Numerical Investigation of Natural Convection Heat Transfer from V-fin Arrays with Constant Heat Flux

Jebir SK*
Department of Mechanical Engineering, AL-Nahrain University, Iraq

Abstract

Natural convection heat transfer from rectangular V-fins had been investigated numerically with different heat flux values (175, 350, 525, 700 and 875 Watt per square meter). Fin thickness (5) mm, fin high (18) mm, space between fin and other (10) mm, the heat sink base plate was heated by an attached maximum electric heater 2225 W/m² with an identical size as the base plate. The mathematical model of the base plate and fins are solved numerically using a COMSOL (5.0) after describing the mesh model using the COMSOL (5.0) and assume the properties of air variation with film temperature. After finding the numerical results make validation between numerical and the experimental results, where found good agreement between them. Empirical correlations for the overall Nusselt number versus average Rayleigh numbers for these configurations are obtained and compared to other correlations cited in the literature. The range of Rayleigh numbers, Nusselt number and base plate temperature, 1.7*10⁷ to 12*10⁷, (37°C to 83°C) and (25.6°C to 81.34°C).

Keywords: Heat flux; Nusselt number; Rayleigh number

Introduction

Natural convection heat transfer from these heat sinks occur when there is a relative motion between a hot surface and a fluid flowing over the surface and there is a temperature difference between the surface and the air. If the fluid motion is due to density difference caused by temperature variation in the fluid, then it is called natural convection. The convective heat transfer rate depends on the properties of the fluid flow Natural convective heat transfer from a heat sink with rectangular fin has been studied for many years, a comprehensive review of these studies are presented in many heat transfer. By numerical study such as, Senol baskaya and mecil and oce [1] focused about effect of parameter (length, width, high, spacing and the temperature) on natural convection heat transfer. Fins made from aluminum in rectangular shape. Heat sink in horizontal position solved case by (CFD) this study model.

Abdullatif Ben-Nakhi and Ali J. Chamkha [2] are focused on the analytical study of steady state heat transfer, laminar flow, natural convection in a square plate base enclosure with an inclined thin rectangular fin. Fins material was aluminum the range of Rayleigh (104-108). A numerical solution based on the finite-volume method is obtained. Aularasan R. and veraj R. [3] designed modern heat sink fins made from aluminum in rectangular shape. Heat sink in horizontal position solved case by (CFD) this study model.

M. Baris and Mahmetarik [4] studied the effect of much material (copper, aluminum, parotic graphite thermal annealed on fin efficiency where is respect main factor to making electronic devise. Yaciln et al. [5] studied about natural convection heat transfer from a fin array in horizontal position. CFV code used to solve fin model. The range of Rayleigh number is (2*10⁴ - 3.5*10⁷). Ali Al-Qusam [6] had studied the execution numerical steady of natural convection heat transfer from rectangular fins. Fins made from aluminum. Heat sink in horizontal position the range of Rayleigh number (4*10⁷- 2*10⁸), range of high (0.1-0.5) m.


Sam [9] studied natural convection from rectangular interrupted fins in horizontal position. Where the fins made from aluminum, the continuous heat sinks of different designs have been carried through (CFD) simulations. The range of input power (5-25) w and the range of Rayleigh number (104 - 107).

Numerical Analysis

The governing equations, boundary conditions, numerical domain and the corresponding, the assumptions and the mesh independency are discussed. Some of the present numerical results are also presented in this chapter as validation where compared against well-established analytical model available in the literature.

Computational Domain

Different patterns of heat sinks have been modeled. The heat sink geometry can be shown in Figure 1.

Governing Equations

The heat transfer in the heat sink is take place in three ways, conduction, convection, and radiation. The temperature field is obtained by solving the energy equation Maher [10]. The heat conduction in solid is governed by;

\[ \rho c_p \frac{\partial T}{\partial t} - \nabla \cdot (k \nabla T) = Q \]  

(1)

The heat convective from all external surfaces to ambient is governed by;

*Corresponding author: Jebir SK, Assistant professor, Department of Mechanical Engineering, AL-Nahrain University, Iraq, Tel: +964 770 733 1007; E-mail: suhakareem@eng.nahrainuniv.edu.iq

Received April 18, 2016; Accepted May 16, 2016; Published May 20, 2016


Copyright: © 2016 Jebir SK. This is an open-access article distributed under the terms of the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original author and source are credited.
Fluid flow and heat transfer in a horizontal finned heat sink. For modeling the channel, since the geometry repeats under the no-slip boundary condition because the air flow over heat sink does not have relative velocity from fins where the hot air ascends and cold air lands. No slip boundary condition is very low where the velocity of air caused by thermal radiation within the enclosure opens from the top. In this case, the velocity of air above the heat sink is considered to be convergent when the relative error was less than 1.0×10^-6 in each field between two consecutive iterations. We see in group of figures the increase of mesh even incoming to mesh independence that’s give optimum value of measurement factor at 25 Watt as a heat flux and (25°C) at an environmental temperature. Complete mesh consists of 36294 domain elements, 21206 boundary elements, and 2622 edge elements from Figures 2 and 3.

\[ -n(-kVT) = h(T_{amb} - T) \] (2)

The heat radiation from all external surfaces to ambient is governed by,

\[ -n(-kVT) = h(T_{amb} - T) \] (3)

The following is a summary of the assumptions made to model the fluid flow and heat transfer in a horizontal finned heat sink.

- Steady state, laminar flow, i.e., Rayleigh number \(Ra < 10^6\).
- Incompressible flow
- Two-dimensional flow and heat transfer inside the channels.
- Symmetric flow and identical heat transfer in all the channels.
- Iso-heat flux boundary condition for the base plate
- Negligible air velocity entrance in side channels. (The fresh air inflow and outflow from the outmost channels was small compared to the air flow entering from the side of the fin array).

The physical properties of the fluid varied with temperature, density can be shown to follow a simple inverse relationship (ideal gas) with a small correction term:

\[ \rho = \frac{351.99 - 344.84}{T^2} \text{ kg/m}^3 \] (4)

\[ \mu = \frac{1.45927}{109.1 + T} \text{ kg/s} \] (5)

\[ k = \frac{2.334 \times 10^{-7}}{164.54 + T} \text{ W/m.K} \] (6)

Specific heat follows a quadratic relationship:

\[ Cp = 1030.5 - 0.199975T + 3.9743 \times 10^{-7} T^2 \frac{J}{(kg.K)} \] (7)

Boundary Conditions

The boundary conditions employed inside the enclosure where the enclosure opens from the top. In this case, the velocity of air above the heat sink is very low where the velocity of air caused by thermal radiation from fins where the hot air ascends and cold air lands. No slip boundary condition because the air flow over heat sink not have relative velocity to heat sink. For modeling the channel, since the geometry repeats itself, a single channel has been chosen to represent the computational domain according to the flow visualization and velocity measurement of the field flow for a finned plate reported in. Thus, a two-dimensional analysis (instead of three dimensional) is adequate for the purpose of our simulation.

Computational Grid

The governing equations were discretized using a finite-volume method and solved using COMSOL computational fluid dynamic (CFD) package. A computational quadratic meshes were used for all types of heat sinks. Independent of the grid size has been examined. The coupled set of equations were solved iteratively, and the solution was considered to be convergent when the relative error was less than 1.0×10^-6 in each field between two consecutive iterations. We see in group of figures the increase of mesh even incoming to mesh independence that’s give optimum value of measurement factor at 25 Watt as a heat flux and (25°C) at an environmental temperature. Complete mesh consists of 36294 domain elements, 21206 boundary elements, and 2622 edge elements from Figures 2 and 3.

Validation

The results of the CFD model were verified with experimental results. The results of the average temperature for the heat sinks of CFD model were verified with experimental results in the same condition of the external (ambient) temperature \(T_{amb}\) for different levels of power (heat flux). The computed average temperature shows in good agreement with the experimental average temperature measured in heat sinks.

Result and Discussion

Figure 4 indicated the variation between temperature different (base plate temperature minus from ambient temperature) in calicoes and heat input in watt for five models of fins with different geometry. The computed average temperature shows in good agreement with the experimental average temperature measured in heat sinks.

Figure 7 was prepared for sixth heat input in watts (5, 10, 15, 20, 25, 50 watts), note that the maximum heat transfer coefficient in meddle of heat sink width and decrease whenever approach to end of heat sink width because of the end losses by conduction heat transfer, where is the super heat sink must be insulated from ends and bottom to decrease the different between center temperatures and end temperature as much as possible to decrease heat transfer by conduction. Incropera in 2005 had studied the computational of heat transfer by conduction from bottom of heat sink 3% from heat transfer by conduction and convection so it neglected in heat transfer coefficient compute.
A. Mesh 1592 domain elements, 1038 boundary elements, and 529 edge elements
B. Mesh 4636 domain elements, 3228 boundary elements, and 1026 edge elements
C. Mesh 36294 domain elements, 21206 boundary elements, and 2622 edge elements
D. Mesh 40069 domain elements, 22718 boundary elements, and 2740 edge elements

**Figure 2:** Mesh independence of V-fin.
Average Nusselt numbers decrease with increasing the finning factor for same heat input, in this case must V-fin the lowest but heat transfer coefficient depended also on configuration of air flow.

References


Conclusions

1. For the present research, for the V-fin arrays, it can be concluded that Nu as a function for Ra is: $Nu = 1.1303 Ra^{0.2142}$.
2. The less average difference between base plate temperature and tip fin temperature in V-fin heat sink had higher efficiently from other cases and equal (78%).

