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Research Article

CFD Simulation of a Conjugate heat transfer problem in a rectangular cavity

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Abstract

A computational investigation for the conjugate heat transfer of pin fins inside a rectangular cavity has been performed in the present paper. The CFD analysis for the thermo-hydraulic characteristics of triangular and circular pin fins have been done. It was observed that a higher inlet velocity leads to larger convective heat transfer from the fins to the bulk fluid resulting in a larger increase in fluid temperature for both types of fins. At the same time pressure drop also increases. For the Reynolds number ranging from 200 to 1100, the global Nusselt number varies from 4.2 to 14.5 for circular pin fin heat exchanger whereas this variation is 3.0 to 7.5 for triangular pin fin heat exchanger. In contrast the pressure drop for circular fins is less than the triangular fins at each Reynolds number.

Keywords: *Conjugate heat transfer, Pin-fins, CFD, Nusselt Number, Pressure drop*

1. Introduction

Conjugate heat transfer involves conduction, convection and radiation heat transfer. In industrial applications like electronics cooling, turbulent flow and heat transfer past solid objects (fins) involves conduction and convection. Radiation may be neglected as the temperature is maintained at 70-80^0 C. The fins (circular and triangular shape in this case) are arranged in a rectangular cavity.

W.M.Kays (1955) performed probably the most extensive study of pin fins as elements for heat transfer enhancement. Hwang et al. (1999) simulated the trapezoidal pin-fin-cooling cavity in a typical turbine blade. They measured the log-mean heat transfer and overall pressure drop in a pin-fin trapezoidal duct. Results revealed that the trapezoidal pin-fin duct with lateral outlet flow only has higher log-mean Nusselt numbers than the straight counterpart. Brigham et al. (1984) investigated the effects of number of rows of pin in the stream wise direction and pin length (height) on array-averaged heat transfer. The results showed that the number of pin row has a slight effect on the array-averaged heat transfer. Metzger et al. (1982) investigated the heat transfer characteristics in a rectangular channel with staggered fin arrays. Sahiti et al. (2005) demonstrated that pin fins are the devices that offer the most effective way of heat transfer enhancement within a given volume of the heat exchanger.

A least-material optimization procedure was demonstrated by Iyengar et al. (1998) for vertical cylindrical pin fin, and longitudinal plate and triangular fin arrays in natural convective heat transfer. Experimental investigation for the fluid dynamics of the pin fin arrays in order to clarify the physics of heat transfer enhancement was performed by Ames and Dvorak (2005). They observed that in early rows where turbulence is low, the strength of shedding increases dramatically with Reynolds number. Goldstein et al. (2006) have provided a review of heat transfer enhancement, including numerical and experimental studies. Short et al. (2004) showed that at low to moderate Reynolds number, cast pin fin cold walls provide the best performance and also involve a low cost for electronic applications. Experimental and numerical investigation of air side heat transfer and pressure drop characteristics of a slotted fin surface was performed by Tao et al. (2005). They have showed that slotted fin surfaces provide excellent heat transfer performance compared to the plain plate fin surfaces. A very few experimental and numerical papers on triangular fins have been reported in the literature. A conjugate heat transfer problem for a triangular fin was studied by Hsu et al. (1998). Results indicated that elastic effect in the flow can increase the local heat transfer coefficient and thus enhance the heat transfer capability of the fin. Further a better heat transfer is obtained with a larger Pr. An experimental investigation of heat transfer for triangular fin and pin fins at constant heat flux was performed by Al-Jamal and Khashashneh (1998).The results obtained from the experiment indicated that Nu for pin fin array is higher than that for triangular array at constant heat flux. Conduction in an array of triangular fins with an attached wall was modeled by Abratet and Newnham (1995) using

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the finite element method. The effects of the wall thickness and fin spacing were examined for various Biot numbers. In the present paper a computational study of three-dimensional heat transfer in a rectangular duct with triangular pin fins is investigated and a comparison was made with the circular fins of the same exposed surface area for heat transfer.

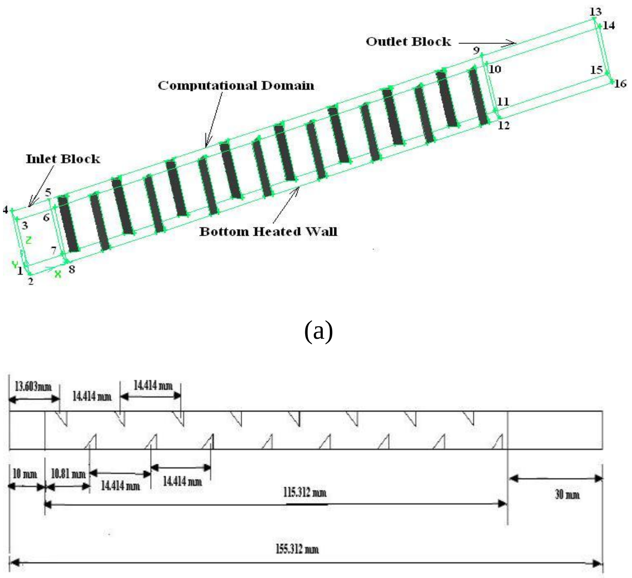
2. Computational Domain

The computational configuration of the heat exchanger is shown schematically in Fig.1. Triangular fins are brazed/ welded in a staggered manner to the bottom plate. The bottom plate is heated due to the heat generated inside the electronic appliances. The objective is to transfer heat from the bottom plate to the atmosphere. The fin material is aluminum of density ρ = 2700 kg/m^3 and thermal conductivity k = 202 W/m.K. The hydraulic diameter in this case is 2.0 mm which is calculated as follows:

$D_h = 4 * A_c * L / A_t$

Here A_c denotes the minimum cross sectional area of the duct, L the length of the fins and A_t the total heat transfer area of the compact heat exchanger.

The bottom heated plate is maintained at the constant temperature. As can be seen from the fin arrangement, the flow pattern is periodically repeating in nature. The height of the fins are 23.0 mm and the side of the triangular cross section is 2.407 mm. Cold air is blown from the inlet which takes heat away from the fins and hot air is rejected to the atmosphere. Inlet block of length $5D_h$ is attached to make the flow fully developed before entering the computational domain and outlet block of length 15 D_h is meant to reduce the effect of backflow.



(a)

(b)

Fig.1 (a) 3D computational domain for triangular fins (b) Top view

3. Governing Equations

Due to the complexity of the flow over a bank of tubes or pin fins (Gerrard, 1966) some assumptions are made to simplify the analysis:

- Flow is turbulent and 3D in mean.
- Flow is steady in mean and incompressible.
- Radiation heat transfer and body forces are negligible.

Considering the geometry and physical problem (Fig.1), the RNG k-ε turbulent model is employed to simulate the turbulent heat transfer and fluid flow characteristics along with the governing continuity, momentum and energy equations. The physical properties of working fluid (air) considered are : ρ = 1.225 kg.m^{-3}, μ = 1.789x10^{-5} kg.(ms)$^{-1}$ and C_p = 1006.43 Jkg^{-1} K^{-1}.

Conservation of mass equation: Div (u) = 0

Conservation of momentum equations:

X-equation: $$\rho div(u_x u) = -\frac{\partial p}{\partial x} + div\left[(\mu+\mu_t) grad(u_x)\right]$$

Y-equation: $$\rho div(u_y u) = -\frac{\partial p}{\partial y} + div\left[(\mu+\mu_t) grad(u_y)\right]$$

Z–equation: $$\rho div(u_z u) = -\frac{\partial p}{\partial z} + div\left[(\mu+\mu_t) grad(u_z)\right]$$

Conservation of Energy equation:

$$\rho c_p div(Tu) = div\left[(k+k_t) grad(T)\right]$$

Where, k denotes the thermal conductivity of the working fluid.

3.1 RNG k-ε turbulence model:

The transport equations for turbulent kinetic energy (k) and turbulent dissipation rate (ε) by Yakhot et al. (1992) are:

$$\rho div(Ku) = div\left[(\mu + \frac{\mu_t}{\sigma_k}) grad(K)\right] + G - \rho\varepsilon$$

$$\rho div(\varepsilon u) = div\left[(\mu + \frac{\mu_t}{\sigma_\varepsilon})\right] + C_1 \frac{\varepsilon}{K} G + C_3 \frac{G^2}{\rho K} - C_2 \rho \frac{\varepsilon^2}{K}$$

$$G = \mu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\frac{\partial u_i}{\partial x_j}, \quad \mu_t = \rho c_\mu \frac{K^2}{\varepsilon}$$

Here K denotes the turbulent kinetic energy and ε the turbulent dissipation rate. The model constants are:

$$C_\mu = 0.0845, C_1 = 1.42, C_2 = 1.68, C_3 = 0.25, \sigma_K = \sigma_\varepsilon = 0.7178$$

3.2 Boundary Conditions

In the present work (Fig.1) the planes 1-2-8-7, 11-12-16-15, 3-4-13-14 are assumed to be adiabatic. Symmetry boundary conditions were applied to the sections 3-14-16-

2 and 4-13-15-1. The no slip and constant temperature boundary condition is applied to the bottom heated plate: u

= v = w = 0 and T_w = 343 K.

Inlet boundary condition:

$u = u_{in}$, v = 0, w = 0, T_{in} = 293K.

Turbulence intensity was set to be 5% for the inlet. The turbulent intensity level, I, is defined the ratio of the root mean square of the velocity fluctuation, u', to the mean flow velocity, u_{mean}, as I = (u'/u_{mean}) *100.

Outlet boundary condition:

P_{out} = 1.01325 bar

4. Results and Discussion

For the validation purpose the result for circular fins is compared with the available experimental data by W.M.Kays (1955). The RNG K-ε model with standard wall function better predicts the numerical result. So this model was used for all subsequent analysis.

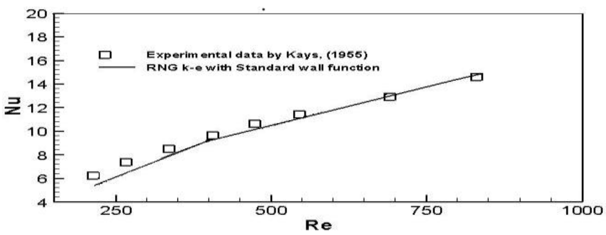


Fig.2 Global Nusselt number vs. Reynolds number

For both the fin geometries, global Nusselt number (Nu) and pressure drop are calculated and plotted as a function of Reynolds number (Figs.3 and 4). The Reynolds number ranges from 200 to 1100. In the given range of Re, Nu varies from 3 to 7.5 for triangular fins heat exchanger whereas Nu varies between 4.2 to 14.5 for circular pin fins heat exchanger. The difference of Nu between circular and triangular fins increases with an increase in the inlet air velocity. At Re = 1100, the Nu for circular fin is twice that for triangular fins.

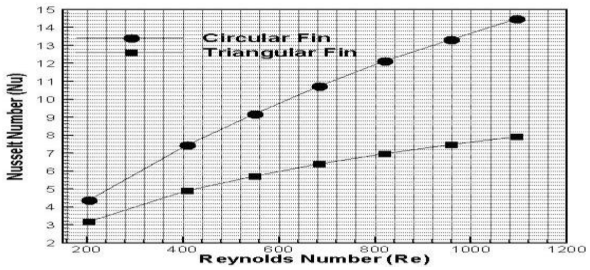


Fig. 3 Comparison of Nusselt number variation with Reynolds number between circular and triangular fin heat exchangers.

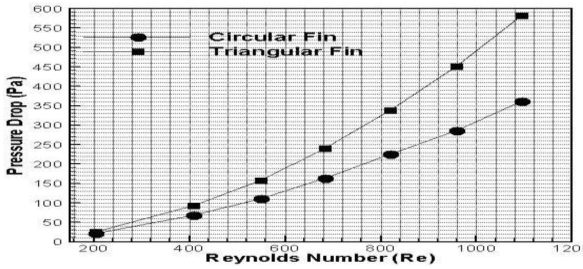


Fig.4 Comparison of pressure drop variation with Reynolds number between circular and triangular fin heat exchangers.

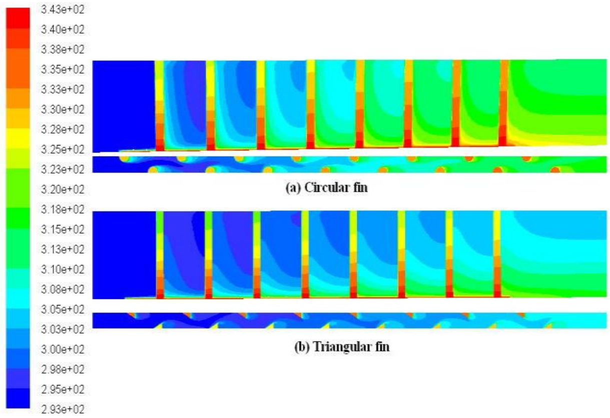


Fig.5 Temperature contours for circular and triangular fin heat exchangers at Re =550.

In contrast, the pressure drop for circular fin heat exchanger is less than that for the triangular fin at each Reynolds number (Fig.4). As the inlet velocity increases, the rate of rise of pressure drop for triangular fin is more than the rate of rise of pressure drop for circular fin. Hence it seems that the use of circular pin fin array is more effective than the use of triangular pin fin array for a given operating condition.

The temperature contours for circular and triangular fins are shown in Fig. 5. It is seen that the bulk temperature of fluid at the outlet for circular fins is more than that for the triangular fins. The fin temperature and bulk fluid temperature at any position in the computational domain is more for circular pin fins thereby meaning that the heat transfer is more in circular fins compared to that for triangular fins.

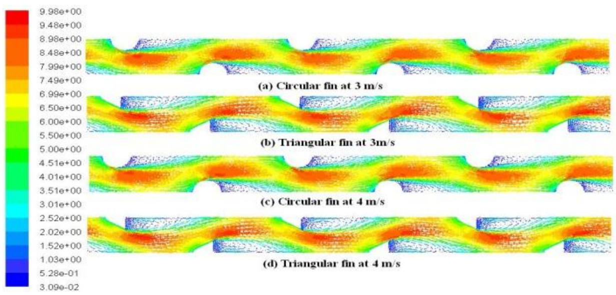


Fig.6 Streamlines for circular and triangular pin fin heat exchangers.

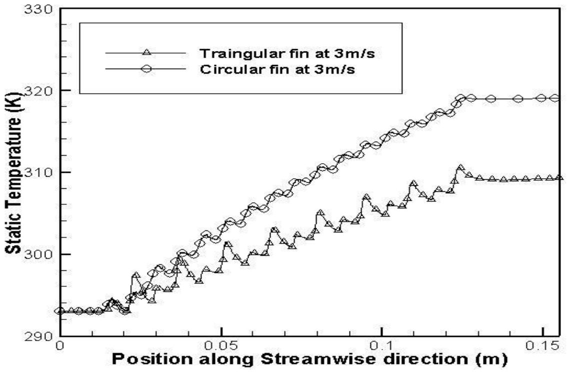


Fig.7 Static temperature variation

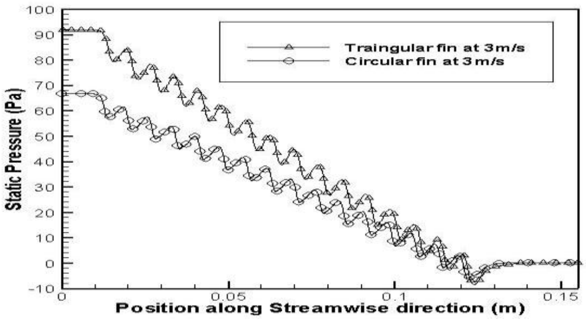


Fig.8 Static pressure variation

Fig.7 shows that the outlet temperature of circular fins is more than that of the triangular fins. Hence LMTD is more for circular fins than that of the triangular fins.

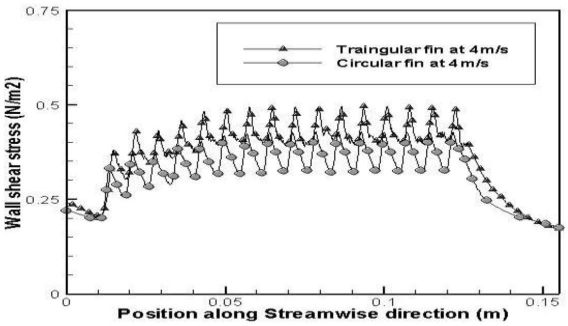


Fig. 9 Wall shear stress variation

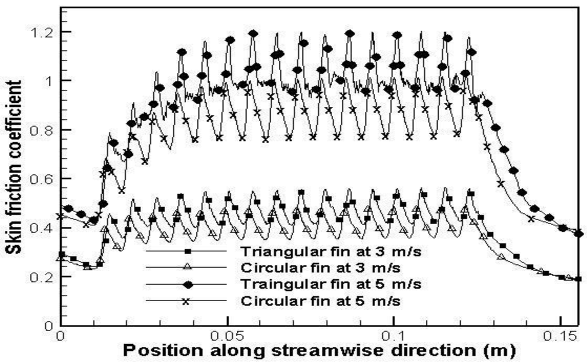


Fig.10 Skin Friction coefficient

The pressure drop, wall shear stress and skin friction coefficient is also more for triangular fins compared to that for circular fins (Figs.8, 9 and 10). Hence more pumping power is required in running a triangular pin fin heat exchanger compared to the circular pin fin heat exchanger at the same operating conditions.

5. Conclusion

The performance of a triangular pin fin was assessed and its comparison with that of circular pin fin was performed. The results showed that the circular pin fins perform far better than the triangular pin fins in terms of heat transfer and pressure drop for the same operating conditions. The streamlines showed that there is better mixing and more recirculation which causes more heat transfer with less pressure drop in a circular pin fin heat exchanger compared to that for triangular pin fin heat exchanger.

Nomenclature

A_c Minimum heat exchanger flow area
A_t Total heat transfer area
Bi Biot number
Cp Specific heat at constant pressure
D_h Hydraulic diameter
d Diameter of circular fins
dp Pressure drop
h Heat transfer coefficient of air
L Length of the fins
k Thermal conductivity
K Turbulent kinetic energy
Nu Nusselt number
P Pressure
Pr Prandtl number
Re Reynolds number ($= \rho_a u D_h / \mu_a$)
T Temperature
u Velocity vector
u_x, u_y, u_z Velocity components along x , y , z directions
x, y, z Streamwise, spanwise and vertical directions

Greek letters

ε Turbulent dissipation rate
μ Dynamic viscosity
ν Kinematic viscosity
ρ Density
τ Shear stress

Subscripts

a Air
in Inlet
h Hydraulic
out Outlet
t Turbulence
w Wall

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