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

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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 an COMSOL (5.0) after describing the mesh model using the COMSOL (5.0) and assume the properties of air variation with film temperature. After find 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, $1.7*10^7$ to $12*10^7$, (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 ozec [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 base plate enclosure with an inclined thin rectangular fin. Fins material was aluminum the range of Rayliegh (104-108). A numerical solution based on the finite-volume method is obtained. Aularasan R. and veraj R. [3] designed modern heat sink to cooling electronic device in numerical work use (CFD) program to determine the natural convection from rectangular fin.

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. Yaclin et al. [5] studied about natural convection heat transfer from a fin array in horizontal position. CFD code used to solve fin model. The range of Rayleigh number is (2×104 - 3.5×107). Ali Al- Qusamy [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), rang of high (0.1-0.5) m.

Abdullah H and M. AL-Essa [7] focused about natural convection from rectangular fins in horizontal position the fin mad from aluminum material. Ilker Tari and mehdi [8] made comparing between horizontal and incline heat sink with rectangular fins for natural convection heat transfer.

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;

$$\rho C_p \frac{\partial T}{\partial t} - \nabla.(k\nabla T) = Q \tag{1}$$

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

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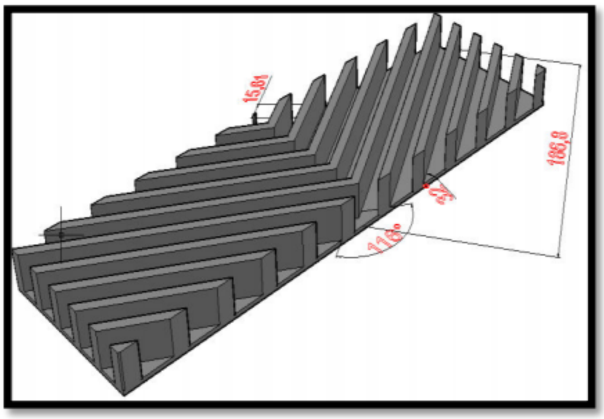


Figure 1: Sketch for the present study.

$$-n.(-k\nabla T) = h(T_{amb} - T) \quad (2)$$

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

$$-n.(-k\nabla T) = h(T_{amb} - T) \quad (3)$$

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 $<10^9$.
- 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:

$$\rho = \frac{351.99}{T} + \frac{344.84}{T^2}\left[\frac{kg}{m^3}\right] \quad (4)$$

$$\mu = \frac{1.4592\,T^{\frac{3}{2}}}{109.1+T}\left[10^{-6}\frac{(N.s)}{m^2}\right] \quad (5)$$

$$k = \frac{2.334\times10^{-3}\,T^{\frac{3}{2}}}{164.54+T}\left[\frac{W}{(m.K)}\right] \quad (6)$$

Specific heat follows a quadratic relationship:

$$Cp = 1030.5 - 0.199975T + 3.9743\times10^{-4}\,T^2\left[\frac{J}{(kg.k)}\right] \quad (7)$$

Boundary Conditions

The boundary conditions employed inside enclosure where the enclosure opens from 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 ascent and cold air landing. 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 ware used for all types of heat sinks. Independent of the grid size has been examined. The coupled set of equations ware 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 increase of mish even incoming to mesh independence that's give optimum value of measurement factor at 25Watt 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 Figures 2 and 3.

Validation

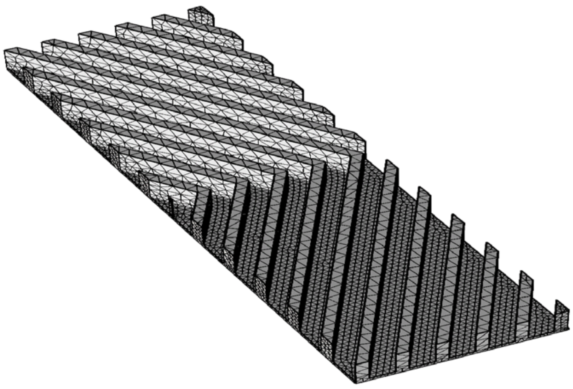
The results of the CFD model were verified with experimental results. The results of the average temperature for the heat sinks of CFD model were verified with experimental results in the same condition of the external (ambient) temperature (T_{ext}) for different levels of power (heat flux). The computed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

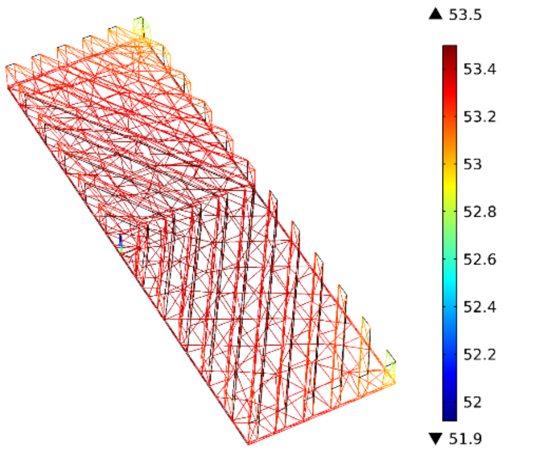
Result and Discussion

Figure 4 indicated the variation between temperature different (base plate temperature minus from ambient temperature) in calicoes and heat input in watt for five model of fins with deferent geometry (continues fins,1-interrupted fins ,4-interrupted fins, inclined fins and V-fins) , as a rule when increase the heat input, increase (ΔT) because of increase the convection and radiation heat transfer, as well as, note (ΔT) of 4-interrupted fins more than rest case becose of small surface area , where the surface area effect on heat transfer, in case of 4-interrupted fins have smallest surface area, lead to weak capacity to carry the heat and Couse high base temperature. They agreed with Senol Baskaya et al. [11] and Salila Ranjan Dixit and Dr. Tarinicharana Panda [12].

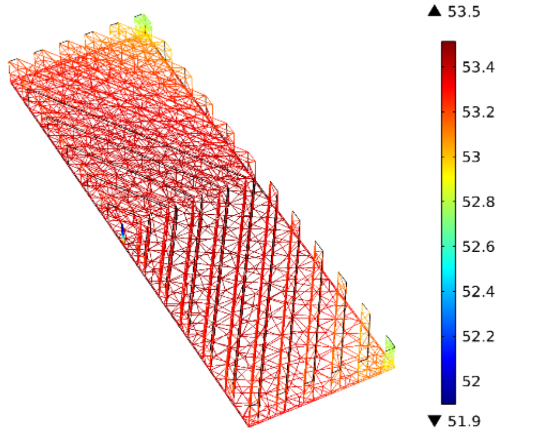
Figure 5 shows the variation of heat input and the tip fin temperature for five models of fins. The variation of heat transfer coefficient with length in (mm) shown Figure 6 indicates the variation between heat transfer coefficient and heat input for five configuration of fins. The result show when increase heat input increase heat transfer coefficient.

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.

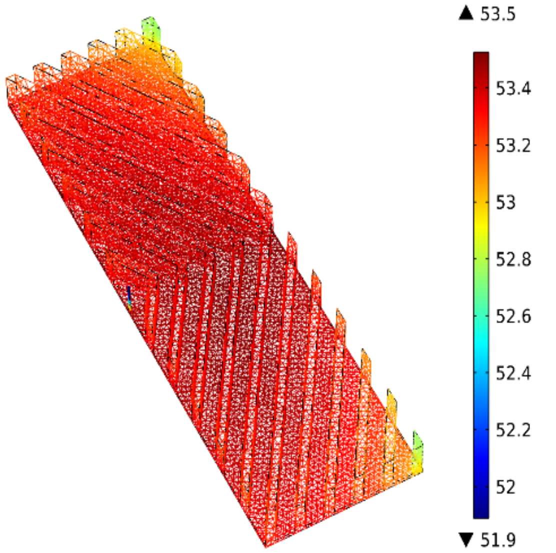




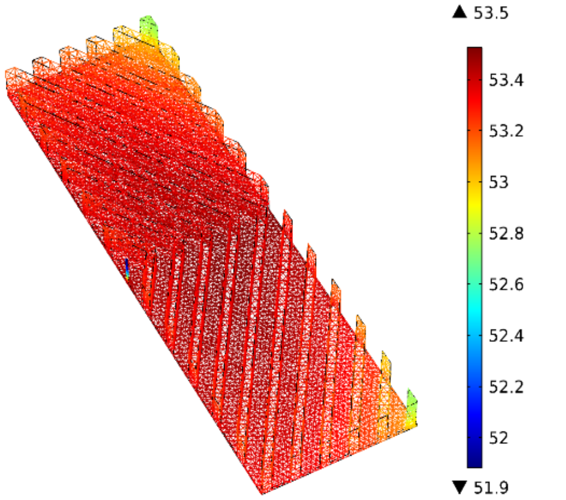
A. Mesh 1592 domain elements, 1038 boundary elements, and 529 edgeelements



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.

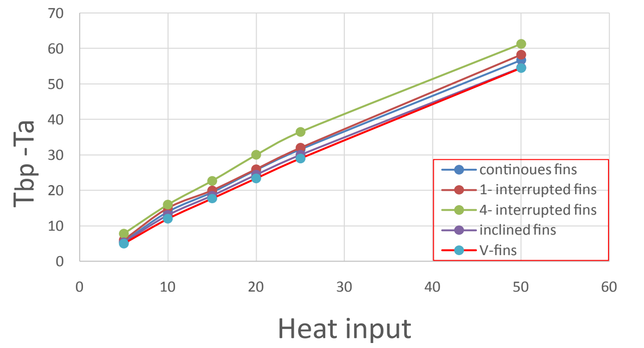


Figure 3: The variation between temperature different in calicoes and heat input.

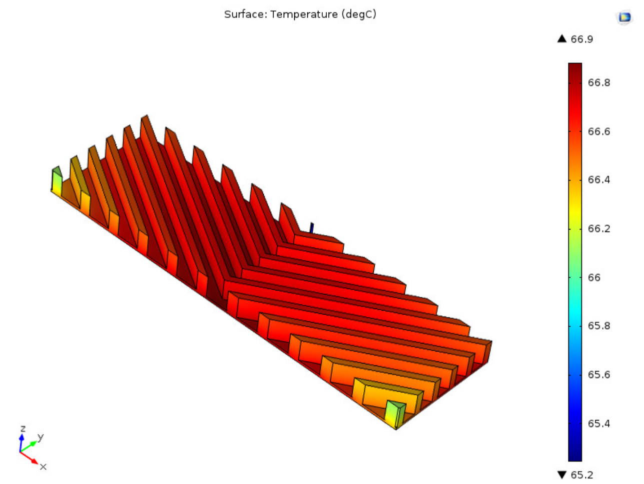


Figure 4: Base temperature distribution of V- fin.

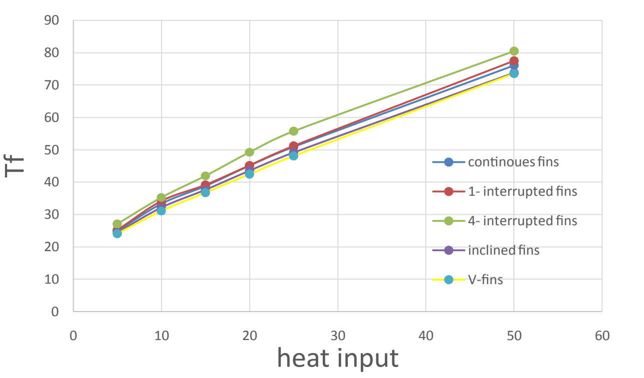


Figure 5: The variation of heat input and the tip fin temperature.

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%).

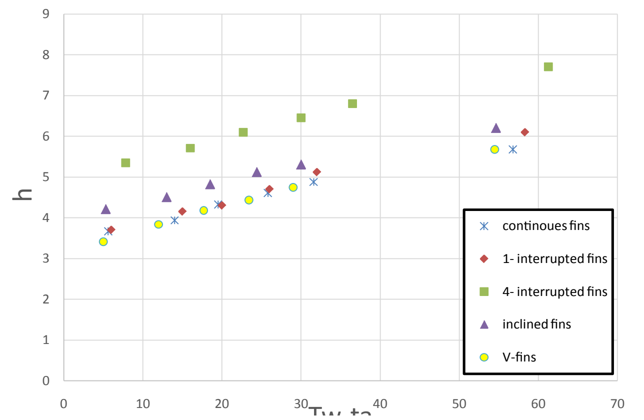


Figure 6: The variation between heat transfer coefficient and heat input.

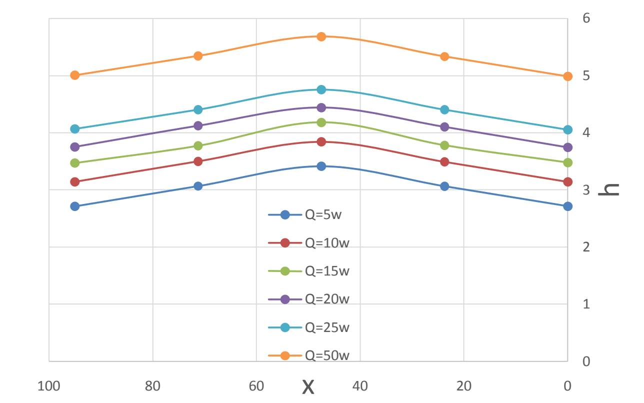


Figure 7: The variation of heat transfer coefficient with length in (mm) for V-fins.

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.

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