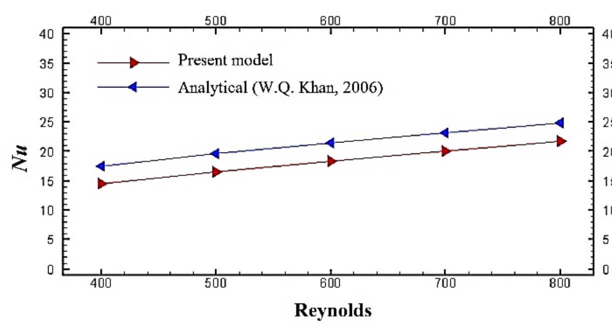


Figure 3 Comparison of Nusselt number from Ref.[19] and numerical simulation

4.0 RESULTS AND DISCUSSION

In order to examine the influence of corrugated fins on the heat transfer characteristics and flow structure for FTCHEs, a comparative investigation for FTCHEs with and without corrugated form of fins is performed. The numerical outcomes for the FTCHEs with Reynolds number based on the inlet velocity ranging from Re = 400 to 800 for in-lined arrangements are shown in Figure 3 to Figure 10. Figure 3 and Figure 4 illustrate the streamline patterns along the flow direction for baseline and enhanced configuration, respectively.

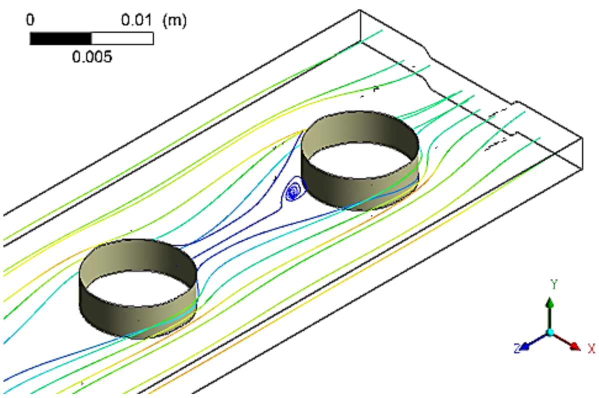


Figure 4 The streamline patterns in the corrugated FTCHE for Re = 400.

There is a recirculation zone behind each tube for the in-line array of tube bank compare to the conventional fin shape at the range of Reynolds number of the present study in Figure 3. The corrugated form of fin in Figure 4 shows relatively a steady decrease in wake region behind the tubes as an improved 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 400-800. Figure 5 and Figure 6 illustrate the temperature contour for the in-lined arrangements in the baseline case and corrugated type of fin, respectively. As the air approaches the heat transfer is significantly enhanced. The corrugated fin shape obviously changes the temperature distribution in the FTCHE and enhances the local heat transfer across the tube bank.

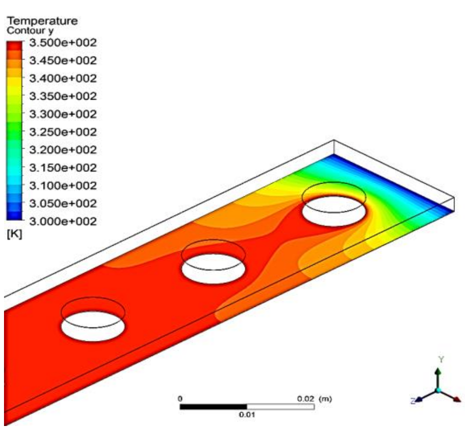


Figure 5 Temperature distribution in the conventional FTCHE for Re = 400

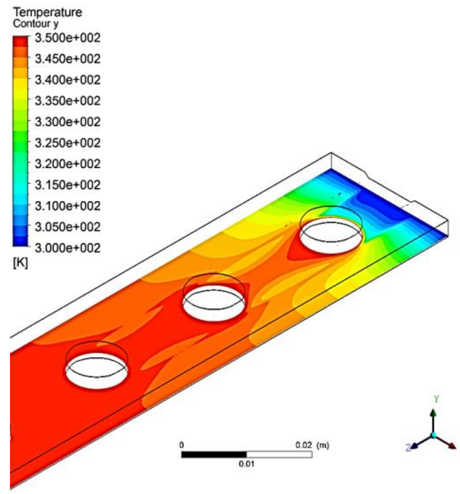


Figure 6 Temperature distribution in the Corrugated FTCHE for Re = 400

The numerical results indicated that the average temperature difference of air between in and out flow for corrugated fin is greater than that for flat continuous fin as the baseline case. Additionally, the heat transfer enhancement improved when using a corrugated form of the fin. Figure 7 and Figure 8 show the Nusselt number and the overall performance criterion (JF factor = Colburn factor / fraction factor$^{1/3}$) [6] increases for the variations in the Nusselt number and JF factor of fin against Reynolds number. In this study, fluid flow characteristics of the corrugated fin compared to the conventional form of the fins in the Reynolds number ranging from 400 to 800.

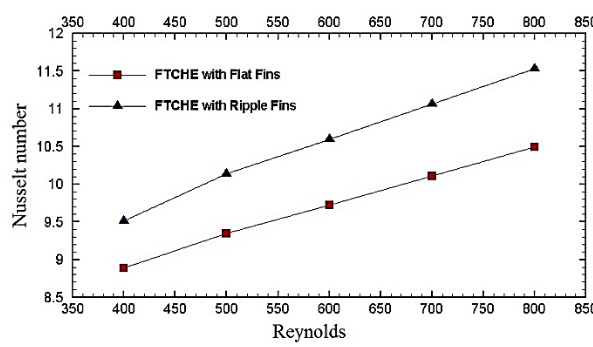


Figure 7 Variations of Nusselt number versus Re number

The calculated pressure difference and friction factor in various Reynolds numbers ranged from 400 to 800 are shown in Figure 9 and Figure 10 for a FTCHE with a four-row deep bundle of cylinders. It is seen that the friction factor of corrugated pattern is higher than that of flat configuration of fin for Reynolds number ranged from 400 to 800. In additions, the variation pressure drop in the computational domain shown that the flat fin has the larger pressure drop in various Reynolds.

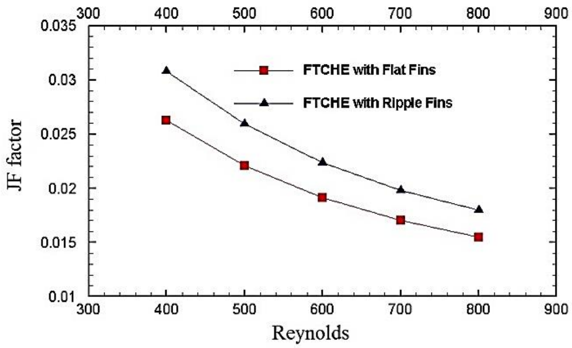


Figure 8 JF factor (overall performance criterion) versus Re number

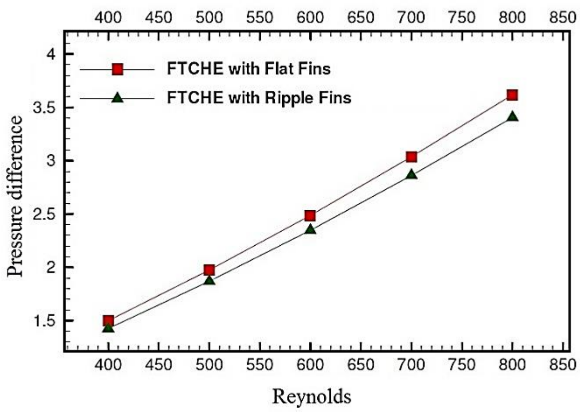


Figure 9 Variations of Pressure drop versus Re number

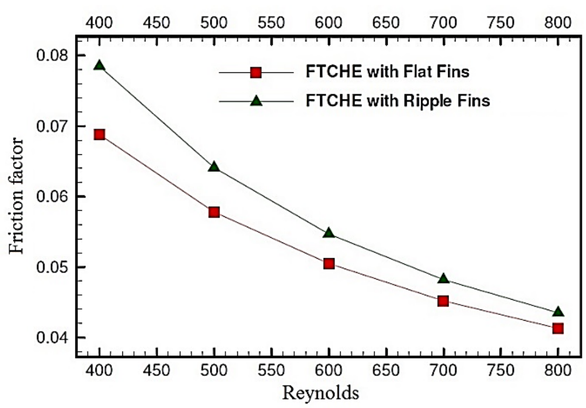


Figure 10 Variations of friction factor versus Re number

4.0 CONCLUSION

The purpose of this paper is to present results of an investigation into the overall variation of heat transfer coefficient around the in-line array of circular tubes with a newly design of the fin shape in low Reynolds number flow and to consider the effect of varying fluid. In this paper, the corrugated fin is proposed, and the performance of heat transfer and fluid flow of corrugated fin is investigated by a three-dimensional numerical simulation. The conclusions are summarized as follows:

1. The corrugated form of fins could enhance the thermal mixing of the fluid, delay the boundary layer separation, and reduce the size of the wake and the recirculation region behind tubes compared with the conventional form of the fin at the range of Reynolds number used in this study.

2. The heat transfer coefficient of the FTCHE was improved compared with the conventional form of fins. The corrugated fins significantly enhance the heat transfer performance of FTCHEs. In addition, the corrugated fin has relatively lower pressure drop penalty in comparison to conventional form of fin.

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