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Influence of Chimney Flow Pattern on Natural Convection Heat Transfer of Open Channel Finned Plates

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1. Introduction

The passive cooling system driven by natural forces has been continuously studied for the cooling of nuclear systems[1-3]. The enhancement of the efficiency and effectiveness of the passive cooling system, have long been the topic of those studies.

In this study, we investigated the heat transfer enhancement of finned plates, especially the chimney effect appeared in finned plates. The fin not only enlarges the heat transfer area but also draws fresh fluid from the open side of the channel composed of the base plate and fins, which further enhances the cooling capability of finned plate – a chimney flow pattern.

This study aims at investigating the influence of the chimney flow pattern on the natural convection heat transfer of the finned plate. To analyze the phenomenological study, both experimental and numerical analyses were performed.

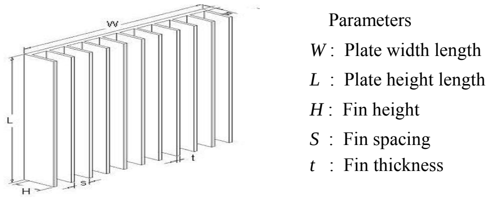


Fig. 1. Parameters of rectangular plate fin.

2. Previous studies

Natural convection heat transfer phenomena for vertical plates are well known and many heat transfer correlations have been developed. For the laminar natural convection, Bejan [4] performed the scale analysis from governing equations and proposed a scale relation. Le Fevre suggested the Equation (1) for the laminar natural convection heat transfer correlation for vertical plates [5].

$$Nu_H = 0.67(Gr_H Pr)^{1/4} \text{ at } Gr_H < 10^9 \quad (1)$$

The steady-state natural convection heat transfer from aluminum vertical rectangular fins extending perpendicularly from vertical rectangular base was investigated experimentally. Experiments were performed for fin heights varied from 5 to 25 mm. The convective heat transfer rates from the fin arrays increases with fin height and base-to-ambient temperature difference [6].

As the fin height increases, heat transfer from the fins to the fluid is directly proportional to the surface area of the fins and it is observed that distinguishably more fluid enters from the open side of the heat sink throughout its length [7].

3. Analyses

3.1 Experimental analysis

Figure 2 shows the experimental apparatus. It was consisted of an open top acrylic tank, and an anode and a cathode that was made of copper. The size of acrylic tank is of width 0.4m, height 0.3m, and length 0.4m. The sulfuric acid concentration and the copper sulphate concentration were 1.5M and 0.05M respectively. The Prandtl number was 2,014. As natural convection occurs by the buoyance due to the reduced cupric ion concentration near the cathode surface, the finned plate is used as the cathode. A flat plate was used as the anode. The base plate width and length are 5.0×10^{-2} m, fin thickness is 3.0×10^{-3}, and the fin spacing is 3.0×10^{-3} the fin height is various among 2.5×10^{-3}m-5.0×10^{-2}m In order to compare with experimental results, three different fin height were tested: 2.5×10^{-3} - 1.25×10^{-2} m. The electric potential was applied by a power supply (Vu¨POWER, DC Power supply-AK1205) and the current was measured by a dual-display multi-meter (METEK, MXD-4660A).

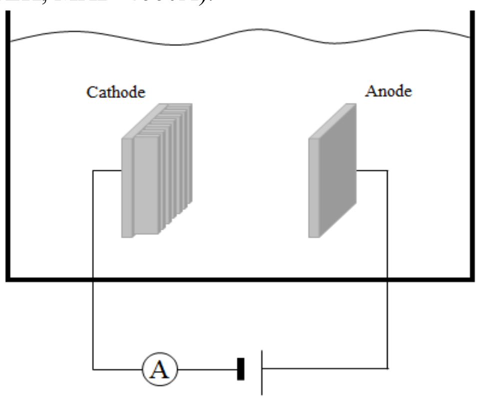


Fig. 2. Electrical circuit.

3.2 Numerical analysis

The basic governing equations for natural convection laminar flow on a finned plate, are as follows;

• Continuity

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

• Momentum

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=\nu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)-\frac{1}{\rho}\frac{\partial p}{\partial x}, \tag{3a}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=\nu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)-\frac{1}{\rho}\frac{\partial p}{\partial y}+g\beta \tag{3b}$$

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=\nu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right)-\frac{1}{\rho}\frac{\partial p}{\partial z}. \tag{3c}$$

• Energy

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right). \tag{4}$$

The boundary conditions are

$$u=v=w=0 \text{ at all the walls.} \tag{5}$$

$$T=T_0 \text{ on heated wall.} \tag{6}$$

$$T=T_\infty \text{ as fluid} \tag{7}$$

The plate length (L) and the fin spacing(S) were 5.0×10^{-2}m and 3.0×10^{-3} m, respectively. The Prandtl number was fixed at 2,014. Table 1 presents the test matrix for the numerical analysis and the underline case also carried out experiments.. The single Grashof number of 4.8×10^{6} presents a laminar flow condition. This aimed at investigating the influence of Chimney flow pattern.

Table 1. Test matrix for numerical analysis.

Pr	Fin spacing(m)	Length of base plate(m)	Fin height(m)
2,014	0.003	0.050	0.0025, 0.005, 0.0075, 0.010, 0.0125, 0.020, 0.030, 0.040, 0.050

As shown in Fig. 3, the numerical analysis was performed for a single fin domain with symmetric conditions for adjacent fins and pressure outlet condition for the pool region. The heated wall was 300 K of a constant temperature condition. The temperature of the fluid was 200K. To calculate the natural convection heat transfer as an open channel, pressure inlet and outlet conditions were set to simulate the open channel condition. All of pressure was set to zero. Average CPU time was about 8 hours using the Intel-core, 3.4 GHz CPU with 4GB RAM computer.

The numerical analysis calculation programs that generate the solution grid, GAMBIT 2.4.6, and calculate the numerical analysis, FLUENT ver. 6.3.26 were used.

The pressure equation was solved by PRESTO (PREssure STaggering Option). The segregated solver was used with a second-order upwind algorithm for momentum and energy equation in the laminar model. For pressure discretization, the standard algorithm was adopted, whereas the SIMPLE (Semi-Implicit Method for the Pressure Linked Equations) algorithm was used for pressure–velocity coupling discretization.

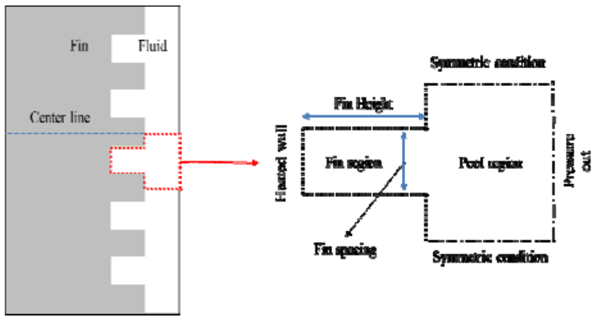


Fig. 3. Plane view of the simulated geometry.

3. Results and Discussion

Figure 4 shows the numerical results and experimental results. The blue line presents the natural convection heat transfer correlation at the flat plate by Le Fevre. The Nusselt number of both results is higher than the correlation. It means that the fin grows up the heat transfer. The average relative error between these two sets of data was less than 1%, which indicates that the numerical analysis was reliable. The Nusselt number has increased until fin height is 0.015m. After 0.015m the Nusselt number decreased. The reason of peak is explained by chimney flow pattern effect. The fin makes a chimney effect that draws the flow from pool region to fin region called chimney flow pattern. Until fin height is 0.015m the chimney flow pattern has increased, and over the 0.015m the chimney flow pattern is not able to effect to the baseplate. That is why the Nusselt number decreased. Therefore the optimal fin height(peak) exists.

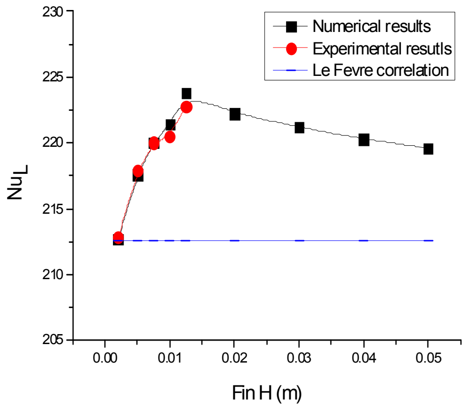


Fig. 4. Comparison between numerical results and experimental results with Le Fevre correlation.

As following fig. 5, the thermal boundary layer was appeared extremely thin at the heated wall in Pr 2,014. Because the thermal boundary layer is very thin at the high prandtl number [8]

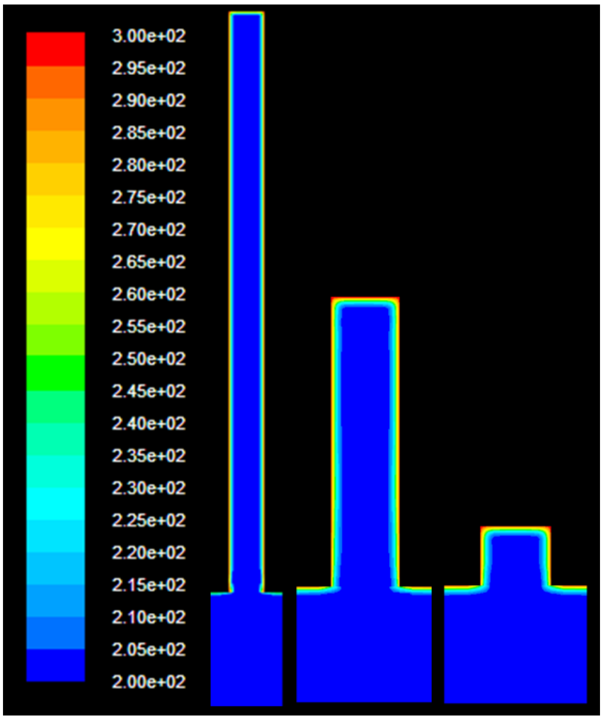
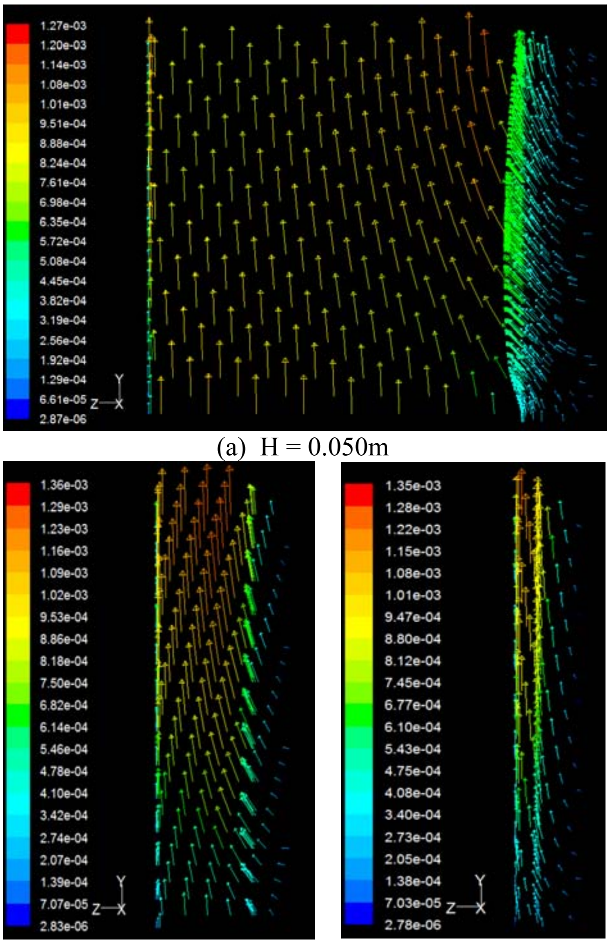


Fig. 5. Temperature field for top of fins.

Figure 6 presents the vector field with velocity profiles.
The fastest velocity occurred at the fin region which effect on the chimney flow pattern. As following the vector direction, the buoyancy draw the fresh fluid from the pool region. And the case (a) shows that the chimney flow pattern measured not to effect to the inside of fin when the fin height is high.



(a) H = 0.050m

(b) H = 0.015m (c) H = 0.0025m

Fig. 6. velocity field by vector direction.

4. Conclusions

Numerical analysis was performed for the natural convection heat transfer of a finned plate in an open channel. In order to investigate the influence of the chimney flow pattern the heat transfer, several fin height were simulated and compared.

The temperature profiles varied drastically depending on the values of the Prandtl number. As the Prandtl number increases, the thermal boundary layer reduces.

The comparisons of the results with Le Fevre natural convection heat transfer correlation for vertical plate shows that as the fin height increases, the Nu_L of the finned plate becomes large until the specific value by the fins.

Acknowledgment

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