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Thermohydraulic Performance of a Fin and Inclined Flat Tube Heat Exchanger: A Numerical Analysis

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ARTICLE INFO	ABSTRACT
           	Proper determination of inclination angle of a flat tube may increase the overall heat transfer performance without extending heat transfer surface. In this paper, the inclined flat tube heat exchanger with plain fins is numerically investigated. The influence of flat tube inclination angle and Reynolds number on the thermo-hydraulic performance index was evaluated. Tube pitch, fin spacing and flat tube size are fixed. Solving 3D computational domain with the symmetric boundary condition is used to reduce computation time. The results show that when increasing the inclination angle of the flat tube from 0 to 45°, both heat transfer and pressure loss increase because the free area of air flow decreases leading to an increase in air velocity and impingement heat transfer. The variation of inclination angle from 0 to 15°, the increase in heat transfer is stronger than the increase in the pressure loss penalty, so the performance index reaches a maximum of 0.405 at the angle of 15°. Contours of temperature, pressure and velocity at different inclination angles are presented to clarify the thermo-hydraulic characteristics of finned-tube heat exchangers using inclined flat tubes. The current work yields heat transfer enhancement ability by adjusting inclination angle of a heat transfer flat tube.

1. Introduction

Heat exchanger (HE), which takes charge of thermal energy exchange of hot and cold fluids, is a vital part of thermal systems and networks [1]. Depending on the intended use, heat exchangers have different shapes and structures. The simplest one is attributed to a double-tube HE consisting of tubes concentrically attached to create space for fluid to move and transfer heat [2,3]. Another type is the shell-and-tube heat exchanger which is commonly used in the large-capacity field exchanger [4]. The finned-tube heat exchanger is a compact HE which is widely used in household and industrial applications [5-8]. These can be mentioned as microprocessor coolers that use a liquid coolant and dissipate heat to the surrounding environment via the air blowing through the fins, or the outdoor and indoor units of air conditioners [9]. In addition, by combination with measures to enhance heat transfer on the heat exchanger surface, the heat transfer capacity is increased compared to traditional smooth surfaces and fluids [10-12].

Recently, heat exchangers with non-circular tube are used to speed up heat transfer rate and reduce pressure loss. Wang *et al.,* [13] studied elliptical tubes in a compact heat exchanger. Tubes have ellipticity ratios from 0.4 to 1 and inclination angles from 0 to 90° to investigate heat transfer and pressure loss. They confirmed that the ellipticity ratio of 0.6 and the tilt angle of 30° provide the best heat transfer effectiveness. Yogesh *et al.,* [14] concluded that heat transfer was the highest at ellipticity ratio of 0.6 and inclination angle of 20°. Furthermore, this ellipticity ratio can lead to reduce pressure loss up to 53% compared to that of circular tube. Gholami *et al.,* [15] studied the combination of corrugated fin and elliptical tube in a compact heat exchanger. The results show that the combination reduces pressure loss by 19% and increases heat transfer rate up to 20%. Another kind of non-circular tube is the flat tube which proves to be easier to fabricate and install than the elliptical tube. Zaidan *et al.,* [16] explored flat tubes with perforated circular fin. They came to the conclusion that the flat tube and triangular perforated fins provide high thermohydraulic performance. Carpio and Valencia [17] inserted a vortex generator around flat tubes in a compact heat exchanger. The numerical results show that the thermal performance is increased by 52% compared to the finned flat tube without vortex generator. The most recently, Alnakeeb *et al.,* [18] numerically investigated flat tubes with six different aspect ratios from 0.33 to 1 to show their influence on the thermo-hydraulic properties. The results demonstrate that the performance index is increased by 42% compared with the circular tube. Zheng *et al.,* [19] numerically examined flat tube heat exchanger with inclined ribs inside the tube. They reported that the combination of flat tube and ribs induced four swirl flows which result in the improved heat transfer. Gu *et al.,* [20] experimentally investigated condensation heat transfer and pressure drop inside an inclined flat tube. They proved that the inclined flat tube reveals a better condensation heat transfer coefficient than that of a horizontal circular tube due to the thinner condensate film thickness. The above works denoted high heat transfer rate and low pressure drop characteristics of the flat tube. However, there are few studies on fin and flat tube heat exchanger as confirmed by Sadeghianjahromi and Wang [21].

From the above literature review, it can be seen that the inclined elliptical tube and the horizontal flat tube are capable of increasing the heat transfer rate and reducing the airside pressure loss of a compact heat exchanger. However, a study of finned-tube heat exchanger with inclined flat tube was not found. This study is to fill the gap by investigating the influence of flat tube inclination on thermo-hydraulic performance. The tilt angle varies from 0 to 45° to investigate the air temperature, velocity and static pressure fields in flat tube heat exchangers via CFD (Computational fluid dynamics) analysis. From there, the friction factor and the Colburn factor are deduced to find out the appropriate configuration.

2. Numerical Methodology

Figures 1 and 2 show the three-dimensional (3D) calculation domain with dimensions of 76.2×25.4×1.756 mm and four flat tubes. The tube dimension (aspect ratio of 0.4) and tube pitch are fixed as shown in Figure 1. The angle of inclination of the tube varies from 0 to 45° with increments of 15°. The center of rotation of the tube is located at the center of the tube. The fin in this study is a plain fin with fin spacing H = 2 × 1.756 mm = 3.53 mm. Figure 2 describes the boundary conditions for numerical solutions in ANSYS Fluent 19.2. It should be noted that the computational domain size is adopted from the study of Yogesh *et al.,* [14]. However, the symmetry boundary condition has been applied in this study to reduce the number of grids which leads to a reduction in computation time. Fin and tube surfaces are maintained the fixed temperature of T_s = 373 K while the inlet air temperature is T_{in} = 298 K [14,18]. Atmospheric pressure was applied to the outlet of the

computational domain. Velocity inlet u_{in} in range of 5.7 to 7 m/s was assigned to the left boundary face of the Figure 2 corresponding to Reynolds number of 1300, 1450, and 1600 where the number is defined as [14]

$$Re = \frac{\rho u_{in} H}{\mu}$$
(1)

where ρ and μ are respectively the air density and dynamic viscosity.
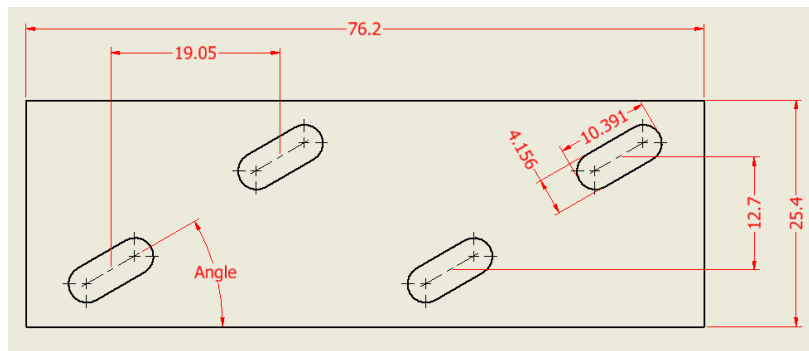

Fig. 1. Geometry of the computational domain (dimensions in mm)
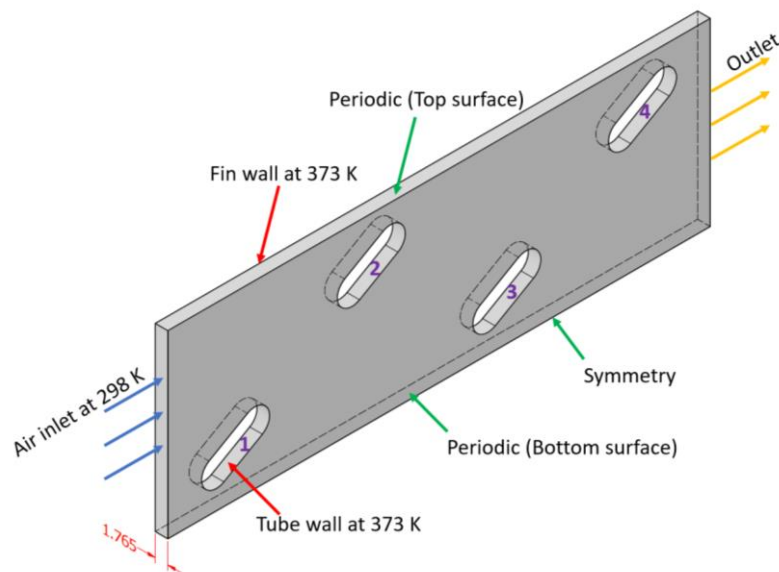

Fig. 2. Description of boundary conditions

Figure 3 shows the grid generation performed in this study. The hexahedral element was selected in this work due to its uniformity and smooth which ensure numerical prediction and computational cost [18]. The numerical computation was performed in ANSYS fluent 19.2 software installed in the workstation desktop computer equipped with Intel Xeon CPU E5-2678 v3, 2.50 GHz, 24 cores, 48 threads (2 processors) and 32 GB RAM.
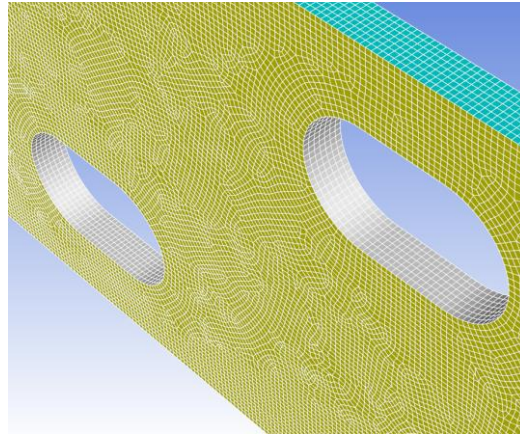
Fig. 3. Mesh generation in hexahedral form

The standard k-ω turbulence model was chosen in this study because it was tested in a previous study [14]. The problem is numerically solved with the following assumptions

   i.   Steady-state conditions and negligible radiation
   ii.   Fin temperature is uniform and equals to the temperature of tube surface [14,18]
   iii.   Airflow is assumed to be incompressible
   iv.   The air properties are temperature independent

Upon the assumptions, the heat and fluid flow in the fin and inclined flat tube heat exchanger were modeled. The used governing equations are presented as below [14]

Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{2}$$

x-momentum equation

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}\right) = -\frac{\partial P}{\partial x} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_T)\left(\frac{\partial u}{\partial x_j} + \frac{\partial u_j}{\partial x}\right)\right] \tag{3}$$

y-momentum equation

$$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z}\right) = -\frac{\partial P}{\partial y} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_T)\left(\frac{\partial v}{\partial x_j} + \frac{\partial u_j}{\partial y}\right)\right] \tag{4}$$

z-momentum equation

$$\rho\left(u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z}\right) = -\frac{\partial P}{\partial z} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_T)\left(\frac{\partial w}{\partial x_j} + \frac{\partial u_j}{\partial z}\right)\right] \tag{5}$$

Energy equation

$$\rho c_p\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}\right) = \frac{\partial}{\partial x_j}\left[\left(\lambda + \frac{\mu_T c_p}{Pr_T}\right)\frac{\partial T}{\partial x_j}\right] \tag{6}$$

Turbulent kinetic energy (*k*) equation

$$\rho\left(u\frac{\partial k}{\partial x} + v\frac{\partial k}{\partial y} + w\frac{\partial k}{\partial z}\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_T}{\sigma_{k3}}\right)\frac{\partial k}{\partial x_j} - P_k - \beta\rho k\omega\right] \tag{7}$$

Turbulent dissipation rate (ω) equation

$$\rho\left(u\frac{\partial \omega}{\partial x} + v\frac{\partial \omega}{\partial y} + w\frac{\partial \omega}{\partial z}\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_T}{\sigma_{\omega3}}\right)\frac{\partial \omega}{\partial x_j}\right] + 2(1 - F_1)\rho\frac{1}{\sigma_{\omega2}\omega}\frac{\partial k}{\partial x_j}\frac{\partial \omega}{\partial x_j} + \alpha_3\frac{\omega}{k}P_k - \beta_3\rho\omega^2 \tag{8}$$

where F_1 is the blending function, *u*, *v* and *w* are velocities in *x*, *y* and *z* Cartesian directions, respectively, μ_T is the turbulent viscosity, and Pr_T is the turbulent Prandtl number.

To reach convergence for highly swirling flows after a tube and reversed flow at the outlet, the PRESTO! pressure interpolation scheme and the under-relaxation factor for momentum equation of 0.6 were employed. The average calculation time of a case is about 10 minutes to obtain all residuals of 10^{-6}. To check grid independence, five different element sizes were performed for the case of Re = 1600 and flat tube angle of 15°. The results shown in Figure 4 revealed that when the number of elements is greater than 150000, the simulation results are almost invariant with the number of elements, so the settings in this element number are applied to all the remaining cases in the study.

Data reduction including Colburn factor *j* and friction factor *f* is carried out as the following. Thermal energy of the air received from fin and tube surfaces is estimated as

$$Q = \rho u_{in}Ac_p(T_{out} - T_{in}) \tag{9}$$

where *A* is flow cross-sectional area ở inlet, and c_p is specific heat of the air. Heat transfer coefficient is calculated from heat transfer equation as [18]

$$h = \frac{Q}{A_s\Delta T_{lm}} \tag{10}$$

where A_s is the surface area of fin and tube, and ΔT_{lm} is the logarithmic mean temperature difference

$$\Delta T_{lm} = \frac{T_{out} - T_{in}}{\ln\frac{T_s - T_{in}}{T_s - T_{out}}} \tag{11}$$

Nusselt number is defined as [14]

$$Nu = \frac{hH}{\lambda} \tag{12}$$

where λ is thermal conductivity of the air. The Colburn factor is calculated by

$$j = \frac{Nu}{RePr^{1/3}} \tag{13}$$

where *Pr* is Prandtl number, $Pr = \mu c_p/\lambda$. The friction factor is derived from air pressure difference from the inlet to the outlet (ΔP) as

$$f = \frac{2\Delta PH}{\rho u_{in}^2 4L} \tag{14}$$

where *L* is flow length. Table 1 presents the thermophysical properties of the air used in this study. Figure 5 depicts a comparison of simulation results in the present study with published data [14]. In this comparison, a horizontal elliptical tube of the same size (aspect ratio of 0.6) was used. The results show that the variation of the Colburn factor with the Reynolds number between the two results is good agreement. Therefore, numerical methodology is applied to all flat tube cases in this study.

Table 1
Thermophysical properties of air

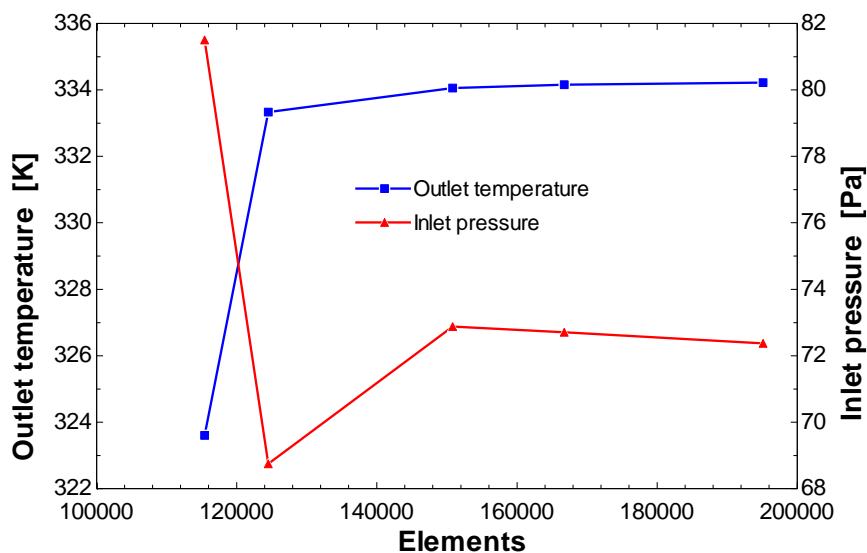| Parameter	Value
Density ρ	1.185 kg/m^3
Heat capacity c_p	1049 J/kg K
Dynamic viscosity μ	1.831×10^{-5} Pa.s
Thermal conductivity λ	0.0261 W/m K

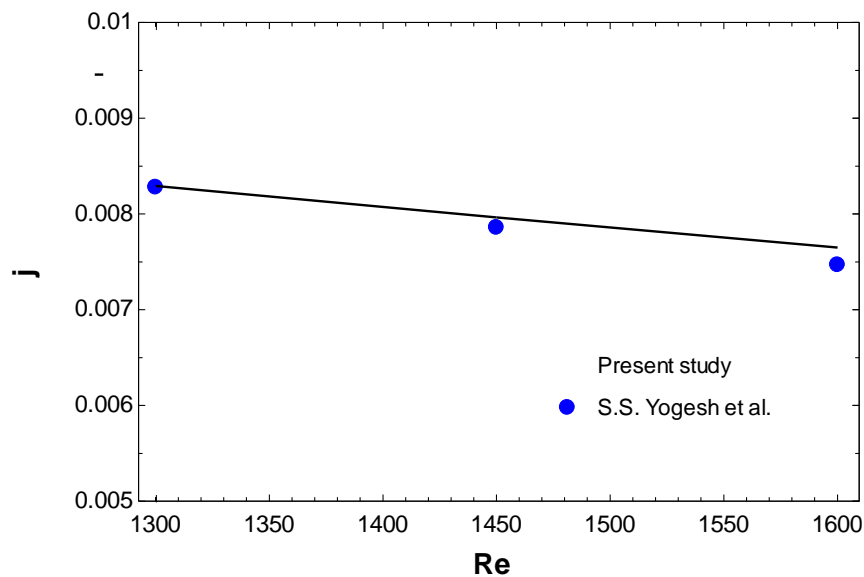**Fig. 4.** Grid independence test in the case of Re = 1600 and angle of 15°

Fig. 5. Validation with the published data [14] for the elliptical tube

3. Results and Discussion

Parametric analysis of the influence of Reynolds number and inclination angle of flat tubes on Colburn factor, friction factor and thermal hydraulic performance is presented in this section. Figure 6 shows the effect of key parameters on the Colburn factor *j*. It is easy to see that as *Re* increases, the factor decreases, as expected. At certain *Re*, when increasing the angle of inclination of the flat tube, the Colburn factor *j* increases. This can be explained by increasing the angle of inclination, the area of cross-section of the fluid flow decreases. Therefore, the air velocity increases leading to an increase in Nusselt number. In addition, the air is in contact with more inclined flat tube surface than that of horizontal flat tube, resulting in enhanced heat transfer due to the impingement heat transfer. At *Re* = 1600, when increasing the tilt angle from 0 to 45°, the factor *j* increases from 0.01 to 0.0115, i.e., 15%.
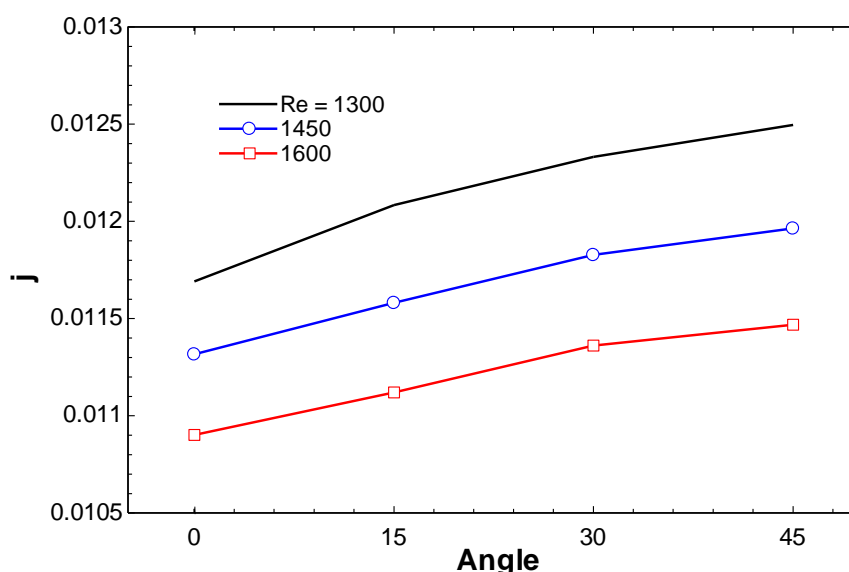

Fig. 6. Effect of flat tube angle and Reynolds number on Colburn j-factor

Figures 7 and 8 present temperature contours at different cross-sections to clarify the heat transfer mechanism. Figure 7 extracts the temperature at the midplane, i.e., the symmetry plane. It can be clearly seen that the angle of inclination 45° achieves the highest heat transfer rate, i.e., the outlet air temperature is much higher than the other cases. In addition, it can be seen that the inclined flat tube acts as a guide, causing the higher air temperature to concentrate at the top of the outlet. Figure 8 shows the outlet temperature distribution between the two fin surfaces. The highest temperature occurs at the maximum angle of inclination. The trend of hot air moves upwards with the angle due to the guide vane effect of the inclined flat tube. From this figure, it is perceived that the areas with small air temperature need improvement measures to ensure uniform temperature distribution.

Figure 9 shows the influence of the independent parameters on the friction factor. It is clear that as the Re number increases, the friction factor decreases. The friction factor increases significantly as the flat tube inclination angle increases. When the tilt angle increases from 0 to 45°, the friction factor increases by about 68%. Figures 10 and 11 show the increase of air velocity and pressure with the angle of inclination. As the tilt angle increases, the jet impingement area increases. It can be clearly seen in Figure 10 that at the maximum angle of inclination, the low air velocity area behind the first two rows of tube is larger than that of the two rear rows. This may conclude that the first rows of tube have a lower convection heat transfer coefficient than the latter rows as shown in the

literature [22]. High velocities in the upper and lower regions of a tube lead to very low pressures in these regions in the case of the greatest inclination.
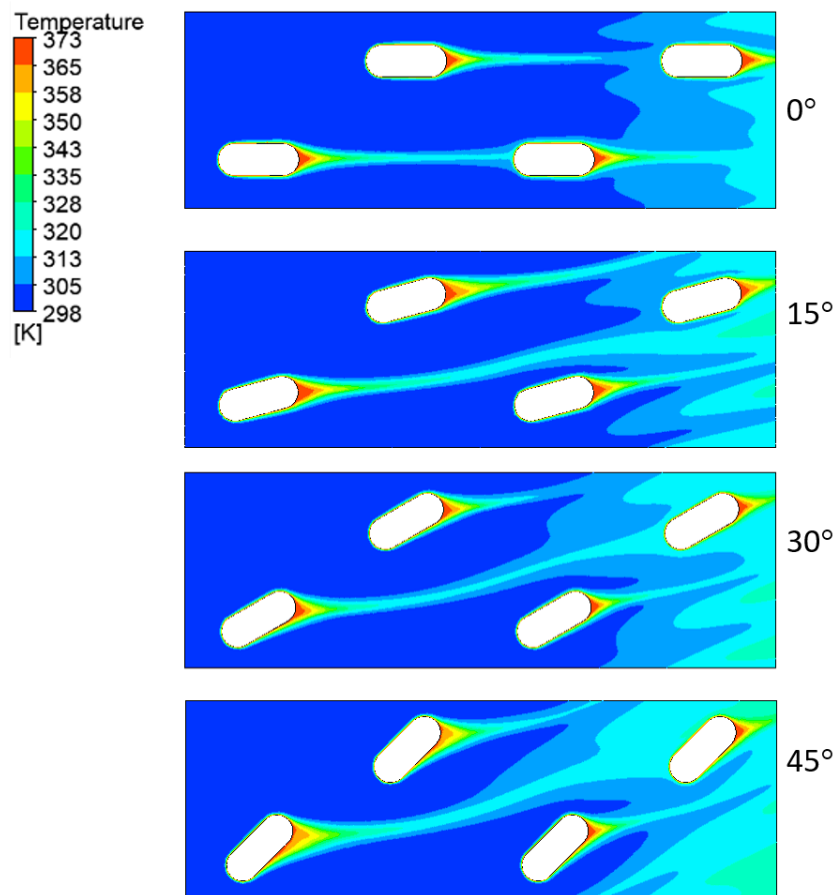

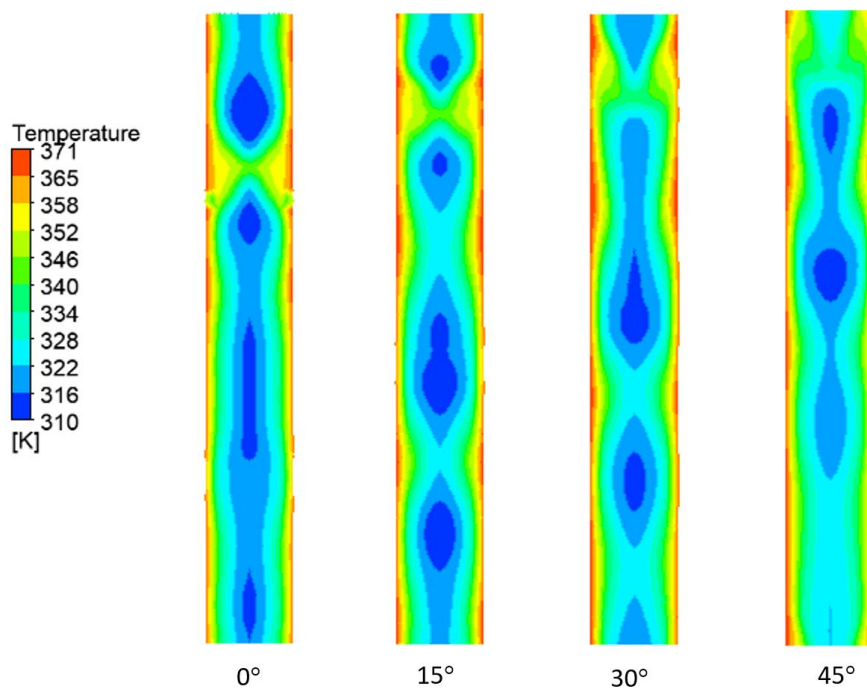**Fig. 7.** Temperature contours at Re = 1300

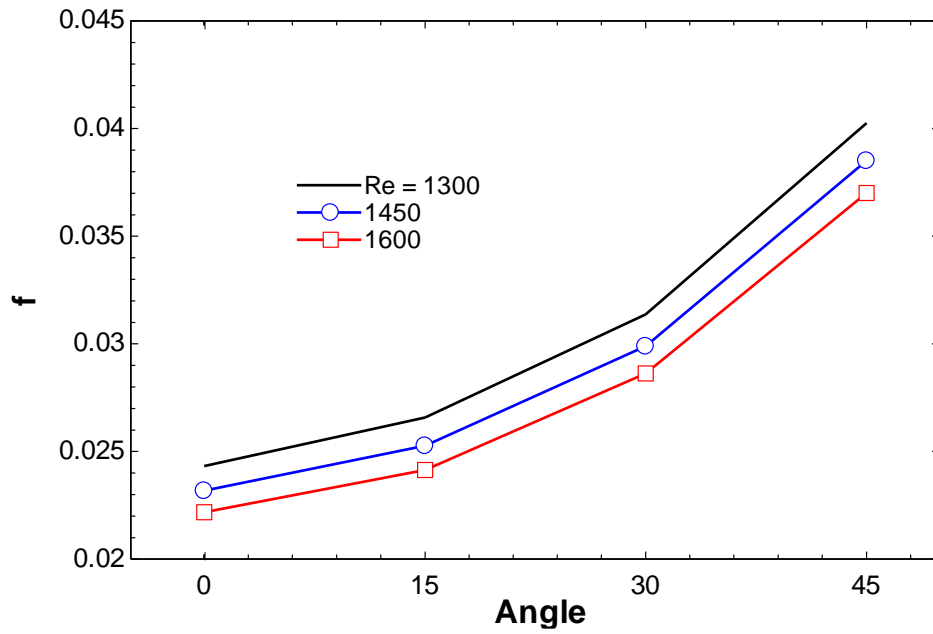**Fig. 8.** Temperature contours on outlet at Re = 1300

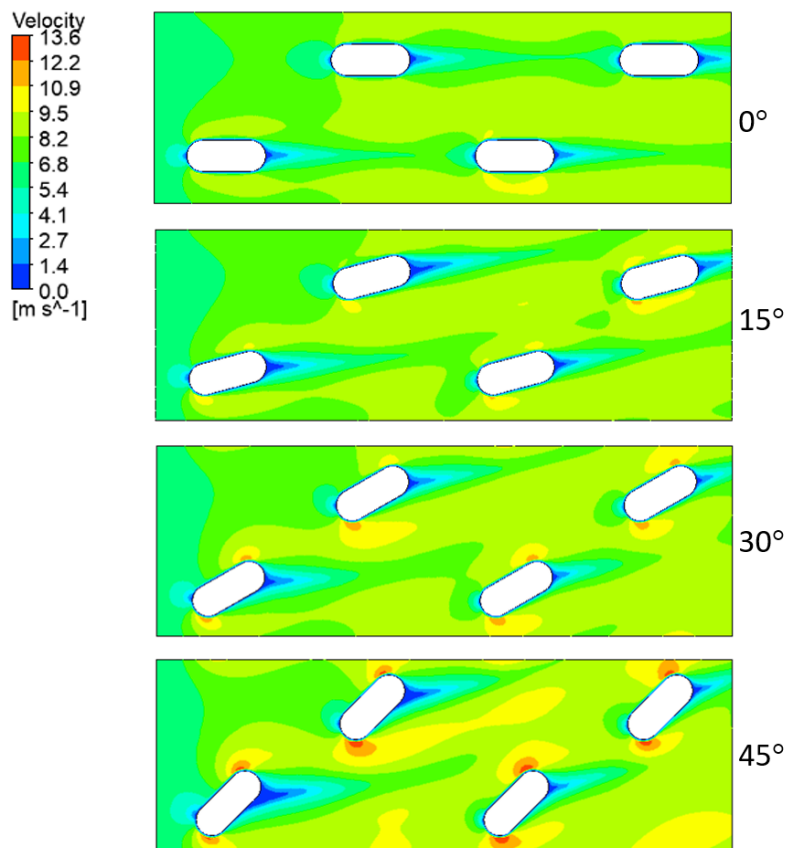**Fig. 9.** Effect of flat tube angle and Reynolds number on friction factor

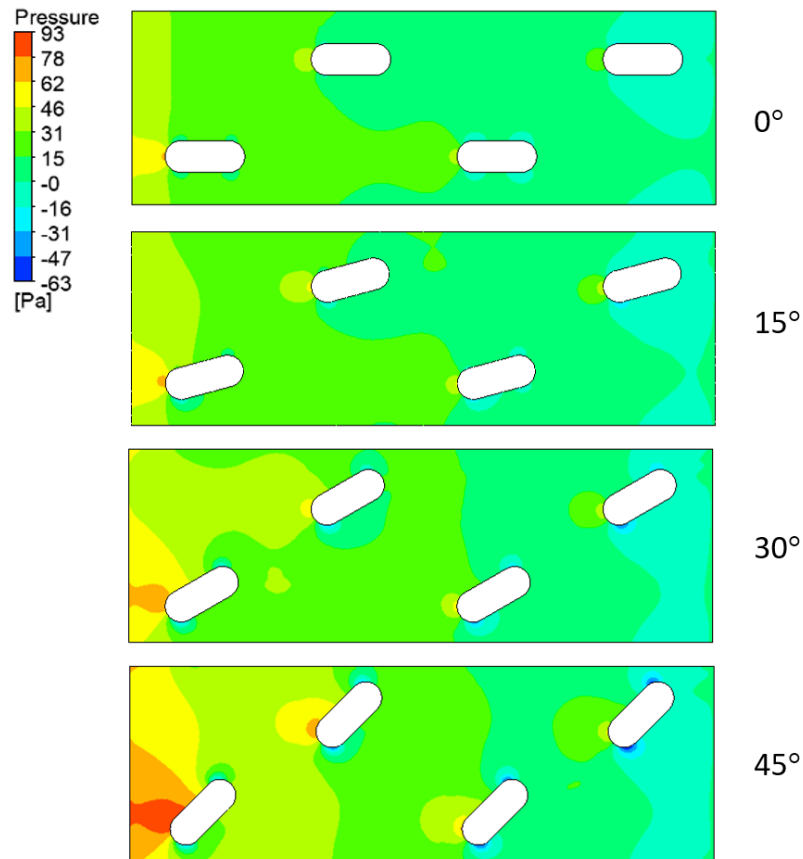**Fig. 10.** Velocity magnitude contours at Re = 1300

Fig. 11. Static pressure contours at Re = 1300

From the above analysis, it can be observed that when increasing the tilt angle, both heat transfer and pressure loss increase. To choose the pertinent angle of inclination, it is necessary to evaluate the hydro-thermal performance index. This index is defined as the ratio between the Colburn factor j and the friction factor f: $j/f^{1/3}$ [23]. Figure 12 shows the performance index with independent parameters.
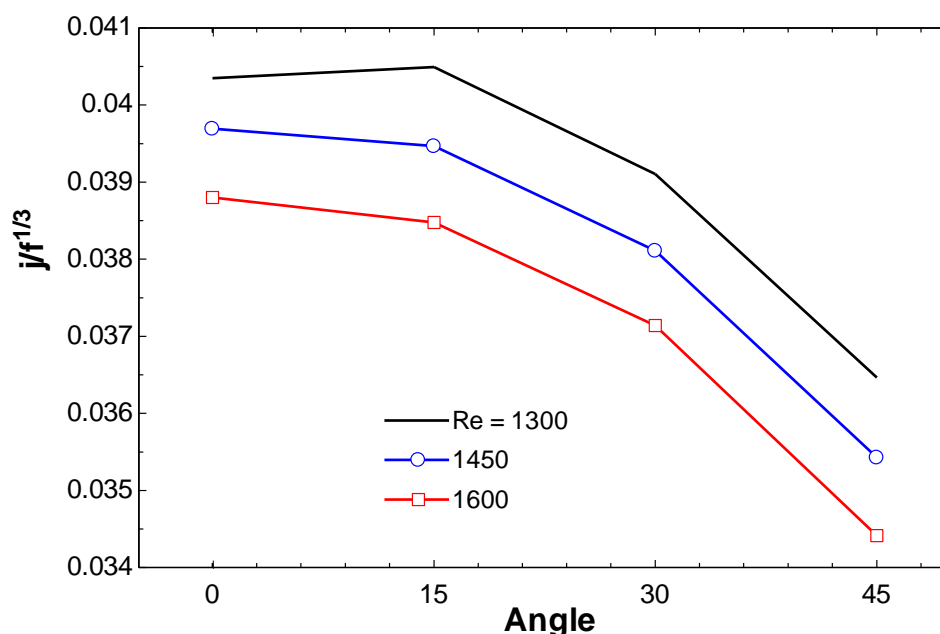

Fig. 12. Effect of flat tube angle and Reynolds number on performance index

The largest index can be seen at the smallest Reynolds number and inclination of 15°. At small Re and small tilt angle, the increase in heat transfer is slightly higher than the increase in pressure loss, thus the index is highest. As the Re number and inclination angle are increased, the pressure loss increases dramatically. Therefore, the index decreases. The maximum value of the index is around 0.0405.

4. Conclusions

The 3D numerical investigation using the standard k-ω turbulence model for compact heat exchanger having inclined flat tubes was carried out in this study. The thermal and hydraulic parameters are evaluated when changing the Reynolds number from 1300 to 1600 and the inclination angle of the tubes from 0 to 45°. The main results from the study are drawn as follows

i. The Colburn factor increases by 15% and the friction factor increases by 68% when the angle of inclination increases from 0 to 45°.
ii. The inclined flat tube acts as the guide vane, causing the hot air to concentrate at the top of the outlet.
iii. The maximum thermohydraulic performance index of 0.0405 is achieved at Re = 1300 and the inclination angle of the flat tube of 15°.
iv. The areas of poor heat transfer shown in the inclined flat tube configurations are proposed for further study to achieve uniform temperature distribution at the outlet.

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