ISSN 2229-5518

Numerical investigation of natural convection heat transfer from V-fin arrays with constant heat flux

Suha K. Jebir
suhakareem@eng.nahrainuniv.edu.iq

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 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 , 1.3*10716*107, (37 – 83) and (25.6 – 81.34°C) .

Nomenclature

Symbol	Description	Units
A_{bp}	Area of un finned portion of base plate	m^2
A_{fp}	Projected area of pin fins m^2	
A_t	Total heat sink surface area (the sum of fins and base plate areas)m^2	
Asurw	Exposed area by radiation heat radiation m2	
g	Acceleration due to gravity m/s2	
H	High of the pin fin Mm	
h	Heat transfer coefficient W/m2. ºC	
I	The current across the heater A	
k	Thermal conductivity of air W/m.ºC	
L	Length of the base plate Mm	
Lc	characteristic length of the pin fin Mm	
Nu	Nusselt number ---	
Pr	Prandtl number ---	
Qcond	Conduction heat transfer W	
Qconv	Convection heat transfer W	
Qrad	Radiation heat transfer W	
Qnet	Net heat transfer W	
q	The removed heat by the cooling air from the fin array W	
Ra	Rayleigh number ---	
Tbp	Average surface temperature of base plate ºC	
Twall	Temperature of the wall of the environmental chamber ºC	
Tsurw	Temperature of the surrounding environmental chamber ºC	
Tamb	Temperature of ambaint ºC	
Tf	Film temperature ºC	
Tm	Mean temperature ºC	
ΔT	Temperature difference between base plate and surrounding air ºC	
ΔTLM	Log mean average temperature ºC	

Greek symbol

ISSN 2229-5518
σ Stefan–Boltzmann constant= 5.6697× 10-8 W/m2K4
β Thermal expansion coefficient 1/K
ρ Density of air Kg/m3
ν Kinematic viscosity of air m2/s
α Thermal diffusivity of air m2/s
ε Radiative emissivity ---
Φ Finning factor ---
Cp Specific heat of air KJ/kg. ºC
Subscripts
Symbol Description Units
bp Base plate ---
conv Convection ---

1.Introduction

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 (2000) [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. The study model studied by Abdullatif Ben-Nakhi and Ali J. Chamkha (2007) [2] They is 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 rang of Rayliegh (10^4 - 10^8). A numerical solution based on the finite-volume method is obtained .Aularasan R. and veraj R. (2008) [3] Design 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 (2008) [4] Steady the effect of many material (copper, aluminum, parotic graphite thermal annealed on fin efficiency where is respect main factor to making electronic devise .Yaclin .etc. (2008)

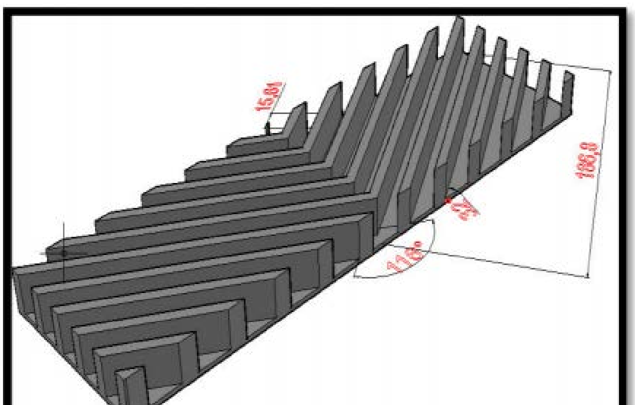
ISSN 2229-5518
[5]
They study about natural convection heat transfer from a fin array in
horizontal position . CFD code used to solve fin model. The rang of Rayligh number ($2*10^4$ - $3.5*10^7$).Ali Al- Qusamy (2011)[6] Execution
numerical steady of natural convection heat transfer from rectangular
fins. fins made from aluminum. heat sink in horizontal position the rang
of rayligh number ($4*10^7$- $2*10^8$) , rang of high (0.1-0.5)m .Abdullah H
and M. AL-Essa (2012) [7] Focus about natural convection from rectangular fins in horizontal position the fin mad from aluminum material ..Ilker Tari and mehdi (2013) [8] Making comparing between
horizontal and incline heat sink with rectangular fins for natural convection heat transfer . R.Sam .etc. (2013)[9] They steady 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 rang
of input power (5 - 25)w and the rang of Rayliegh number (10^4 - 10^7).

2.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.

3. Computational domain

T ig.1.



ISSN 2229-5518

Fig. 1 sketch for the present study

4. Governing Equations

The temperature field is obtained by solving the energy equation Maher [10] (2015). The heat conduction in solid is governed by;

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

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

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

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

$$-n.(-k\nabla T) = \varepsilon\sigma\left({T_{amb}}^4 - T^4\right) \quad \ldots \; 3$$

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:

$$\rho = \frac{351.99}{T} + \frac{344.84}{T^2} \qquad \left[\frac{kg}{m^3}\right] \qquad \ldots \; 4$$

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

$$k = \frac{2.334 \times 10^{-3} T^{\frac{3}{2}}}{164.54 + T} \quad \left[\frac{W}{(m.K)}\right] \qquad \dots 6$$

Specific heat follows a quadratic relationship:

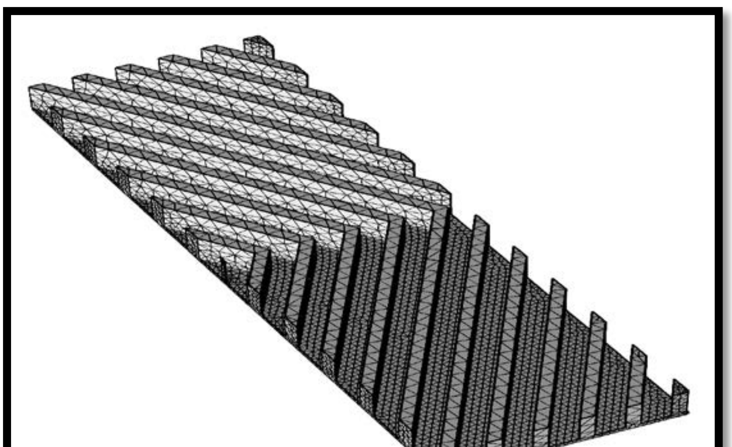
$$Cp = 1030.5 - 0.199975T + 3.9743 \times 10^{-4}\, T^2 \quad \left[\frac{J}{(kg.k)}\right] \quad ..7$$

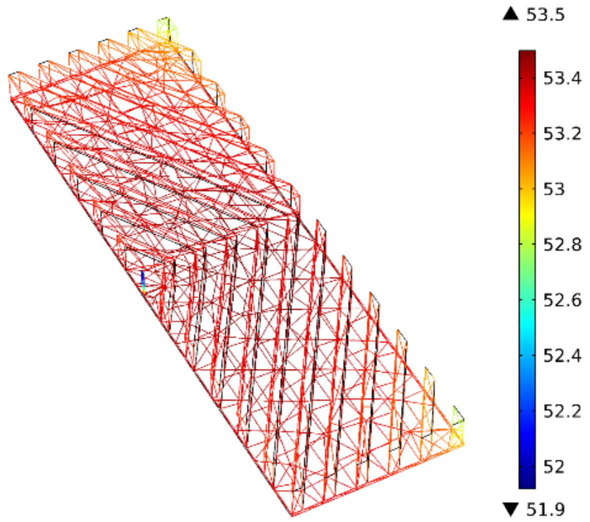
Boundary conditions

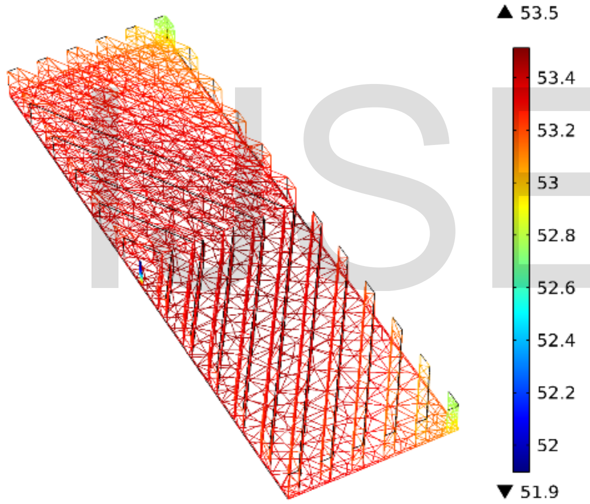
The boundary conditions employed inside enclosure where the enclosure open from top. In this case the velocity of air above the heat sink is very low where the velocity of air caused by thermal radiation from fins where the hot air ascent and cold air landing. no slip boundary condition because the air 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 package. A computational quadratic meshes ware used for this type of heat sinks. The set of governing equations ware solved numerically, and the solution was considered to be convergent when the relative error was less than 1.0×10^{-9} in each field between two consecutive iterations.




▲ 53.5
53.4
53.2
53
52.8
52.6
52.4
52.2
52
▼ 51.9


▲ 53.5
53.4
53.2
53
52.8
52.6
52.4
52.2
52
▼ 51.9

ISSN 2229-5518

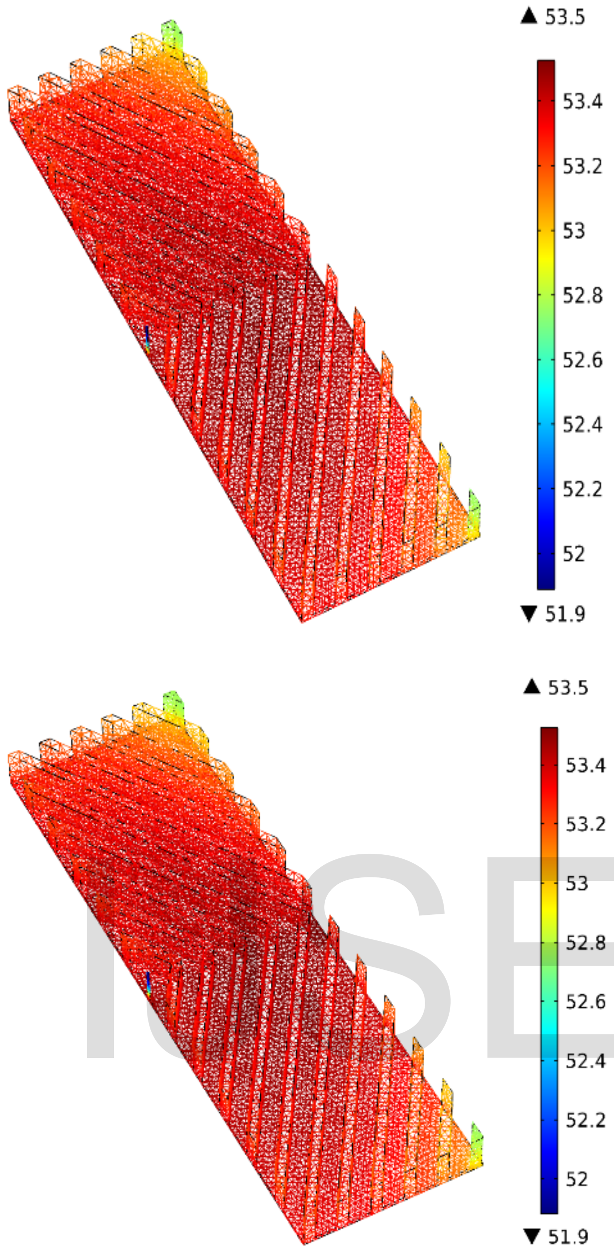


Fig.2 mesh independence of V-fin

For fig.2, V-fin (meshing element)

A- mesh_1682 domain elements, 1055 boundary elements, and 602 edge elements

B- mesh_4825 domain elements, 3354 boundary elements, and 1080 edge elements

C- mesh_3745 domain elements, 21206 boundary elements, and 2854 edge elements

D- mesh_40152 domain elements, 23814 boundary elements, and 2984 edge elements

Result and discussion

Fig.3 indicated the variation between temperature different (base plate temperature minus from ambient temperature) and heat input in watt for five model of fins with deferent geometry (V-fins with others type for comparison) , 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 cause high base temperature. This agreed with **Senol Baskaya etc. [1] (2000)** and **Salila Ranjan Dixit and DrTarinicharana Panda [12] (2013)**.

Fig (4) 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 . Fig.5 indicates the variation between convective heat transfer coefficient and heat input for five configuration of fins. The result show when increase heat input increase convective heat transfer coefficient.

Fig.6 was prepared for sixth heat input in watts (5,10,15,20,25,50 watt) . 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 (2005)**. 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.

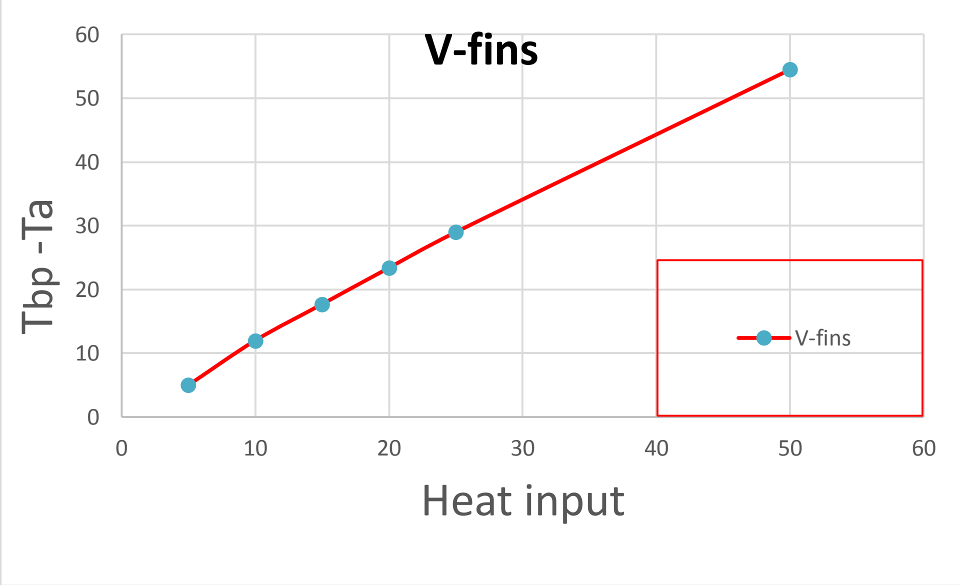


Fig.2 the variation between temperature different in and heat input.

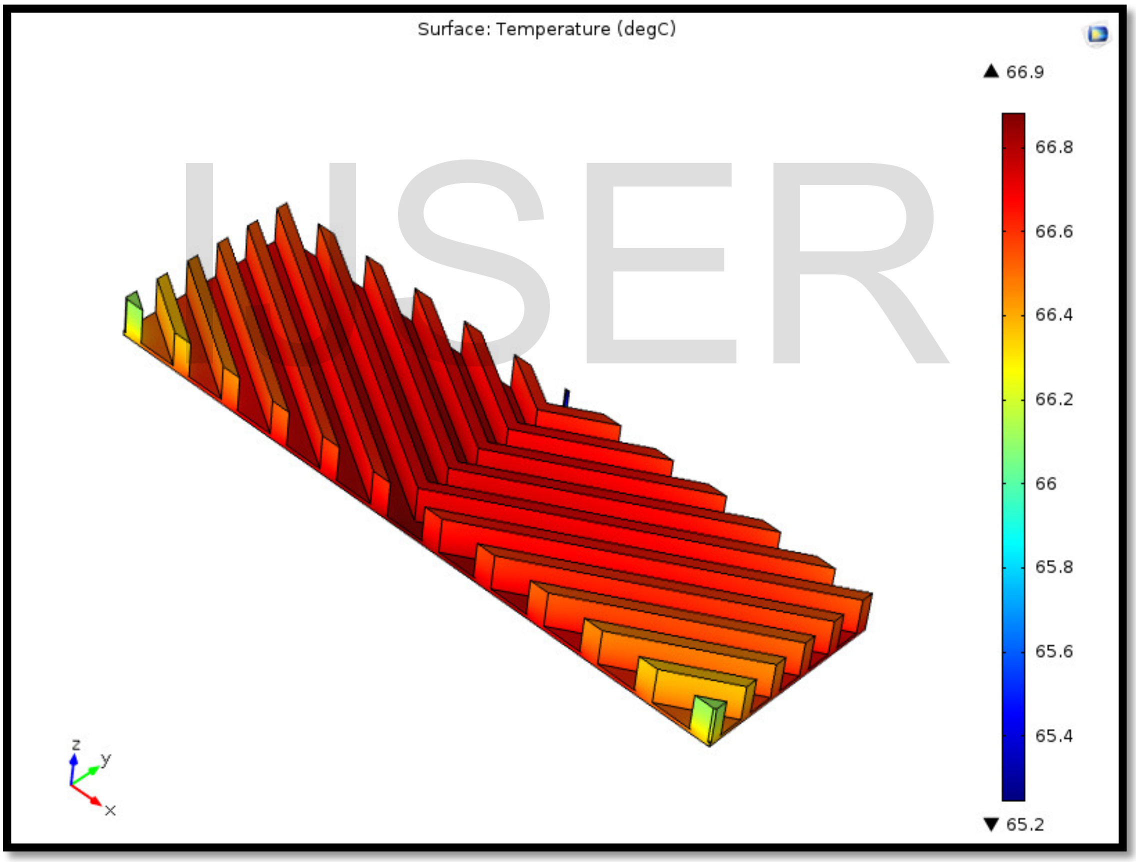


Fig.3 base temperature distribution of V- fin

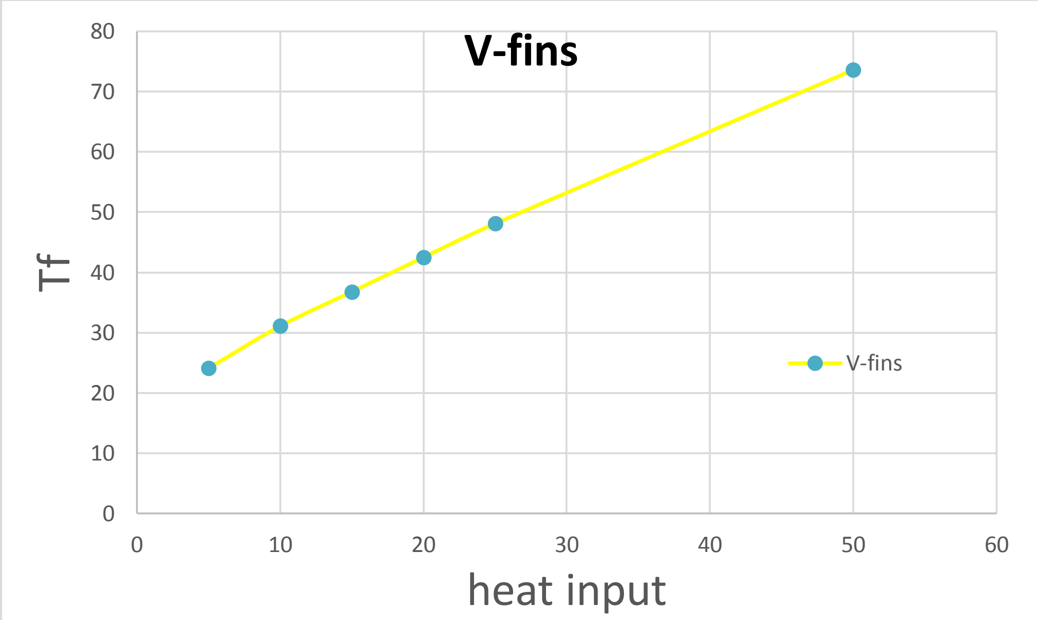


Fig. 4 the variation of heat input and the tip fin temperature

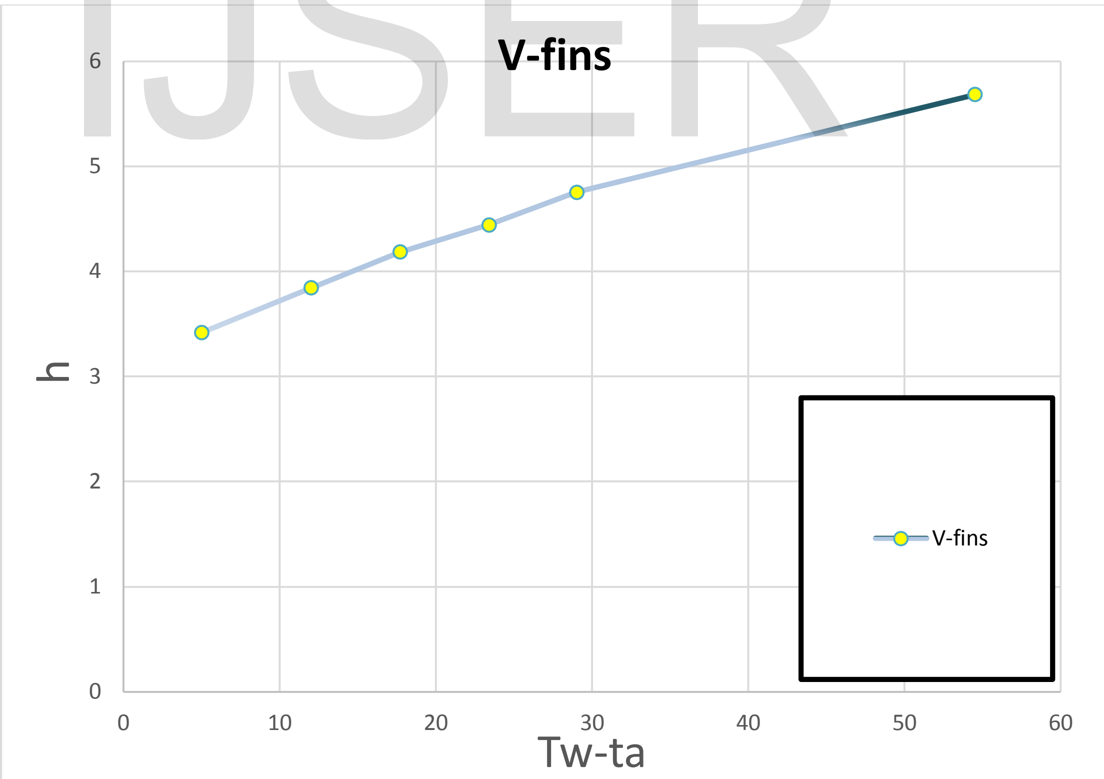


Fig. 5 the variation between heat transfer coefficient and heat input

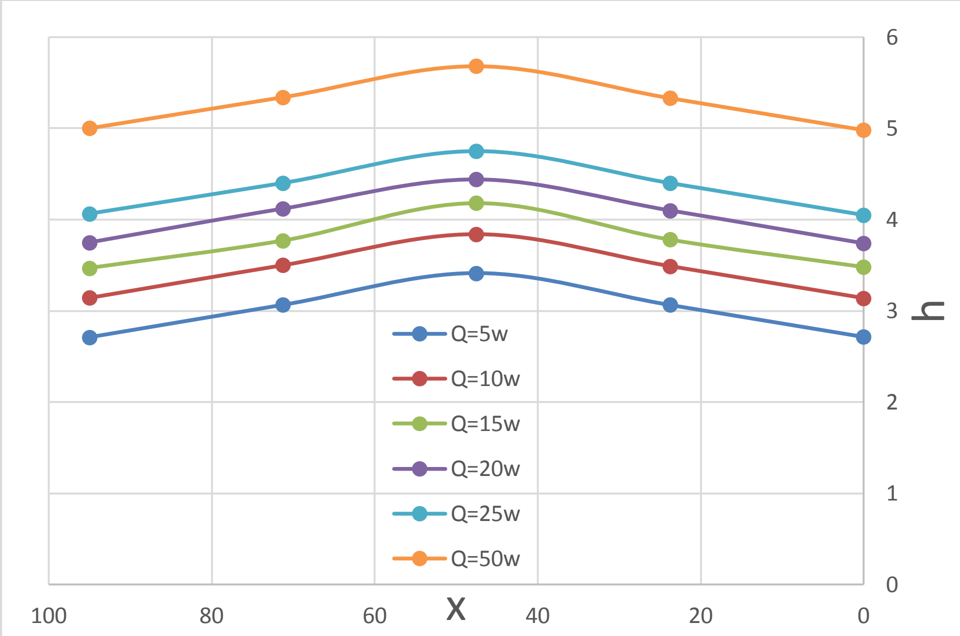


Fig 6 the variation of heat transfer coefficient with length in (mm) for V-fins

Conclusions:

1. For the present research, for the V-fin arrays, it can be concluded that Nu as a function for Ra is: $Nu = 1.1303Ra^{0.2142}$

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%).

3. 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:

[1] Y. Jaluria, Natural convection: Heat and Mass Transfer, 1st edn, Pergamon Press, United States,1980.
[2] A. Bejan, Convection Heat Transfer, 2nd edn, John Wiley & Sons, 1995.
[3] Sarojkumar "CFD analysis of electronic chip cooling" Department of Mechanical Engineering National Institute of Technology Rourkela, 2007.
[4] S. Naik, S. D. Probert and C. I. Wood "Natural-Convection Characteristics of a HorizontallyBased Vertical Rectangular Fin-Array in the Presence of a Shroud" Applied Energy28(1987)pp295-319.

ISSN 2229-5518
[5] C. B. Sobhan, S.P. Venkateshan and K.N. Seetharamu "Experimental analysis of unsteady free convection heat transfer from horizontal fin arrays" Wfirme- und Stoffiibertragung 24, (1989),pp155-160.
[6] C.B. Sobhan, S. P. Venkateshan and K. N. Seetharamu "Experimental studies on steady free convection heat transfer from fins and fin arrays" Wfirme- und Stofffibertragung 25, (1990),pp345-352.
[7] M.J. Sable1, S.J. Jagtap, P.S. Patil, P.R. Baviskar4 and S.B. Barve5 " enhancement of natural convection heat transfer on vertical heated plate by multiple v- fin " November 2010, pp255-262.
[8] Shivdas S. Kharche, Hemant S. Farkade" Heat Transfer Analysis through Fin Array by Using Natural Convection" International Journal of Emerging Technology and Advanced Engineering,April 2012.pp 314-322.
[9] Vinod M. Wankar, 1 S.G. Taji " Experimental Investigation of Flow Pattern on Rectangular Fin Arrays under Natural Convection" International Journal of Modern Engineering Research, 2012 pp4572-4576.
[10] S.G. Taji, G.V. Parishwad and N.K. Sane "Enhanced performance of horizontal rectangular finarray heat sink using assisting mode of mixed convection" International Journal of Heat and MassTransfer 72 (2014),pp 250–259. 201-211.
[11] SenolBaskaya, MecitSivrioglu and Murat Ozek "Parametric study of natural convection heattransfer from horizontal rectangular fin arrays" Gazi University, Ankara, Turkey, 2000.
[12] V. Dharma Rao, S.V. Naidu a and B. GovindaRao, K.V. Sharma "Heat transfer from a horizontalfin array by natural convection and radiation A conjugate analysis "International Journal of Heatand Mass Transfer 49 (2006), pp 3379-3391.