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Flow and Temperature Field in Fin Tube Heat Exchanger through the Addition of Convex Strip Vortex Generators

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Abstract **- Enhancement of heat transfer in fin and tube heat exchangers using vortex generators has been carried out by many researchers because of the effectiveness of this method. Therefore, the present work is focused on observing the effect of varying the number of convex-strips vortex generators on heat transfer improvement and their impact on pressure losses in heat exchangers. Numerical simulations by varying the number of convex strips of 4, 8, 12, 16, 20, 24, and 28 installed around the heated tubes were carried out. The k-ε turbulence model is determined by the Reynolds number range from 3,438 – 15,926. The results of this study conclude that fins with 24 convex-strips show the most uniform temperature distribution than those of the other convex-strips.**

Keywords: Heat transfer enhancement, Convex-strips Vortex Generators, Fin and Tube heat exchanger.

I. INTODUCTION

Energy efficiency in fin and tube heat exchangers can be achieved by increasing the rate of heat transfer through the addition of a geometry in the form of protrusions on the surface called vortex generators (VGs) [1-2]. VGs produce a longitudinal vortex (LV) which is able to enhance the mixing of hot fluids with cold fluids which has an impact on increasing the rate of heat transfer [3]. Therefore, improvements to increase heat transfer using various forms of VG have been of concern to researchers [4].

Recently, the impact of vortex generators on heat transfer performance has been extensively investigated. Deng et al. [5] optimizes rectangular and delta winglet pairs of VGs used in cooling cylindrical film holes. They studied it numerically by using k-ε turbulence in their modeling. From the results of their study it was found that the model using rectangular winglet VGs showed optimal results compared to the others. This model produces heat flux reduction 0.08-0.29 higher than the model that does not use VGs. This study also showed that the heat transfer coefficient of the cooling film with VG was 11.45% higher than that of the cooling film without VG.

Gonül et al. [6] also investigated the VGs by varying the angle of attack, the layout of the VGs, the distance between the VGs, the height and length of the geometry of the VGs, and variations of the Reynolds number. Their work found many differences between Nu/Nu0 and f/f0. They observe that the difference in the height of the VG is the most influential factor than the geometry variation factor. Many researchers studied the heat transfer rate increase by varying the VG geometry. Zheng et al. [7] simulated a trapezoidal cross-section longitudinal vortex generator with variations in the length of the front longitudinal vortex generator (LVG), the length of the back of the LVG, and the height of the LVG.

Geometry variation is one of the methods to increase heat transfer efficiency. Saiful et al. [8] conducted a numerical study to increase the rate of heat transfer by optimizing the number of convex strip vortex generator geometries, namely 4, 8, 12, 16, 20, 24, and 28 convex strips. The k–ε turbulence model with a variation of the Reynolds number 3.438 to 15.926 was determined in the numerical simulation. From their research it was found that VGs with a total of 24 convex strips increased the best heat transfer rate compared to the number of other convex strips. In another study, Syaiful et al [9] used VG variations in the form of delta winglet pairs and concave delta winglet pairs. This study obtained results of an increase in heat transfer of up to 53.58% in the installation of 3 pairs of concave delta winglet pairs, compared to VG which did not use winglets. By considering the optimum number of convex-strips capable of producing the highest heat transfer, the current study is therefore focused on obtaining the best number of convex-strips portions in obtaining the highest increase in heat transfer.

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II. RESEARCH METHOD

This research was carried out by making a three-dimensional model which was then simulated with certain boundary conditions.

2.1 Model Description

The present study was carried out by simulating three-dimensional fluid flow at fin and tube in accordance with experiments conducted by Li et al. [10]. The tube material is copper which is staggered with the number of rows and columns 12 and 6 respectively. The dimensions in the case of the four convex-strips refer to previous studies. Whereas in the current work, the number of convex-strips is varied from 4, 8, 12, 16, 20, 24, and 28 to determine the optimum number of convex-strips for the highest heat transfer improvement. Figures 1 and 2 are the geometry of the sides and top of the vortex generator with 24 convex-strips around the tube. Table 1 shows the geometry of the fin and tube.

Figure 1: 24 convex-strips side view

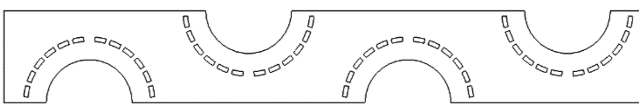

Figure 2: 24 convex-strip top view

Table 1: Fin and Tube Geometry

Parameter	Dimensions (mm)
Fin pitch Fp (mm)	0.15
Fin tickness δ (mm)	19.6
Tube diameter Dc (mm)	42
Tube pitch longitudinal Pl (mm)	36.4
Tube pitch transverse Pt (mm)	536.5
Length (mm)	42
Width (mm)	0.15

2.2 Mathematical Models

In the current study, fluids are assumed to be incompressible with constant physical properties and steady state. The velocity of the fluid in the rectangular channel is varied over the Reynolds number range of 3.438 to 15.926. In this numerical simulation, conjugate heat transfer is calculated to determine the temperature distribution on the fin surface.

The governing equations applied in the current modeling are

Continuity equation

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

Momentum equation

$$\frac{\partial}{\partial x_i}(\rho u_i u_k) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_k}{\partial x_i}\right) - \frac{\partial p}{\partial x_k} \tag{2}$$

Energy equation

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\Gamma \frac{\partial T}{\partial x_i}\right) \tag{3}$$

2.3 Parameter Definition

The parameters used for this study are as follows

Reynolds Number

$$Re = \frac{\rho\, u_m D_h}{\mu} \tag{4}$$

Nusselt Number

$$Nu = \frac{h\, D_h}{\lambda} \tag{5}$$

2.4 Independent Grid Test

The independent grid test in this study was conducted to determine the optimum number of grids where the numerical simulation results are not affected by the number of grids. The simulation was carried out with 5 variations of the number of grids in the range 825,000 to 1,250,000 with a Reynolds number of 15,926. This simulation is carried out by calculating the Nusselt number for various numbers of grids. From the results of the independent grid test, it was found that 1,200,000 grids were selected as independent grids in this study.

III. RESULTS AND DISUCUSSION

Numerical studies of heat transfer intensification in fin and tube heat exchangers with various number of convex-strips portions around the tubes have been carried out. Some of the observed phenomena and some of the influential parameters are discussed in detail as follows,

3.1 Validation

Validation on the current work was carried out by comparing the calculation results of the current study with the experimental results of Li et al. [10]. The Nusselt number and pressure drop simulation results are compared with those from their experimental results for the case of four convex-strips at

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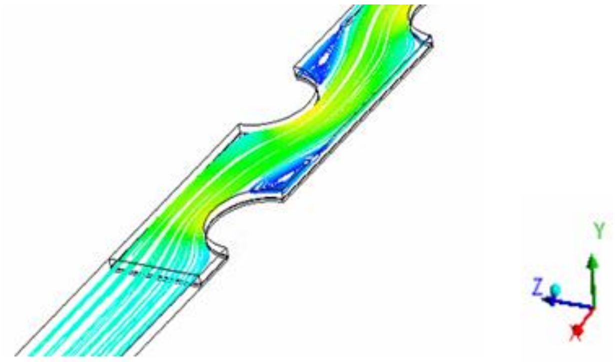
various Reynolds number 3.438 to 15.926. However, the simulation results yield lower than those of Li et al. The simulation results show that the Reynolds number 13,412 and 15,926 the deviation for the Nusselt number and pressure drop was observed to be less than 13% and 18%, respectively.

3.2 The effect of convex-strips on the flow structure

In this study, the area observed for plain fin cases, 24 and 28 convex strips. Figure 3 shows that the flow structure in each convex strip model is observed to be different for cases 20, 24 and 28 convex strips and plain fin at Reynolds number 15.926. To analyze the differences in the streamlined structure in the addition of convex strips, the flow structure is observed in the streamwise plane at Y = 0.88 mm.



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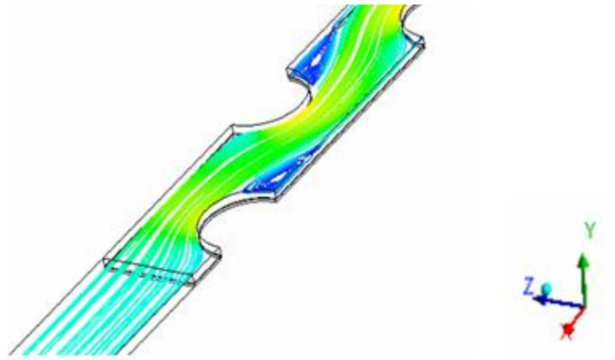

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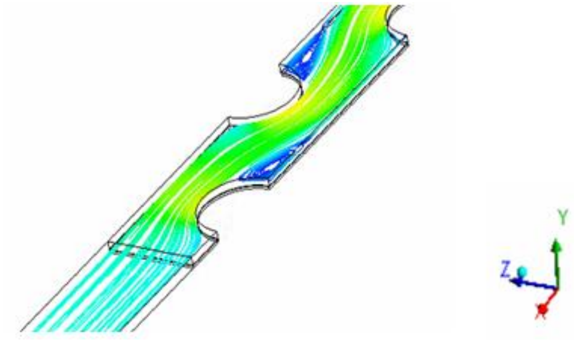

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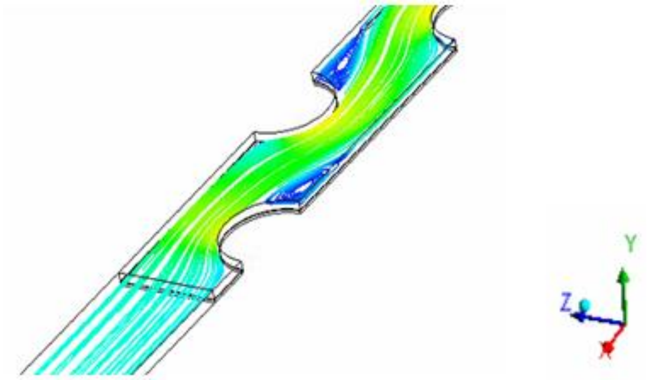

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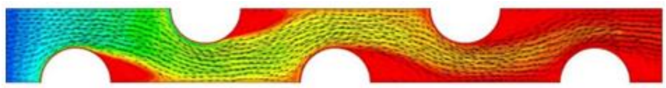
Figure 3: Streamlined velocity of flow in the channel in Re 15.926 for: (a) plain fin; (b) 20 convex strips; (c) 24 convex strips; (d) 28 convex strips

3.3 Effect of convex-strips on temperature distribution

The vortex intensity generated by the convex strips affects the temperature distribution of the fins along the stream. The temperature distribution for each case is observed in the direction of flow plane (XZ plane) at Y = 0.88 mm, as shown in Figure 4. The highest fluid temperature for the plain case is found in the wake area due to recirculation flow which inhibits the rate of heat transfer from the surface of the tubes to the main flow. As for the case with the addition of convex strips, the wake area shrinks due to the induction of the cold fluid from the main stream by the hot fluid in the wake area which results in an increase in the heat transfer rate and a more uniformly distributed temperature. From Figure 4, it is found that the temperature distribution in cases of 24 convex strips obtains more uniform results than those of cases 20 and 28 of convex strips due to the presence of LV.

Temperature distribution for each case. The counter-rotating vortices in the case of the 24 convex strips are larger than those of the others so that the high temperature fluid around the tube wall mixes with the low temperature fluid in the main stream. This results in a more uniform temperature distribution in the case of the 24 convex strips than in the other cases. Whereas in the plain fin case, the temperature is not distributed evenly because LV is not produced which results in slow fluid mixing.



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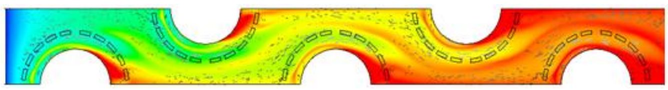

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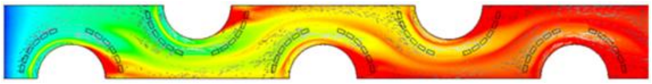

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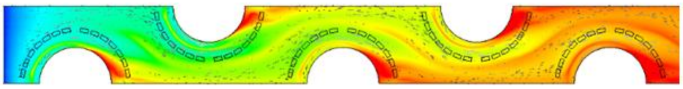

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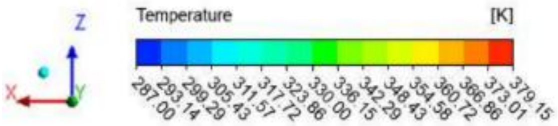

Figure 4:The temperature distribution in the Y plane = 0.88 mm at Re 15.926 for the case of (a) plain fin; (b) 20 convex strips; (c) 24 convex strips; (d) 28 convex strips

IV. CONCLUSION

In this study, numerical simulations have been carried out to determine the optimal number of portions of convex strips in increasing the heat transfer rate in a fin and tube-type heat exchanger. Some of the conclusions drawn from this study are as follows:

1) The highest velocity of the fluid through the 24 convex-strips was observed.
2) A more uniform temperature distribution was observed in the installation of 24 convex-strips.

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