Effect of inclination interrupted fin arrays with isoflux on convection heat transfer

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ABSTRACT

The present study explained heat transfer with convection from horizontal, inclined and vertical fin position for the two cases 1-intrrupted and 4-intrrupted rectangular fin arrays with constant heat flux on the fin base. The numerical analysis was used the free stream function model for the mathematical model and COMSOL 5.0 package was used to find the mesh generation and find the results. Rayleigh number had been used for the range $1*10^7$ to $2.5*10^8$ and temperature difference of 20-28□C.

1.INTRODUCTION

Heat is generated as a by-product in many of the engineering applications, therefore this heat must be removed from the system because of the performance of the systems will decrease. This may cause a resistance to the air flow and boundary layer interference which in return decrease the heat transfer coefficient **[1]**. Heat dissipation of the parallel plates by free convection had been investigated numerically for wide range of Ra, 0.2< Ra <105. It can be determined that in the limit of small gap width, Nusselt number varies proportional to the channel Ra **[2, 3]**. Heat dissipation from the finned systems to the ambient atmosphere can be obtained by using the mechanisms of the convection and radiation heat transfer, The effect of radiation contribution in total heat transfer rate is quite low due to the low emissivity values of the materials used in the fins, such as duralumin and aluminum alloys. The basic equation describing such heat losses is given by the equation:

$$Q_c = h.A.\Delta T \quad (1.1)$$

In this literature , it had been investigated numerically the natural convection from three different plate-fin heat sinks (1-intruppted inclined fins, 4-interrupted fins) horizontal and inclined from the horizontal orientations for the angles $(0°,+30°,+45°,+60°,+90°)$ measured with heat flux values (5,10,15,20,25)watt. The object of the present study to determine the effects of the orientation and the temperature difference between the base and ambient temperatures on the natural convection heat transfer rate for the rectangular 1-intrupted fin heat sinks.

Computational domain

The computational domain for the present problems can be explained in figs.1,2,3

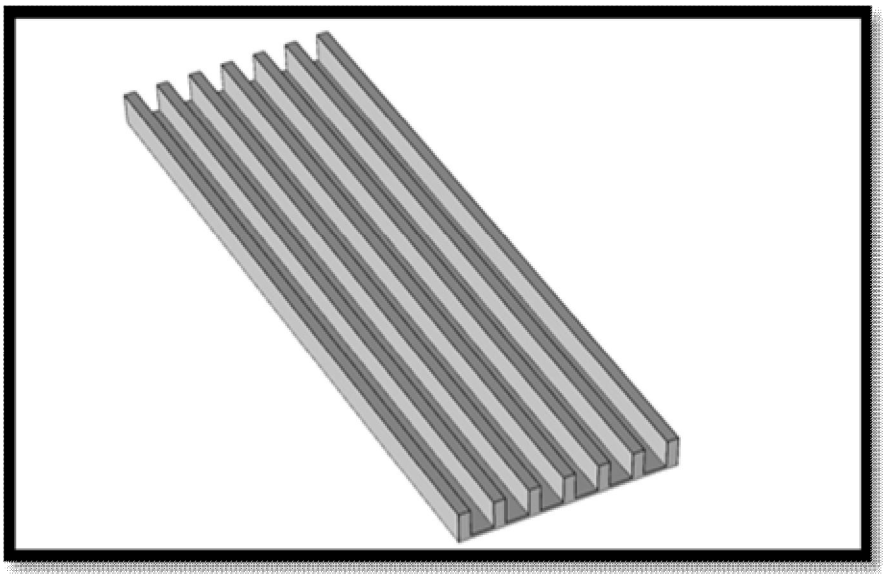


Fig.1 schematic of continuous fin

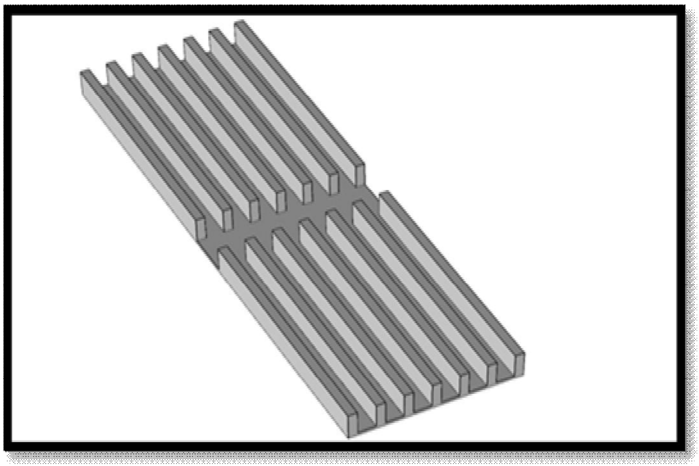


Fig.2 schematic of 1-interrupted fin

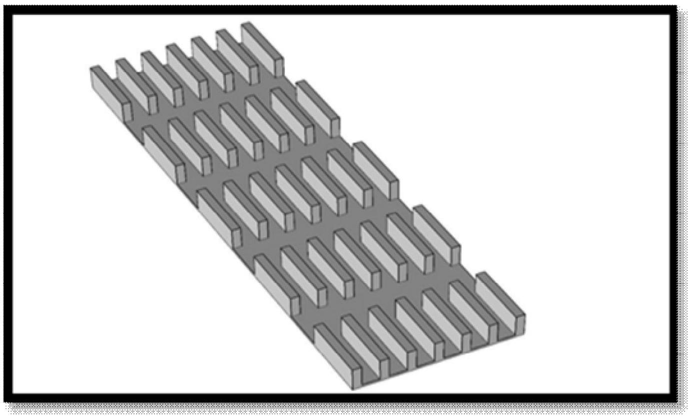


Fig.3 schematic of 4-intruppted fin

Material properties

All the different patterns of heat sinks are made from Aluminum in this study at room temperature of 25°c. Material properties of the heat sinks are shown in Table.1

Table.1 Material properties.

Material	Density ρ [kg/m3]	Heat capacity Cp [J/kg.K]	Thermal conductivity k [W/m.K]	Emissivity ε
Aluminum Alloy 3003-H18 mm	2730	893	155	0.5

Governing Equations

For the present study, the temperature field is obtained by solving the energy equation **[7].**

The heat conduction in solid is governed by;

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

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

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

Radiation heat transfer from all external surfaces to ambient is governed by;

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

Where (ε)is emissivity, and (σ)is Stefan-Boltzmann constant = 1.38E^{-23} [J/K].

$$\rho f\left(\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} - \rho f[1-\beta(T-T_{\infty})]\mathrm{g}\cos\phi + \mu f\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \qquad \ldots 5$$

$$\rho f\left(\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \mu f\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) - \rho f[1-\beta(T-T_{\infty})]\phi \qquad \ldots 6$$

$$\rho w Cpw\frac{\partial Tw}{\partial t} = \left(kw\frac{\partial T}{\partial x} - \frac{p}{Aw}kf\frac{\partial T}{\partial y}\right) \qquad \ldots 7$$

To solve the problem , it can be considered the fins in vertical position and inclined heat sink base. The momentum, energy and mass balance equations for the air medium

The vorticity (ζ), which is defined as follows:

$$\zeta = \frac{\partial \zeta}{\partial t} - \frac{\partial u}{\partial x} \qquad \ldots 8$$

By using mathematical analyses to above equations by compensation equivalent ones then we find:

$$\rho f\left(\frac{\partial \zeta}{\partial t} + u\frac{\partial \zeta}{\partial x} + v\frac{\delta \zeta}{\delta y}\right) = \rho f g\beta\left(\frac{\partial T}{\partial y}\cos\phi - \frac{\partial T}{\partial x}\sin\phi\right) + \mu f\left(\frac{\partial^2 \zeta}{\partial x^2} + \frac{\partial^2 \zeta}{\partial y^2}\right) \qquad \ldots 9$$

$$u = \frac{\partial \psi}{\partial y} \quad , \quad v = -\frac{\partial \psi}{\partial x} \qquad \ldots 10$$

The vorticity equation in terms of stream function as follows:

$$\zeta = \frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} \qquad \ldots 11$$

Stream functions ψ can be evaluate using Eq. 11, if the vorticities ζ are known the velocity components **v** and are to be computed from the values of ψ through eq.10.

Density can be shown to follow a simple inverse relationship (ideal gas) with a small correction term :

$$\rho = \frac{351.99}{T} + \frac{344.84}{T^2}\left[\frac{kg}{m^3}\right] \qquad \ldots 12$$

It can be used the correlate viscosity and thermal conductivity with temperature

$$\mu = \frac{1.4592\, T^{\frac{3}{2}}}{109.1 + T}\left[10^{-6}\frac{(N.s)}{m^2}\right] \qquad \ldots 13$$

$$k = \frac{2.334\times 10^{-3} T^{\frac{3}{2}}}{164.54 + T}\left[\frac{W}{(m.K)}\right] \qquad \ldots 14$$

Specific heat follows a quadratic relationship:

$$Cp = 1030.5 - 0.199975T + 3.9743\times 10^{-4}T^2\left[\frac{J}{(kg.k)}\right] \qquad \ldots 15$$

Mesh grid

A computational quadratic meshes ware used for two types of the studied heat sinks. Independent of the grid size have been examined, the mesh grid can be shown below.

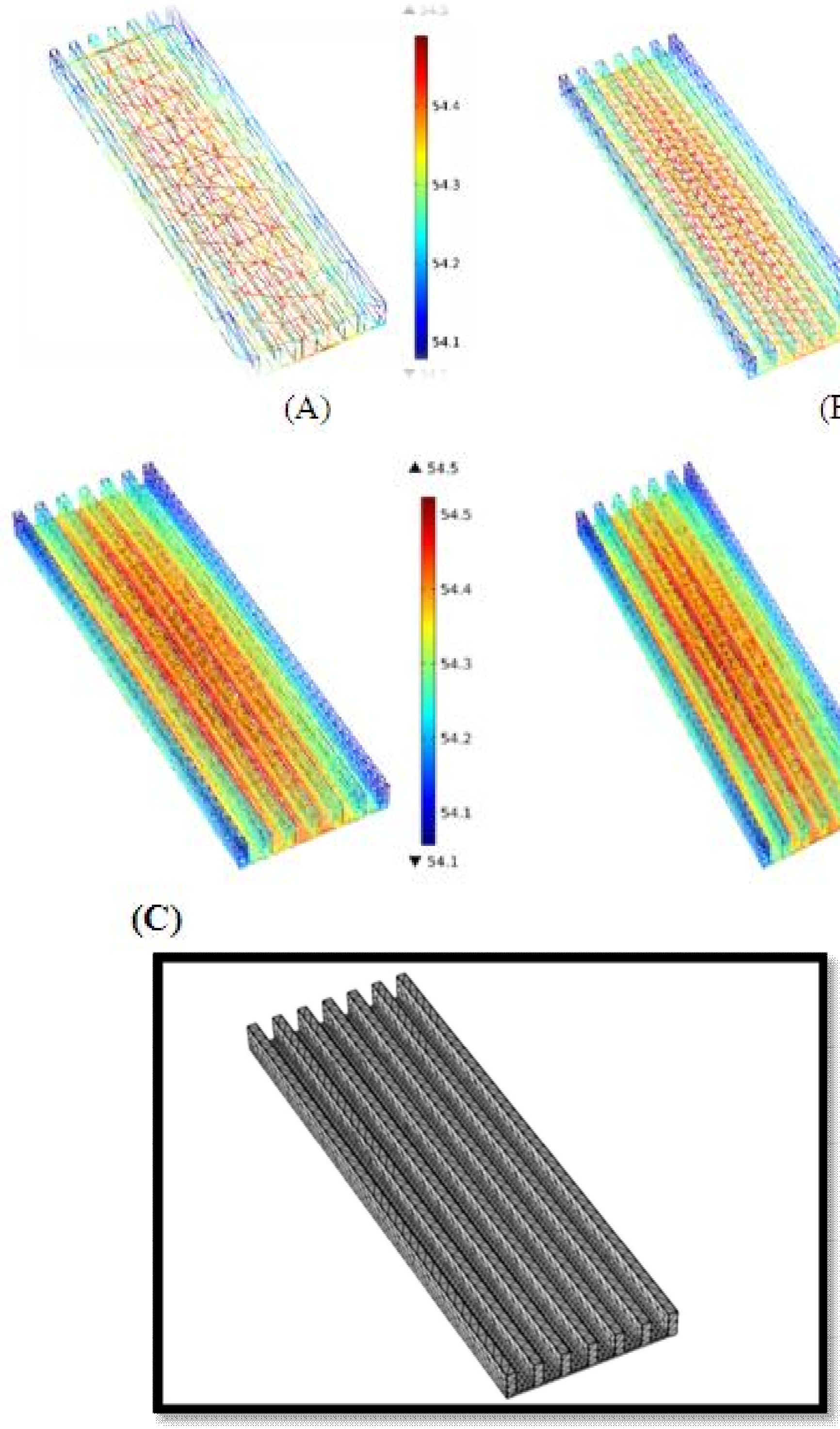


Fig. 4 mesh independence of continuous fin

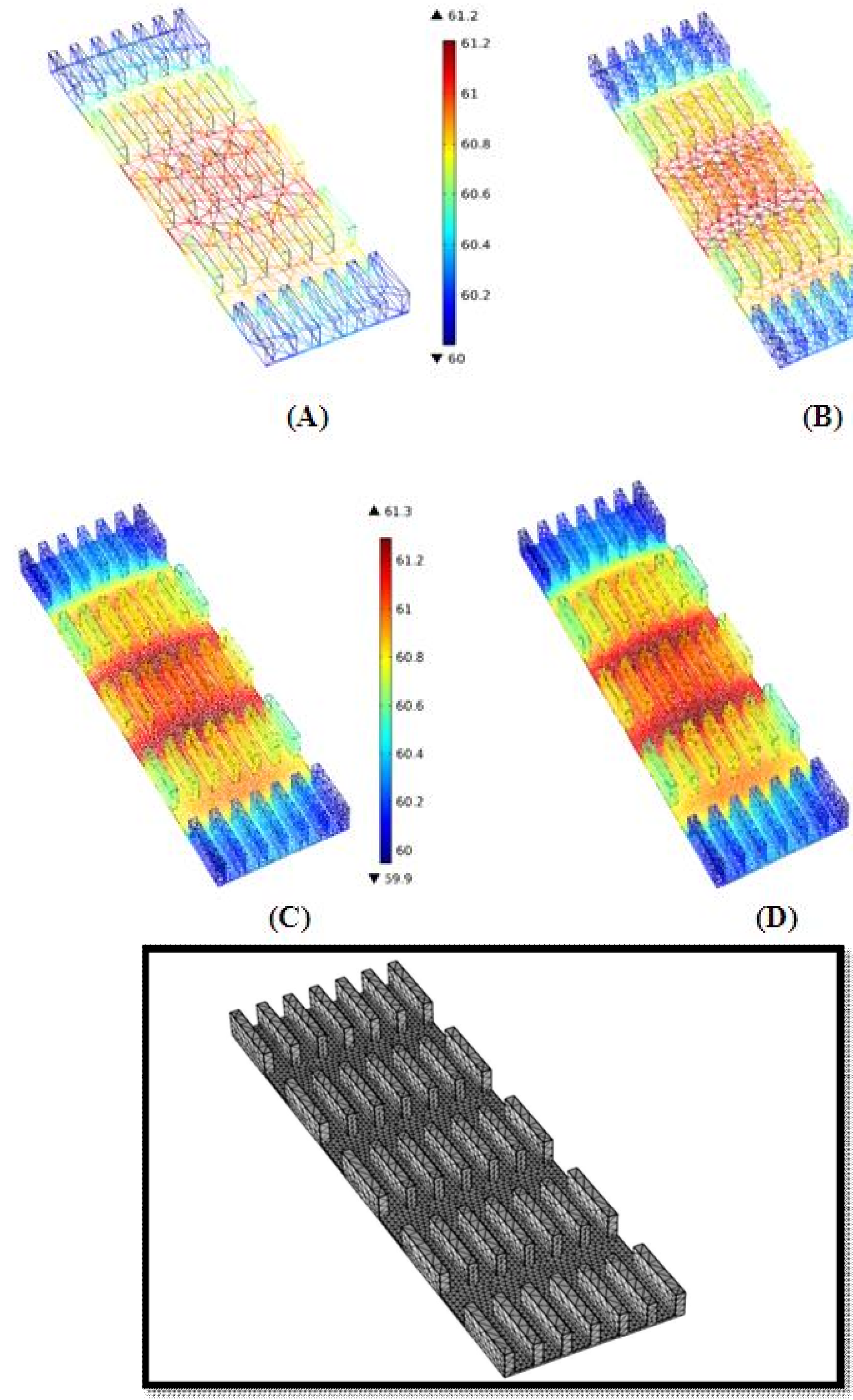


Fig.5 mesh independence of 4-interrupted fin

Table .2 mesh independence

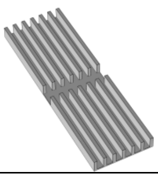
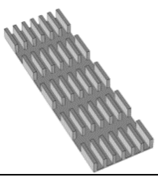
Type of fin	Figure	Mesh element
Continuous fin	A	1360 domain elements-750 boundary elements-361 edge elements
	B	4169 domain elements-2712 boundary elements-752 edge elements
	C	32049 domain elements-18742 boundary elements-1925 edge elements
	D	45345 domain elements-24458 boundary elements-2248 edge elements

	E	Complete mesh consists of 32049 domain elements, 18742 boundary elements, and 1925 edge elements
4-Interrupted Fin	A	1175 domain elements, 860 boundary elements, and 512 edge elements
	B	5170 domain elements, 3782 boundary elements, and 1099 edge elements
	C	30363 domain elements, 18594 boundary elements, and 2136 edge elements
	D	40586 domain elements, 23902 boundary elements, and 2375 edge elements
	E	Complete mesh consists of 30363 domain elements, 18594 boundary elements, and 2136 edge elements

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.

Table .3 Comparison between CFD model and experimental results in Celsius.

Heat sink type	Power = 5 W		Power = 10 W		Power = 15 W		Power = 20 W		Power = 25 W	
	$T_{ext} = 20.2$		$T_{ext} = 19.9$		$T_{ext} = 17.6$		$T_{ext} = 20.6$		$T_{ext} = 24.4$	
	T_{ag} (CFD)	T_{ag} (Exp)	T_{ag} (CFD)	T_{ag} (Exp)	T_{ag} (CFD)	T_{ag} (Exp)	T_{ag} (CFD)	T_{ag} (Exp)	T_{ag} (CFD)	T_{ag} (Exp)
	26.25	26.5	34.6	33.8	36.25	35.6	44.8	43.76	55.9	54.86
	$T_{ext} = 20.2$		$T_{ext} = 20.8$		$T_{ext} = 22.4$		$T_{ext} = 23.6$		$T_{ext} = 24.8$	
	T_{ag} (CFD)	T_{ag} (Exp)	T_{ag} (CFD)	T_{ag} (Exp)	T_{ag} (CFD)	T_{ag} (Exp)	T_{ag} (CFD)	T_{ag} (Exp)	T_{avg} (CFD)	T_{avg} (Exp)
	26.62	25.12	34.07	33.37	44.13	41.6	53.21	52.64	58.24	56.6
	$T_{ext} = 22.5$		$T_{ext} = 23.3$		$T_{ext} = 27.6$		$T_{ext} = 27.3$		$T_{ext} = 26$	
	T_{avg} (CFD)	T_{avg} (Exp)	T_{avg} (CFD)	T_{avg} (Exp)	T_{avg} (CFD)	T_{avg} (Exp)	T_{avg} (CFD)	T_{avg} (Exp)	T_{avg} (CFD)	T_{avg} (Exp)
	30.20	29.31	38.4	36.86	48.82	46.66	54.75	53.26	62.36	61.6

Results and discussion:

The following results show the effects of the fin orientation and the number of fin interrupted on temperature distribution, convection heat transfer numerically. It can be found that the nimber of itruppted had an effect on the temperature distribution at the same angle of orientation as shown in fig.5 , because of the surface area that exposed to the air flow and the temperature difference which is the buoyancy forces because of the temperature difference between the surface temperature and the environment, therefore these parameters will effect on the air density and therefore it will be the temperature distribution for the 4-intruppted is lower than that for the 1-intruppted and continuous fin.

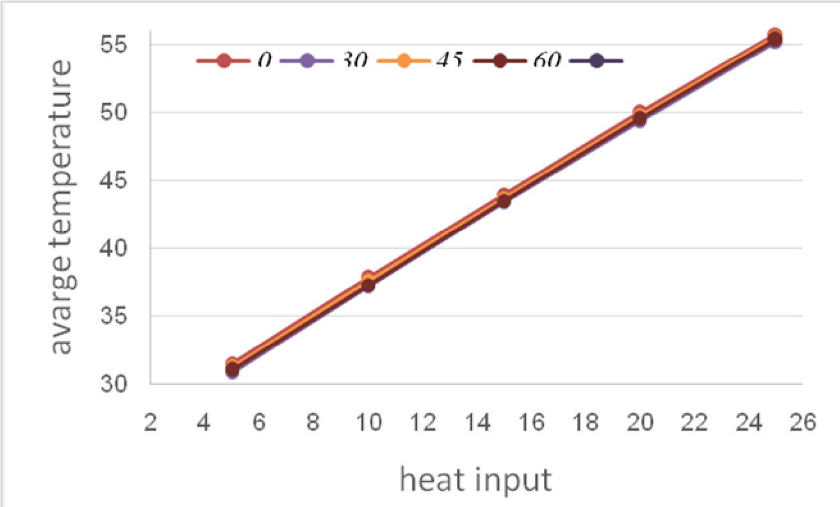


Fig.5 effect of the heat flux and the fin orientation on temperature distribution for 1-intruppted fin.

Fig.5 shows for the 1-intruppted rectangular fin , the effect of the heat input on the temperature distribution for horizontal, inclined and vertical orientation, which explained the temperature for horizontal is higher than that for the other positions because of the buoyancy force for the inclined and vertical position than that for the horizontal therefore the convection heat transfer will be smaller than that for the other.

The following equations had been found from the numerical analysis for 1- and 4- interrupted rectangular fin, and it could be concluded from these equations the effect of interrupted number on the heat transfer for each angle of inclination, among them.

Angle of inclination	Type of the fin	*Concluded equation*
0°	1-intruppted	$Nu = 5E\text{-}11Ra^{1.4887}$ $R^2 = 0.9394$
	4-intruppted	$Nu = 3.8E\text{-}11Ra^{1.652}$ $R^2 = 0.9855$
30°	1-intruppted	$Nu = 1E\text{-}06Ra^{0.9462}$ $R^2 = 0.9559$
	4-intruppted	$Nu = 3E\text{-}08Ra^{1.1435}$ $R^2 = 0.9312$
°45	1-intruppted	$Nu = 1E\text{-}07Ra^{1.0944}$ $R^2 = 0.9506$
	4-intruppted	$Nu = 8E\text{-}10Ra^{1.3425}$ $R^2 = 0.9379$
°60	1-intruppted	$Nu = 4E\text{-}07Ra^{1.0195}$ $R^2 = 0.9516$
	4-intruppted	$Nu = 9E\text{-}09Ra^{1.2119}$ $R^2 = 0.9377$
°90	1-intruppted	$Nu = 5E\text{-}06Ra^{0.8766}$ $R^2 = 0.9523$
	4-intruppted	$Nu = 3E\text{-}07Ra^{1.0269}$ $R^2 = 0.9268$

Conclusions

The results for the two types of the rectangular fin with interrupted explained some conclusions can be shown in the following:

1. Heat transfer by convection for the horizontal is smaller than that for the other because of the buoyancy forces.
2. Convection heat transfer from 4-itrrupted fin higher than that for 1-intrrupted fin because of the surface area which exposed to the working fluid which is air.

References

[1]. Avdhoot Walunj, “ PARAMETRIC ANALYSIS OF PLATE-FIN HEAT SINK OVER HEAT TRANSFER”, International Journal of Research, Volume 1, Issue 4, (2013).
[2]. Ilker Tari , Natural convection heat transfer from horizontal and slightly inclined plate-fin heat sinks , Applied Thermal Engineering 61 (2013) 728e736