**Special Issue of International Journal of Chemical Sciences**
**Featuring the Proceedings of National Conference**
**on Frontiers in Chemical Engineering 2007**
**Indian Institute of Technology, Guwahati**
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*Int. J. Chem. Sci.: 5(4), 2007, 1929-1937*

# THERMAL PERFORMANCE OF A CIRCULAR PIN FIN COMPACT HEAT EXCHANGER

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## ABSTRACT

The present paper deals with heat sinks that are primarily used in dissipating heat generated by electrical and/or electronic components and assemblies. The steady-state, forced-convective cooling of a horizontally based circular pin-fin assembly has been numerically investigated using the commercial CFD code FLUENT 6.2.16. The circular pin-fins in staggered arrangement are placed vertically in a horizontal heated base plate. The effect of the pin fin's thermal conductivity on the performance of heat exchanger is discussed. For this purpose, two fin materials, namely nickel and steel, are considered for comparing the thermal characteristics. The results are generalized, in a nondimensional form, and presented graphically. It has been shown that for a constant temperature of the base plate of the pin-fin assembly, the rate of heat transmission is increased by increasing the thermal conductivity of the pin-fins.

**Key words:** Rectangular duct, Circular pin-fins; Nusselt number; Turbulence modeling.

## INTRODUCTION

Compact heat exchangers are used in a wide variety of applications including air-conditioning condensers and evaporators and automotive radiators, etc. These heat exchangers are usually operated at low air-side Reynolds number (Re < 1000) to avoid high pressure drops. However, vortex shedding and turbulent flow at high Reynolds numbers can serve as heat transfer enhancement mechanisms. Ellison[1] presented the fundamentals of heat transfer and hydrodynamics characteristics of heat sinks. In his book, several useful theories and concepts were provided for heat sink analysis, e.g., fin efficiency, forced convective correlations, applications in heat sinks, etc. Dewan et al.[2] presented an overview of different methods proposed in the literature for heat transfer augmentation. Many analyses have been employed in the literature for investigating the thermal performance of heat sinks[3]. Lee[4] presented an analytical procedure for selecting optimum heat sinks based on the simulation results and

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illustrated the effect of various design parameters on the performance of heat sinks. Sahiti et al. [5, 6] 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. Due to the presence of the pin fins, the predicted values of the pressure drop show complex flow patters with large pressure gradients that generate vorticity, circulation and flow reversal. Moshfegh and Nyiredy[7] compared five different turbulence models for pin fin heat sinks, namely, the standard *k-ε* model, RNG *k-ε* model, the realizable *k-ε* model, the *k-ω* model, and the Reynolds stress transport model. The present work is an extension of the work by Dewan et al[8] who showed that that the RNG *k-ε* model provides the best agreement with the experimental data. They showed that the non-equilibrium wall approach results in a poor agreement with the measured data, while the standard wall function provides a good agreement with the experimental data. The effects of pin-fin geometry, pin-fin spacing, inlet air velocity, turbulence modeling have been investigated by many researchers for a compact heat exchanger. In the present investigation, the effect of thermal conductivity of the pin-fin material on the thermal performance of the compact heat exchanger is investigated.

## MATHEMATICAL MODEL & NUMERICAL SOLUTION

For an incompressible and steady state flow through a circular pin-fin compact heat exchanger, the governing mass continuity, momentum (Navier–Stokes) and energy equations can be written as:

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

$$\rho u_i \frac{\partial u_j}{\partial x_i} = -\frac{\partial p}{\partial x_j} - \frac{\partial \tau_{ij}}{\partial x_i} \qquad \ldots(2)$$

$$\rho c_p u_i \frac{\partial T}{\partial x_i} = k_f \frac{\partial^2 T}{\partial x_i^2} - \frac{\partial}{\partial x_i}(\rho c_p \overline{u_i' T'}) \qquad \ldots(3)$$

Where for Newtonian fluids, the molecular dependent momentum transport term is given by

$$\tau_{ij} = -\mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \frac{2}{3} \delta_{ij} k \qquad \ldots(4)$$

The following transport equations are used in the RNG *k-ε* model:

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left(\frac{\mu_{eff}}{\sigma_k}\frac{\partial k}{\partial x_j}\right)+G_k-\rho\varepsilon \qquad \ldots(5)$$

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}\left(\frac{\mu_{eff}}{\sigma_\varepsilon}\frac{\P\varepsilon}{\P x_j}\right)+C_{1\varepsilon}\frac{\varepsilon}{k}G_k-C_{2i}\rho\frac{\varepsilon^2}{k}-R_\varepsilon \qquad \ldots(6)$$

$$R_\varepsilon=\frac{c_\mu\rho\eta^3(1-\eta/\eta_0)}{1+\beta\eta^3}\frac{\varepsilon^2}{k} \qquad \eta=S\frac{k}{\varepsilon} \qquad \ldots(7)$$

The model constants are $C_\mu = 0.0845, C_{1\varepsilon} = 1.42, C_{2\varepsilon} = 1.68, \sigma_k = \sigma_\varepsilon = 0.7178, \beta = 0.012$. Turbulent heat fluxes in the thermal energy equations are modeled as

$$-\rho\overline{u_i'T'}=-\frac{\mu_t}{Pr_t}\frac{\partial T}{dx_i} \qquad \ldots(8)$$

The eddy viscosity is computed as –

$$\mu_t = \rho C_\mu k^2/\varepsilon \qquad \ldots(9)$$

The governing equations were solved using a commercial CFD package FLUENT 6.2.16. The computational domain used in the present work shown in Fig. 1 is a three-dimensional rectangular duct with pin fins mounted on the base plate. The flow inside a 3D channel is characterized by a complex flow field, which is affected by the blockage and recirculation zones. The pre-processor GAMBIT was used to create the geometry and a computational mesh.

## Boundary conditions

The inlet (velocity inlet), outlet (pressure outlet), wall and symmetry boundary conditions were applied in the computational domain. The boundary conditions are (a) For the inlet section 1-8-16-9 :

$u(0,y,z)=u_{in}$, $v(0,y,z)=0$, $w(0,y,z)=0$ and $T(0,y,z)=T_{in}=293K$

(b) For the bottom heated wall 2-3-6-7:

$u(x,y,0)=v(x,y,0)=w(x,y,0)=0$ and $T(x,y,0)=T_w=343K$

(c) For the section 1-4-5-8:

$$u(x,y,0)=v(x,y,0)=w(x,y,0)=0 \text{ and } \left(\frac{\partial T}{\partial z}\right)_{x,y,0}=0$$

(d) The top wall 9-12-13-16 was considered to be adiabatic, where the no slip condition for the velocity components was applied:

$$u(x,y,z)=v(x,y,z)=w(x,y,z)=0 \text{ and } \left(\frac{\partial T}{\partial z}\right)=0$$

(e) For sections 1-4-12-9 and 8-5-13-16, the symmetry boundary condition was applied and in the outlet section 4-12-13-5, the pressure outlet boundary condition was used.

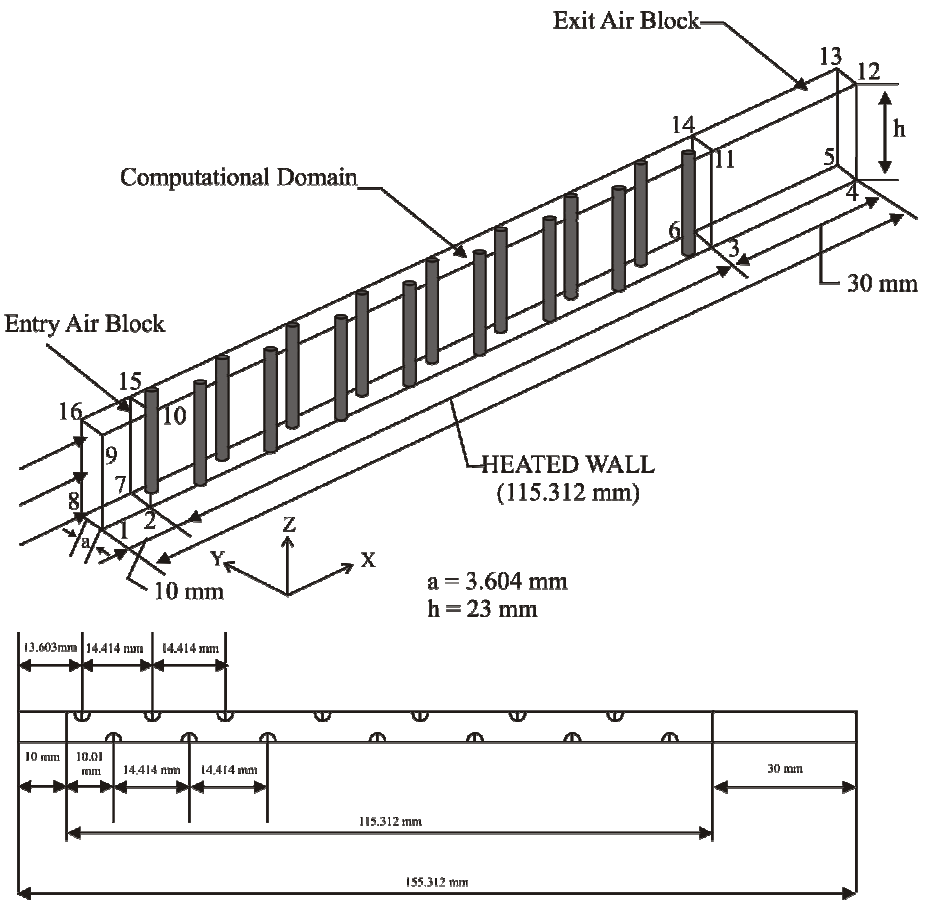



**Fig. 1: Computational domain**

The commercial software FLUENT 6.2.16 was used to solve the governing equations using the above mentioned boundary conditions. It employs the finite volume technique for solving the governing equations. The fluid flow inside the heat exchanger channels was considered to be incompressible and steady. The RNG $k$-$\varepsilon$ model with the standard wall function was used for modeling the turbulence while the conjugate heat transfer model was used for solving the energy equation. The second order upwind scheme was used for the discretization of all the governing equations. The SIMPLE algorithm was used for the pressure-velocity coupling. The fluid was chosen to be air with constant specific heat (= 1006.43 J/kg-K) and thermal conductivity (k = 0.0255 W/m-K). Two materials nickel (thermal conductivity k = 91.74 W/m-K) and steel (k =16.27 W/m-K) were used separately for the solid parts (fins) of the computational domain. Percentage turbulence intensity and hydraulic diameter were specified while solving the turbulence equations. For internal flows, the value at the inlets totally depends on the upstream history of the flow. The value of turbulent intensity was calculated using an empirical formula proposed by FLUENT 6.2.16 which was in the range of 5 to 6%.

## Grid independence and code validation

Grid independence study and the validation of the code were performed by Dewan et. al [8]. Based on their study a mesh consisting of 8,10,000 cells and a convergence criterion of $10^{-4}$ were used for all the computations. The geometric parameters taken in the present work are similar to those used by Kays [9] in their experiments. The staggered pin fin surface which was manufactured by Kays [9] with a good and clean pin-to-plate bond is close to the present smooth surface pins.

# RESULTS AND DISCUSSION

Heat transfer characteristics of both fin materials are shown in Fig. 2. The fin material with higher thermal conductivity (nickel) has higher heat transfer coefficient at the same Reynolds number compared to that for steel. The characteristics of the convective heat transfer from pin arrays can be understood from the temperature contours in fluid and solid parts of the computational domain.

The complete temperature changes in the computational domain, for the plane which coincides with the section 1-4-12-9 and for the plane which passes through half of pin length (z = 11.5 mm) of Fig. 1 for the inlet velocity 6 m/s are presented in Fig. 3. The corresponding Reynolds number is 822 and the average Nusselt number (Nu) for Ni is 10.47 and average Nu for steel is 5.28. It can be seen that the fluid adjacent to the fin attains the maximum temperature in both cases. The temperature difference between the fin and the fluid decreases in the downstream direction of the computational domain. Since the fluid takes heat from the

fin and surface, the bulk temperature of the fluid increases due to the turbulent convective heat transfer and this increase is more for nickel fin than that for the steel fin. Since the base plate thickness is negligible, the material of the base plate does not affect the solution.

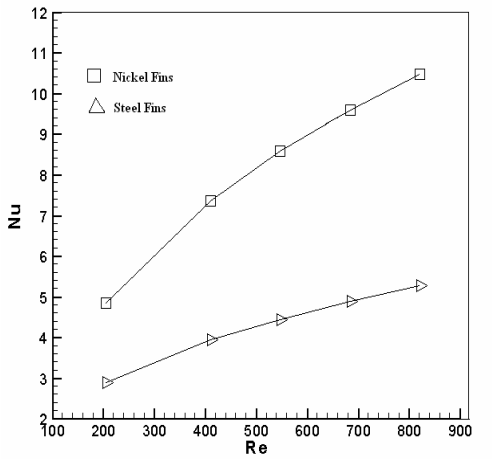



**Fig. 2: Nusselt number versus Reynolds number**

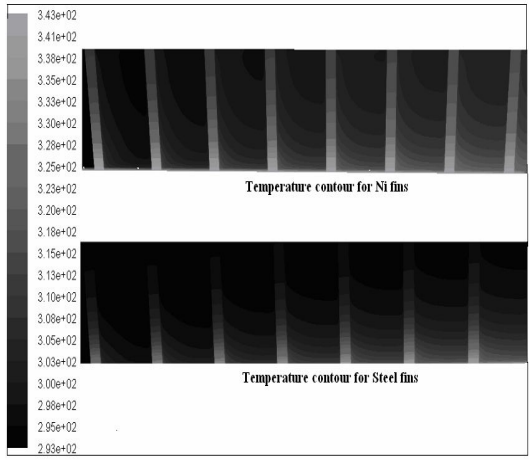



**Fig. 3: Temperature contours**

Figs. 4 to 6 show the temperature and pressure distributions along all the three directions for both fin materials.

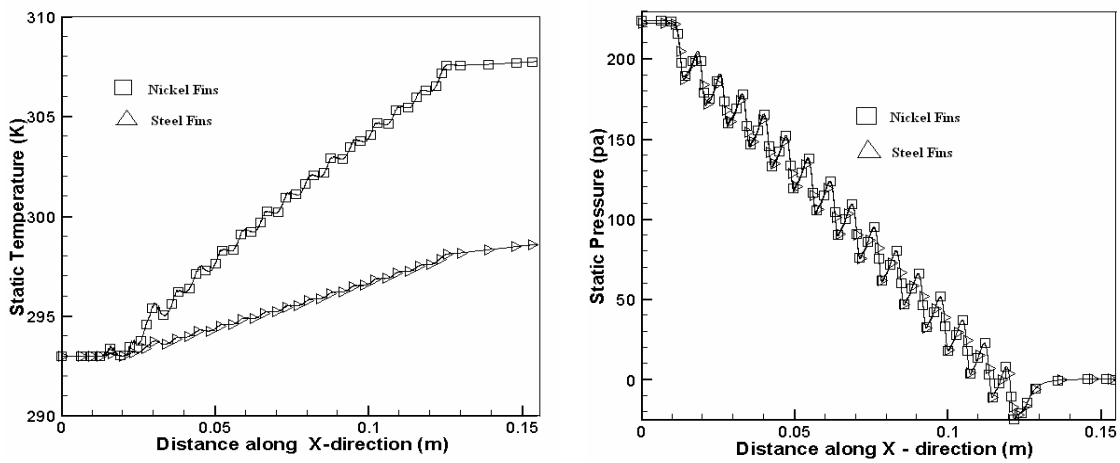



**Fig. 4. Static temperature and static pressure variation along the streamwise (x) direction**

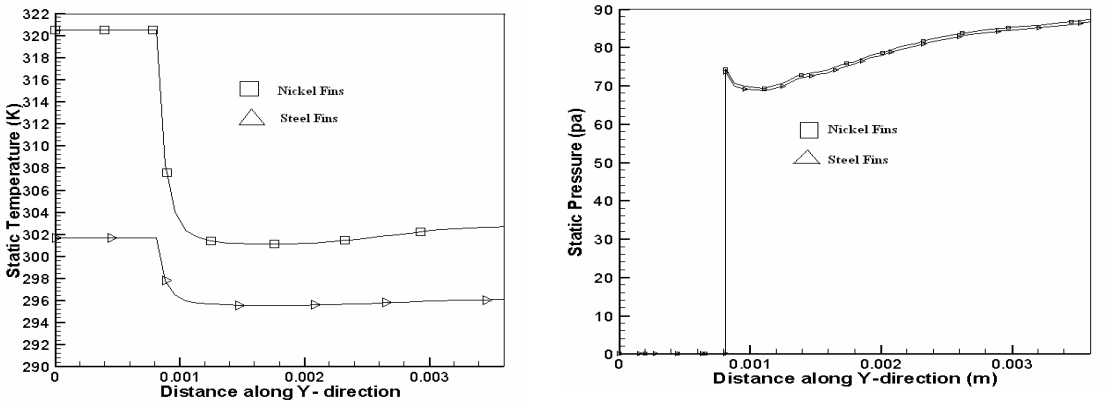



**Fig. 5: Static temperature and static pressure variations along the spanwise (y) direction**

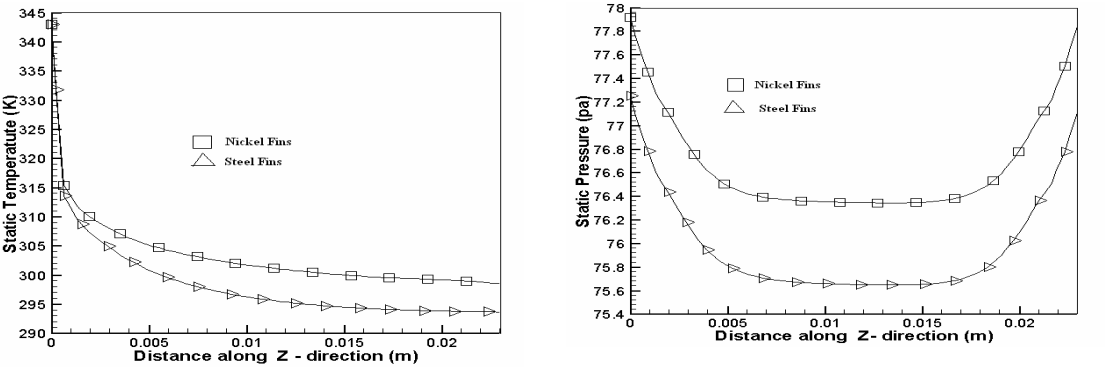



**Fig. 6: Static temperature and static pressure variation along the vertical (z) direction**

All plots are drawn for inlet air velocity of 6 m/s. Pressure and temperature fluctuations occur along the computational domain. One interesting observation is that the pressure variations along the streamwise direction overlap each other for both materials, slight variations occur along the spanwise direction and large variations occur along the vertical direction. The temperature increases along the streamwise direction and the opposite happens along the vertical direction. Along the spanwise direction, the temperature profile has a constant portion followed by a sharp decrease and then slow increment.

## CONCLUSION

A numerical investigation was performed to study the thermal behavior in a rectangular duct with pin fins in staggered arrangement for a turbulent incoming flow using the

RNG *k-ε* model with the standard wall functions. The present investigation shows that besides the fin geometry, fin material also plays a major role in compact heat exchanger performance. The fin material with a high thermal conductivity provides high heat transfer rate. The temperature contours illustrate that there is more conduction and convection heat transfer for nickel with high thermal conductivity than that for the lower thermal conductivity material steel.

## Nomenclature

| | | |
|---|---|---|
| $d_h$ | : | Hydraulic diameter |
| h | : | Heat transfer coefficient |
| Nu | : | Average Nusselt number (= h dh / ka ) |
| k | : | Thermal conductivity, turbulent kinetic energy |
| p | : | Pressure |
| Q | : | Heat flux |
| Re | : | Reynolds number (= $\rho_a$ u $d_h$ / $\mu_a$ ) |
| T | : | Temperature |
| Pr | : | Prandtl number |
| u, v, w | : | Cartesian velocity components |

### Greek Letters

| | | |
|---|---|---|
| ε | : | Effectiveness, heat exchanger efficiency |
| η | : | Efficiency |
| μ | : | Dynamic viscosity |
| ν | : | Kinematic viscosity |
| ρ | : | Density |
| τ | : | Momentum transport term |
| $\mu_t$ | : | Turbulent viscosity |

### Subscripts

| | | |
|---|---|---|
| a | : | Air |
| f | : | Fin |
| in | : | Inlet |

| | | |
|---|---|---|
| out | : | Outlet |
| w | : | Wall |
| t | : | Turbulence |

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