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Effect of the fins configuration on natural convection heat transfer experimentally and numerically

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Abstract

The cooling of the electronic systems, electrical systems, and CPU of the computer is very important; therefore this study is prepared to improve this aims. In this study, natural convection heat transfer from rectangular fins with five different figures (continuous fins, 1-interrupted fins, 4-interrupted fins, inclined fins and V-fins) are investigated at different heat flux values (175, 350, 525, 700 and 875 Watt per square meter).The effect of base to ambient temperature difference for continuous fins, 1-interrupted fins, 4-interrupted fins, inclined fins and V-fins were determined. All types of the fins are made with different geometries by using CNC machine and wire cut machine, but it have some dimensions in common such as fins thickness (5)mm, fins height (18)mm, space between the fins (10)mm, and the volume of the base plat of heat sink (300*95*2)mm. The heat sink base plate was heated by an attached maximum electric heater 2225 W/m^2 with an identical size with the base plate of the heat sink, which could supply a specific heat flux. The steady-state temperature of the base plate was measured by eleven copper-constantan (K-type) thermocouples inserted into different grooves in the base plate and glued with thermal tape and epoxy to ensure good thermal contact. The mathematical model of the base plate and fins are solved numerically using COMSOL (5.0) after describing the mesh model using the COMSOL (5.0) and assuming the properties of air variation with film temperature. After finding the numerical result, the validation between experimental and numerical results has been verified. Good agreement has been found between the experimental and CFD results. Empirical correlations for the overall Nusselt number versus average Rayleigh number for these configurations are obtained and compared to other correlations sited in the literature. The range of Rayleigh number, Nusselt number and base plate temperature are, ($1.7 *10^7$ - $12.5*10^7$), (37 – 83) and (25.6 – 81.34) respectively.

Keywords: Fins; Heat transfer; Natural convection; Heat sink, CFD.

1. 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 fluid [1]. If the fluid motion happens due to density difference caused by temperature variation in the fluid, then it is called natural convection [2]. The convective heat transfer rate depends on the properties of the fluid flow [3]. Experimental works on natural convection heat transfer were: S. naikel and C.I. Wood [4] Focused about natural convective cooling at steady state of horizontally based, vertical rectangular fins. The results are of significance for the designers of electronic arrays, the components of which should be maintained at temperatures less than 65°C.Sobhan et al [5] analyze experimentally the unsteady on horizontal fins arrays. They studied local variations of temperature and heat flux in the central. Venkateshan et al [6] they were dealing with an experimental study of natural convective heat transfer from fins and fins arrays attached to a heated horizontal base where the fins made from aluminum. The heater was consist of chrome wire wound around a thin mica plate and insulated on both sides using mica sheets. M.J. Sable et al. [7] enhanced a technique of natural convection heat transfer by using heat sink with v- fins type. The dimensions of base plates used for the experiment are (3mm – thick, 250mm – height and 250mm as width). They used the electrical heater as a two plates one above other form to ensure maximum uniform surface temperature of the test plates. The heater was supplied with stabilized current through dimmer state and wattmeter. The ranges of heat input are (25, 50, 75, and 100) watt. Shivdas S. karche et al [8]designed heat sink with rectangular fins and showed influence notch on fins. The fins made by aluminum material with dimensions (length 127mm, height 38mm and thickness 1mm and the spacing between the fins is 9mm). Eight thermocouples are used for attaching the fins and plate for temperature measurement. Vandom wanker and S.G. Taji [9]investigated experimentally the flow pattern on natural convection heat transfer from rectangular fins. The heat sink in horizontal position made from aluminum with dimension of fins (200mm as length, 2mm as thickness and 40 mm as height). Inside an enclosure formed as a cubical with a volume of ($1m^3$), Base heat transfer coefficient values increase with optimum spacing and again decreases. S.G. Taji et al [10] used experimental work of heated rectangular fins in horizontal position under natural convection heat transfer. They focused about two cases of flow patterns named, single chimney flow pattern where spacing about (8-10mm), sliding chimney flow pattern where the spacing is less than 6mm.

In other hand, the numerical studies were: Senolbaskaya et al [11] focused on effect of parameter (length, width, height, spacing and temperature) on heat sink in natural convection heat transfer. Fins made from aluminum in rectangular shape. Heat sink in horizontal position. Case solved by (CFD) program and concluded that the overall heat transfer was enhanced with increase in height of the fins and decrease in length.V. Dharma Raoet al [12] studied natural convection heat transfer from a horizontal heat sink, fins made from aluminum. The theoretical model formulated by treating the adjacent internal fins enclosures. Studying convection and conduction heat transfer analysis was carried off three governor equations (conservation of mass, conservation of momentum and conservation of energy). Nakhi and Ali [13] 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 fins. Fins material was aluminum. A numerical solution based on the finite-volume method is obtained. Representative results illustrating the effects of the thin fins inclination angle, length, space, and the thermal conductivity of the thick surfaces on the streamlines and temperature contours. Aularasan R. and veraj [14] designed modern heat sink to cool electronic device. In numerical work, (CFD) program used to determine the natural convection from rectangular fins. The fins made from aluminum with dimensions (200mm as length, 95mm as width,30mm as height, 4mm as thickness and 2mm as space). M. Baris and mahmetarik [15] Studied the effect of many materials (copper, aluminum, parotic graphite thermal annealed, carbon foam and ppc material) on fins efficiency where is respect to a main factor to make electronic devicein horizontal position. In numerical work, (CFD)program used to compute heat dissipation from heat sink. The boundary condition supplied was heat flux on base plate and ambient temperature (22°C). Yaclin et al [16] studied the natural convection heat transfer from a fins array in horizontal position. The aim of their study was investigating the effects of clearance parameters on the steady-state heat transfer. In order to solve the three-dimensional elliptic governing equations, a finite volume based CFD code was used. The range of Rayligh number ($2*10^4$ - $3.5*10^{7)}$.Ali Al- Qusamy [17]studied numerical steady of natural convection heat transfer from rectangular fins. Fins made from aluminum with dimensions (600mm as length, 25mm as thickness, 100mm as height and 20mm as spacing between fins. Heat sink in horizontal position. The goal of steady was to estimate the maximum heat transportation to ambient use ansyes fluent to solve the model, the range of Rayligh number ($4*10^7$- $2*10^{8)}$, range of height (0.1-0.5)m. Meshing in gambit program with interval count of mesh (25916146) nods. Abdullah H and M. AL-Essa [18] focused on natural convection from rectangular fins in horizontal position. The fins made from aluminum material. The aim of the project was to investigate optimum heat removed from heat sink and made comparison between heat sink with square perforation and solid heat sink and determined the effect of dimensions of fins and number of perforation on heat transfer.

Muhsen Torabi [19] investigated the performance of many figures of fins (longitudinal rectangular fins, trapezoidal fins and concave parabolic profiles fins). His study proved the characteristics of convection heat transfer change with variation of thermal conductivity, heat transfer coefficient and surface emissivity with temperature. The convection and radiation sink temperatures were assumed to be non-zero. The calculations are carried out using the differential transformation method (DTM). Ilker Tari and mehdi [20] made comparison between horizontal and inclined heat sink with rectangular fins for natural convection heat transfer. Fins made from aluminum with dimensions (250mm as length 100 mm as width, 25mm as height, 8 mm as space). R. Samet al[21] studied the natural convection from rectangular interrupted fins in horizontal position. Where the fins made from aluminum. The aim of project was to make best heat sink for high efficiency cooling of electronic chip. For an optimal heat sink design, initial studies on the fluid flow and heat transfer characteristics of standard continuous heat sinks of different designs have been carried through (CFD) simulations. In numerical work, the assumption was fully developed heat and fluid flow, no slip and steady state flow. The range of input power (5-25) w and the range of Rayliegh number (10^4 - 10^7). From results it was found that the holes for the interrupted fins has better performance than interrupted rectangular fins of heat sinks. Salilaranjan and Panda [22] used numerical model and (CFD)program in this study. The aim was to reduce the material used and make heat sink with low cost and high efficiency. The shape of the fins was rectangular fins made from aluminum in horizontal position for natural convection. Dimensions of model were (150 mm as length,100 mm as width, 75mm as height and 8mm as spacing between fins). The range of heat input (50-200)W. The range of Rayliegh number ($5.6*10^6$-$7.6*10^6$). Gambit used to mesh, where mesh size (0.002). From results it was found that when there is an increase in heat input the heat transfer coefficient increases also and nusselt number increases. When there is an increase in fins spacing at a constant heat flux, Rayliegh number increase. Shwin Gung wong and Guei Jang Hung [23] Focused on parametric study on the dynamic natural convection from horizontal rectangular fins arrays (L = 128, 254 and 380 mm) was made using a (3- D) unsteady numerical analysis. Fins made from aluminum. The range of height was (6.4 – 38) mm and the range of spacing was (6.4 - 20) mm.

2. The test rig

An experimental test rig has been designed and constructed to study the investigation of the effect of natural convection heat transfer from a horizontal heat sink. Five types of fins are made from aluminum. The types are continuous fins, 1- interrupted fins, 4-interrupted fins, inclined fins and v-fins as shown in Figure 1. The goal of manufacturing test rig is to examine the natural convection of heat transfer from a heat sink with rectangular fins at different heat fluxes. Another objective is to correlate the experimental data of overall Nusselt number for such configurations.

3. Heating system

In experimental study, the heat sink base plate was heated by an attached electric heater with an identical size as the base plate (300mm), which could supply a specified heat flux as shown in Figure 2. The dimensions of the electrical heater placed on the firebrick 40mm thickness were 340mm * 130mm and insulated from bottom to ensure prevent losses by down conduction heat transfer. The heater had a power output of 264 W at 220 V and a current of 1.2 A. Eleven thermocouples have been used on the base plate to measure the base plate temperature as shown in Figure 3.

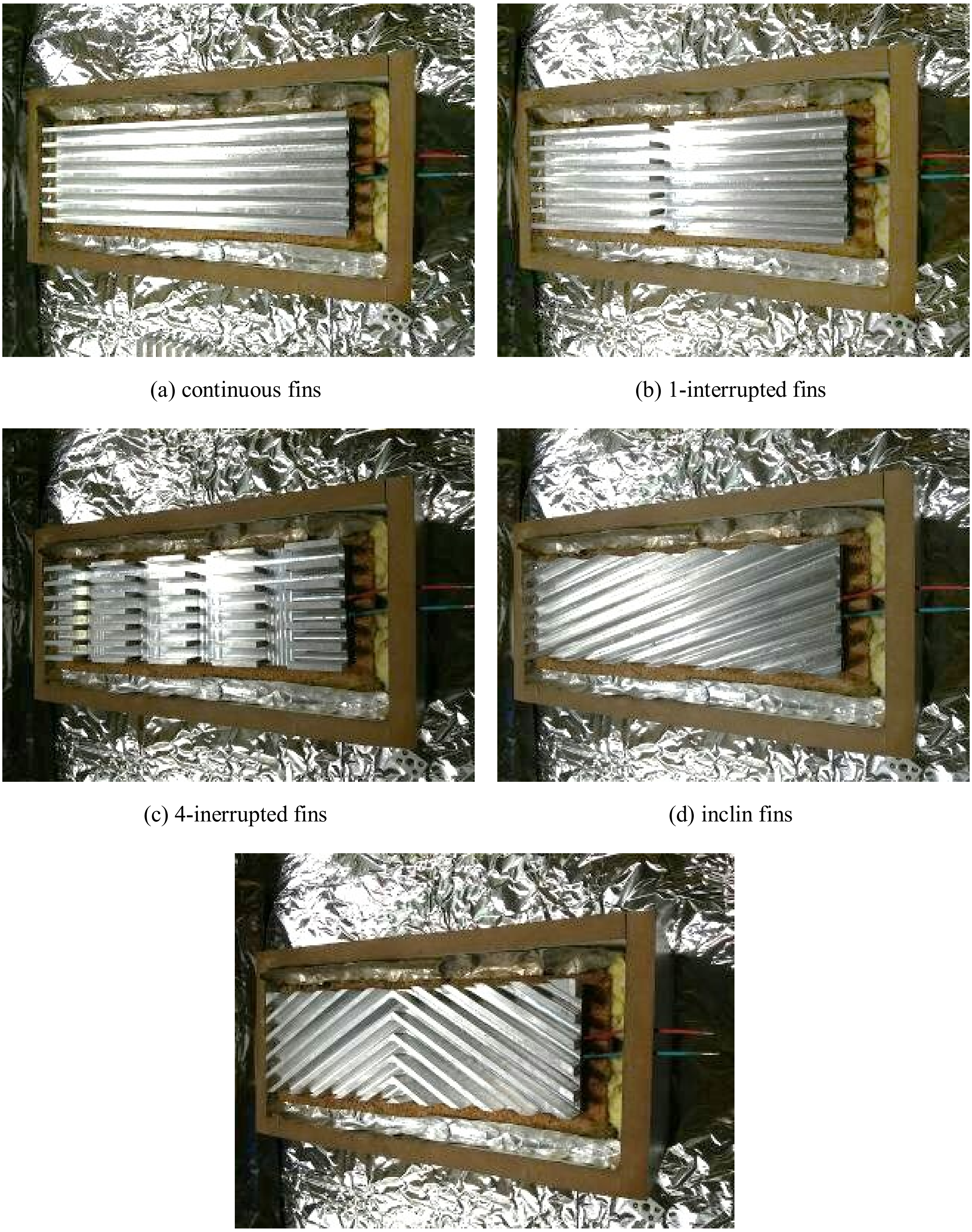
4. Configuration of the whole assembly

After making rectangular fins, the bottom surfaces of the base plate as well as the heater block were insulated by two layers of insulation, 40mm firebrick (k = 1.4 W/m.K) and 100mm glass wool blanket (k = 0.04 W/m.K). The whole assembly, base, heater with associated thermal insulation, was located in a well fitted open-topped wooden box of 20mm thick (k =0.08 W/m.K). The upper edges of the wooden box and the top surface of the laterally placed in thermal insulation were flushed with the upper surface of the heat sink base in which the rectangular fins protruded perpendicularly. The steady-state temperature of the base plate was measured by eleven copper-constantan (K-type) thermocouples inserted into grooves in the base plate and glued with thermal tape and epoxy to ensure good thermal contact (Figure 4).

5. Numerical analysis

On the following subsections, the governing equations, boundary conditions, numerical domain and the corresponding, the assumptions and the mesh independency are discuses. Validation between numerical and experimental results is discuses also.

5.1 Computational domain
Different patterns of heat sinks have been modeled. The heat sinks geometries are shown in Figure 5.



(a) continuous fins

(b) 1-interrupted fins

(c) 4-inerrupted fins

(d) inclin fins

(e) v- fins

Figure 1. Type of fins used in this study

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Figure 2. The heating system

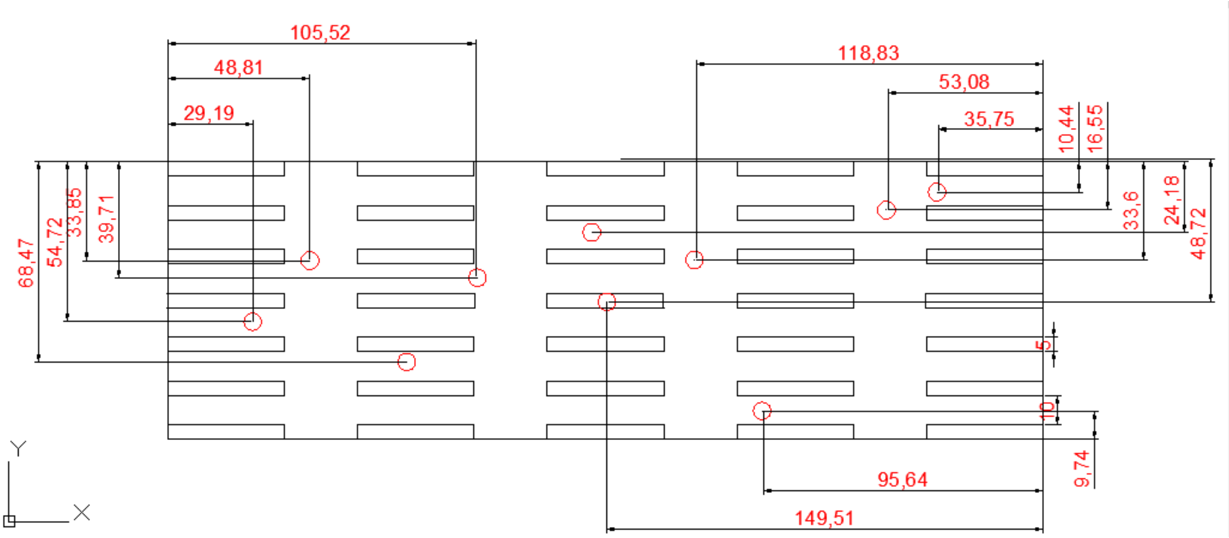


Figure 3. Distribution of thermocouples

5.2 Governing equations

The heat transfer in the heat sinks is take place in three ways, conduction, convection, and radiation. The temperature field is obtained by solving the energy equation [10, 24].

The heat conduction in solid is governed by;

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

where ρ is density [kg/m^3], C_p is heat capacity [J/kg.K], k is thermal conductivity [W/m.K], T is temperature, Q is heat source [W], and t is time [s].

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

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

where n is unit vector normal to the surface, h is convection coefficient [W/m^2.K], and T_{amb} is ambient temperature [K].

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

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

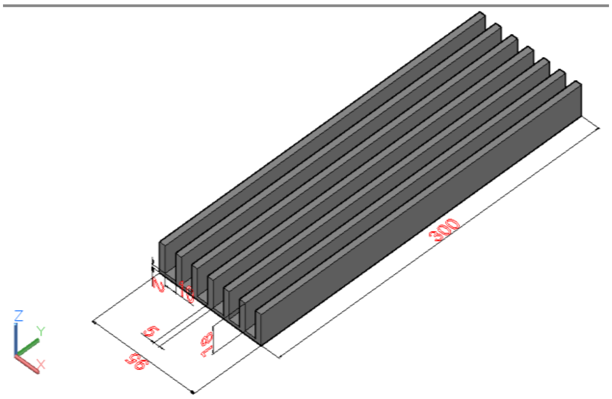
where (ε)is emissivity, and (σ)is Stefan-Boltzmann constant = 1.38E^{-23} [J/K].

(a) (b)

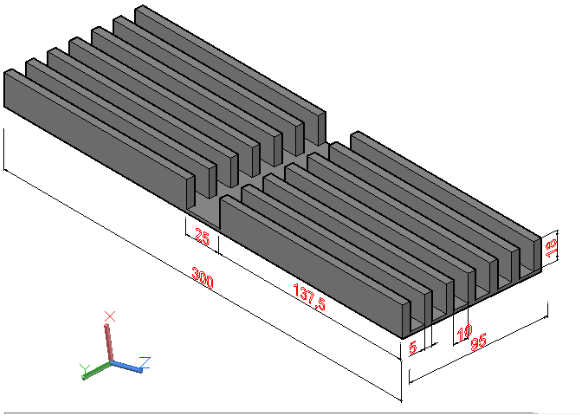
(c) (d)

(e) (f)

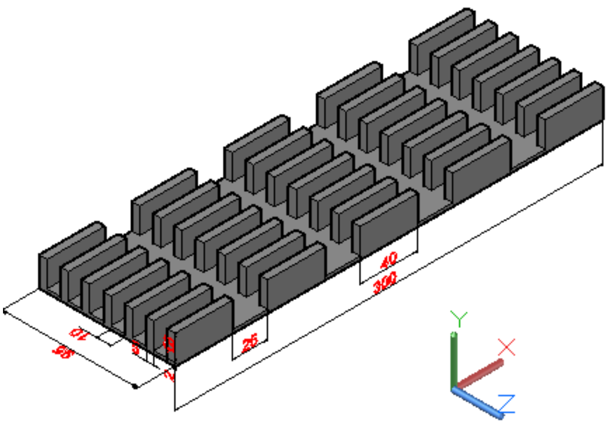
Figure 4. Configuration of the whole assembly



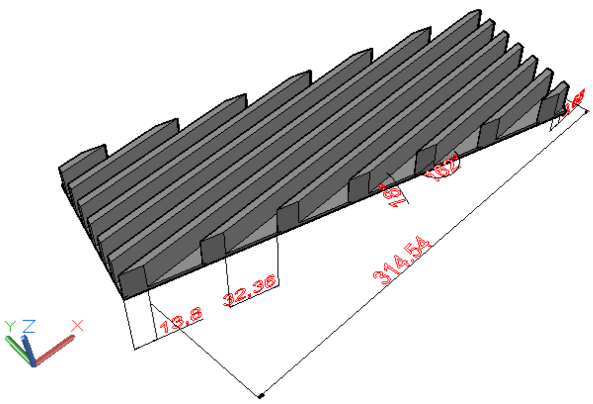
(a) continuous



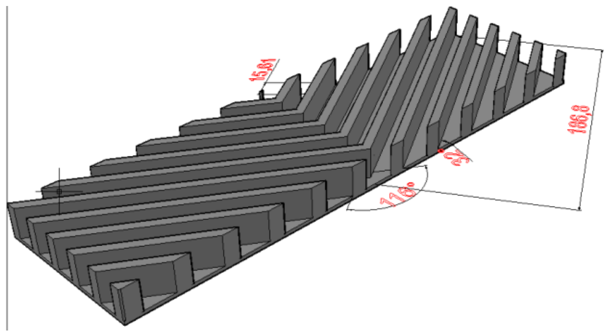
(b) 1-interrupted fins



(c) 4-inerrupted fins



(d) inclin fins



(e) v- fins

Figure 5. Fins model (domains)

5.3 Boundary conditions

Boundary layer mesh was used for regions that are closer to the fins surface, in order to capture the flow behavior with a higher resolution. The boundary condition of model is the base plate of heat sink held at a constant isothermal heat flux, lead to heat transfer to fins by conduction and fins play main rule for thermal radiation and convection transfer to environmental at temperature of ambient, where the emissivity of aluminum (0.5) has been used (Figure 6).

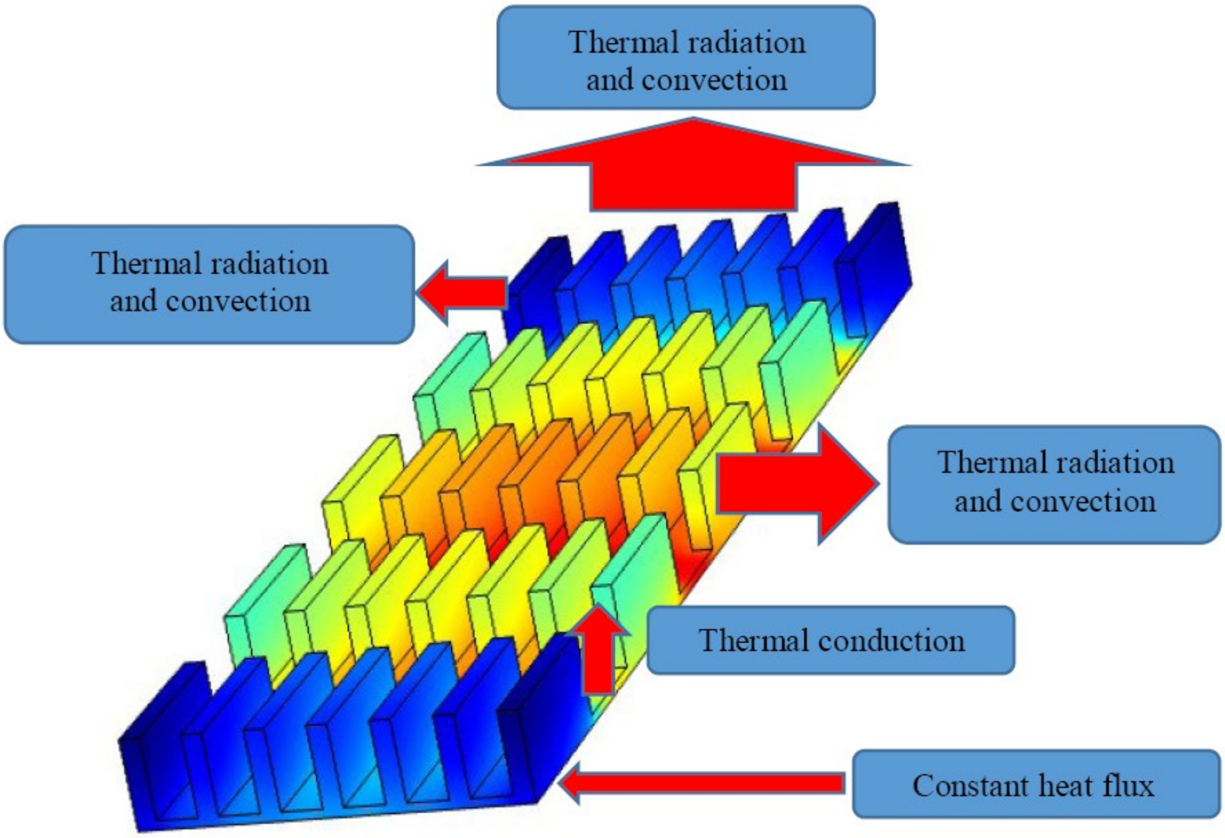
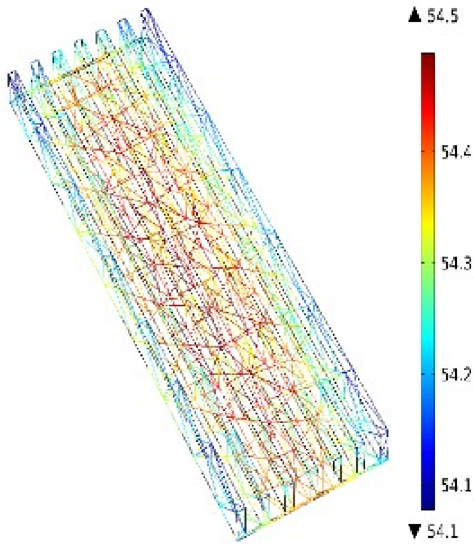


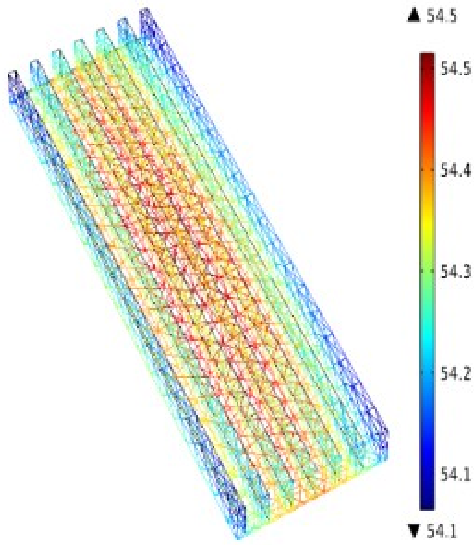
Figure 6. Boundary condition

5.4 Computational grid

The governing equations were discretized using COMSOL(5.0) as a computational fluid dynamic (CFD) [25]. 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 heat flux and (25°c) of an environmental temperature. Where the mesh independence, for Figure 7 (Complete mesh consists of 32049 domain elements, 18742 boundary elements, and 1925 edge elements), for Figure 8 (Complete mesh consists of 31607 domain elements, 18540 boundary elements, and 1962 edge elements), for Figure 9 (Complete mesh consists of 30363 domain elements, 18594 boundary elements, and 2136 edge elements), for Figure10 (Complete mesh consists of 36248 domain elements, 20810 boundary elements, and 2507 edge elements), for Figure 11(Complete mesh consists of 36294 domain elements, 21206 boundary elements, and 2622 edge elements). Table 1 shows the mesh independence for all types of fins.



(a)



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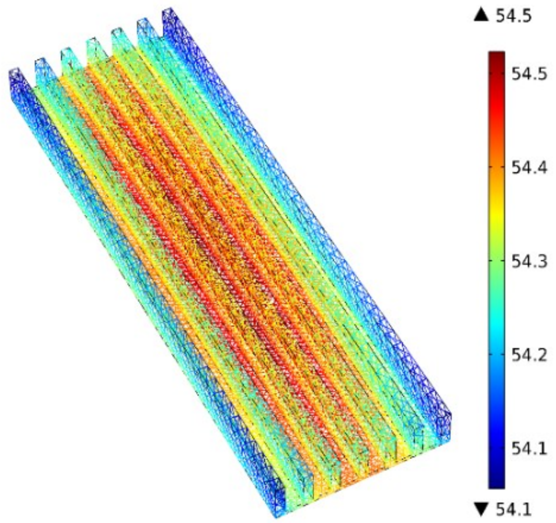

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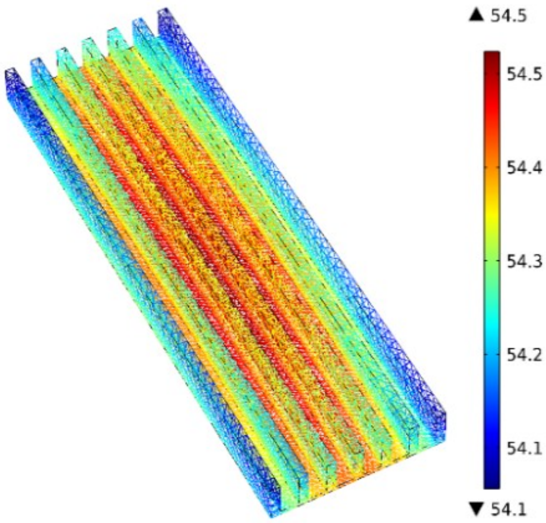

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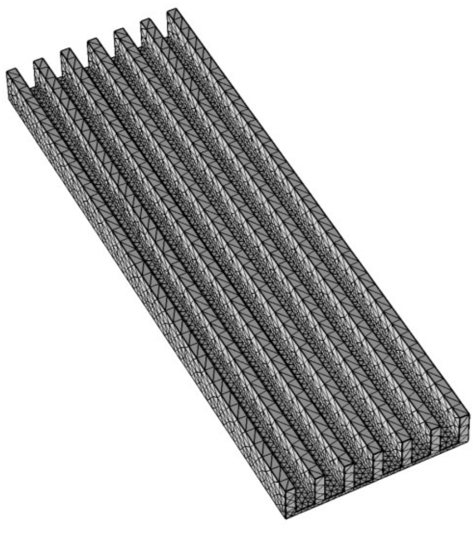
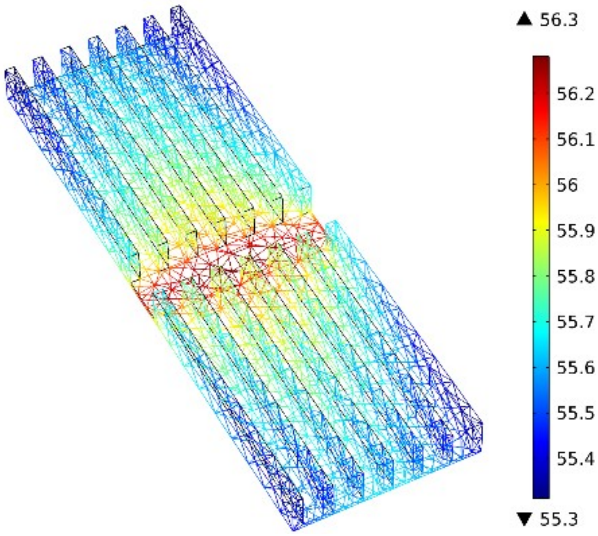
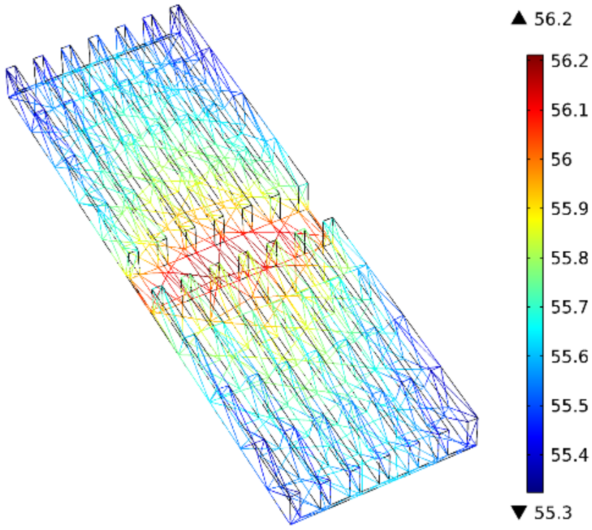


(e)

Figure 7. Mesh independence of continuous fins



(a)



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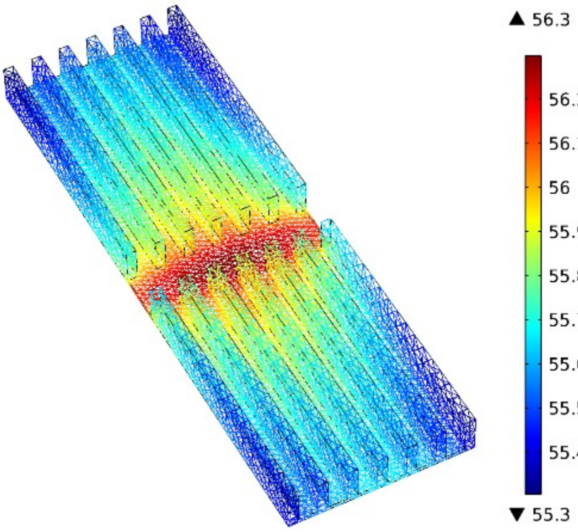

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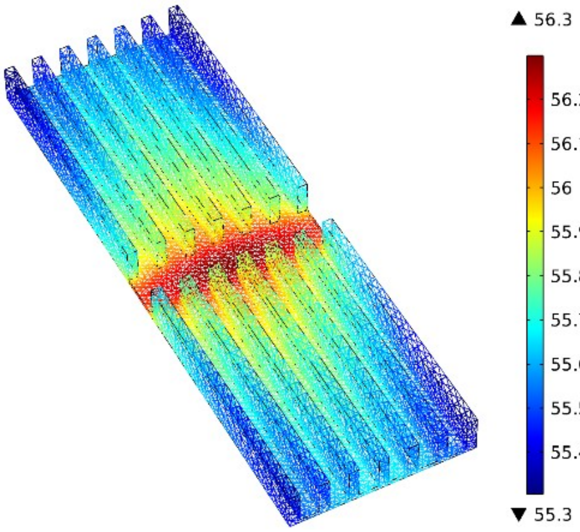

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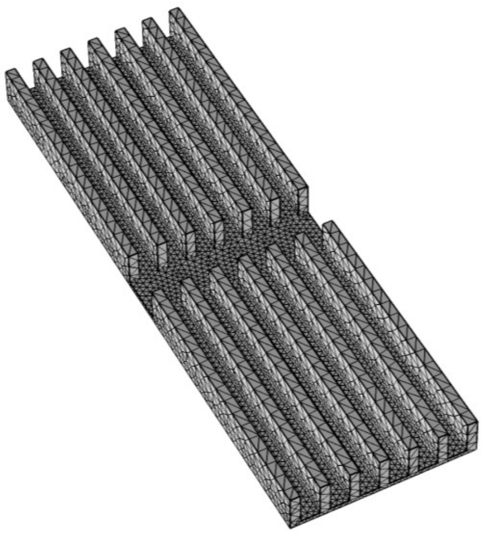


(e)

Figure 8. Mesh independence of 1- interrupted fins

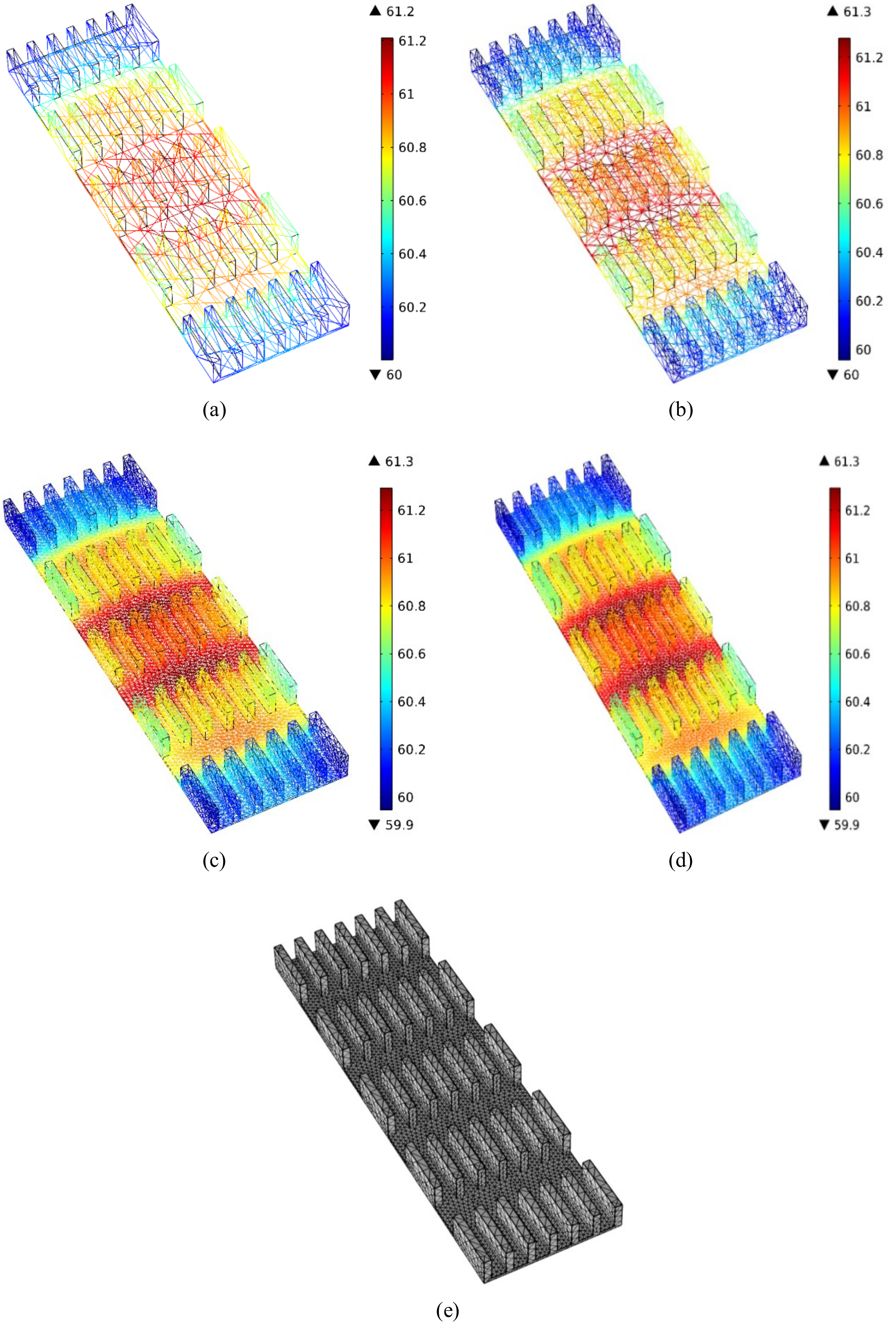


Figure 9. Mesh independence of 4-interrupted fins

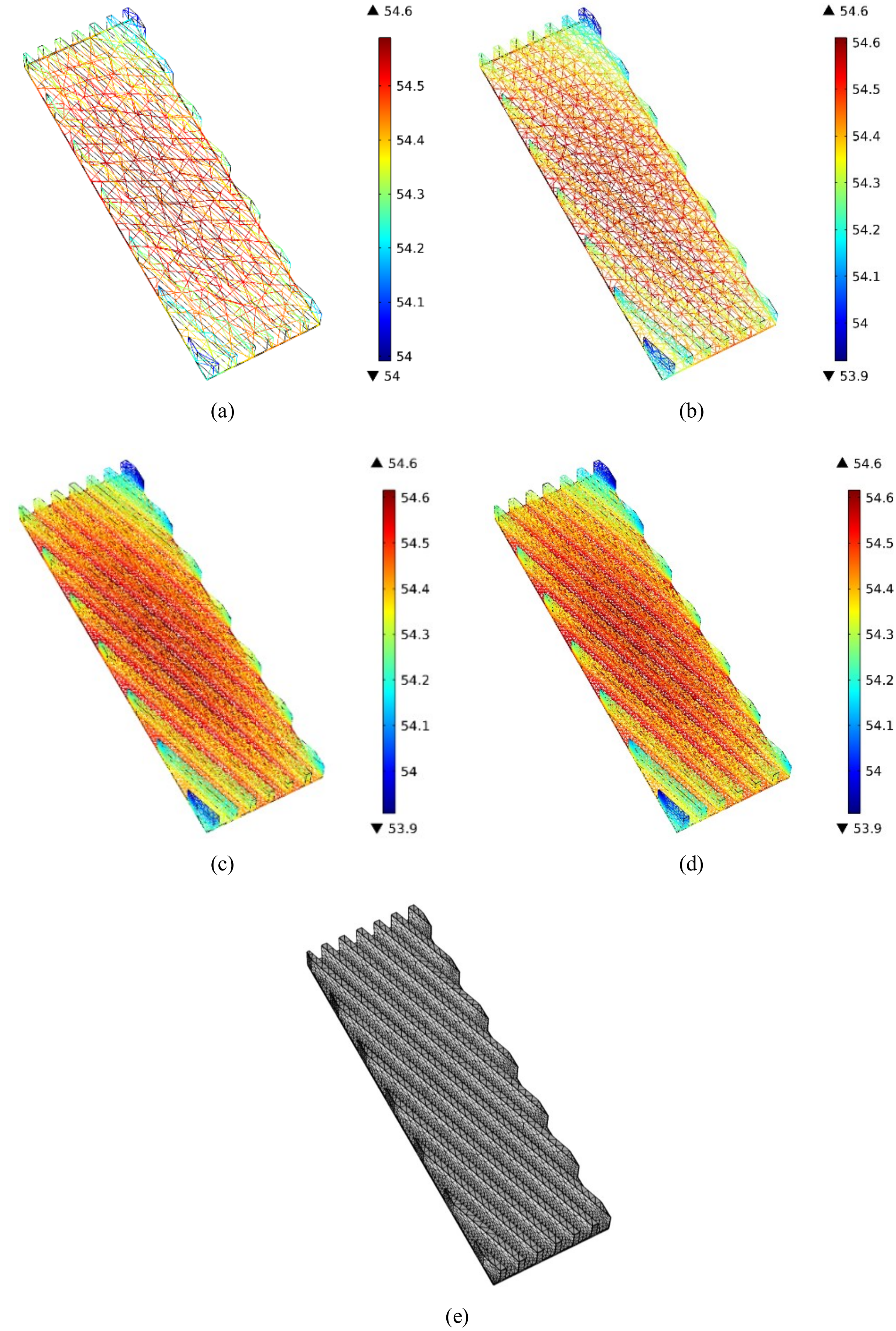


Figure 10. Mesh independence of incline fins

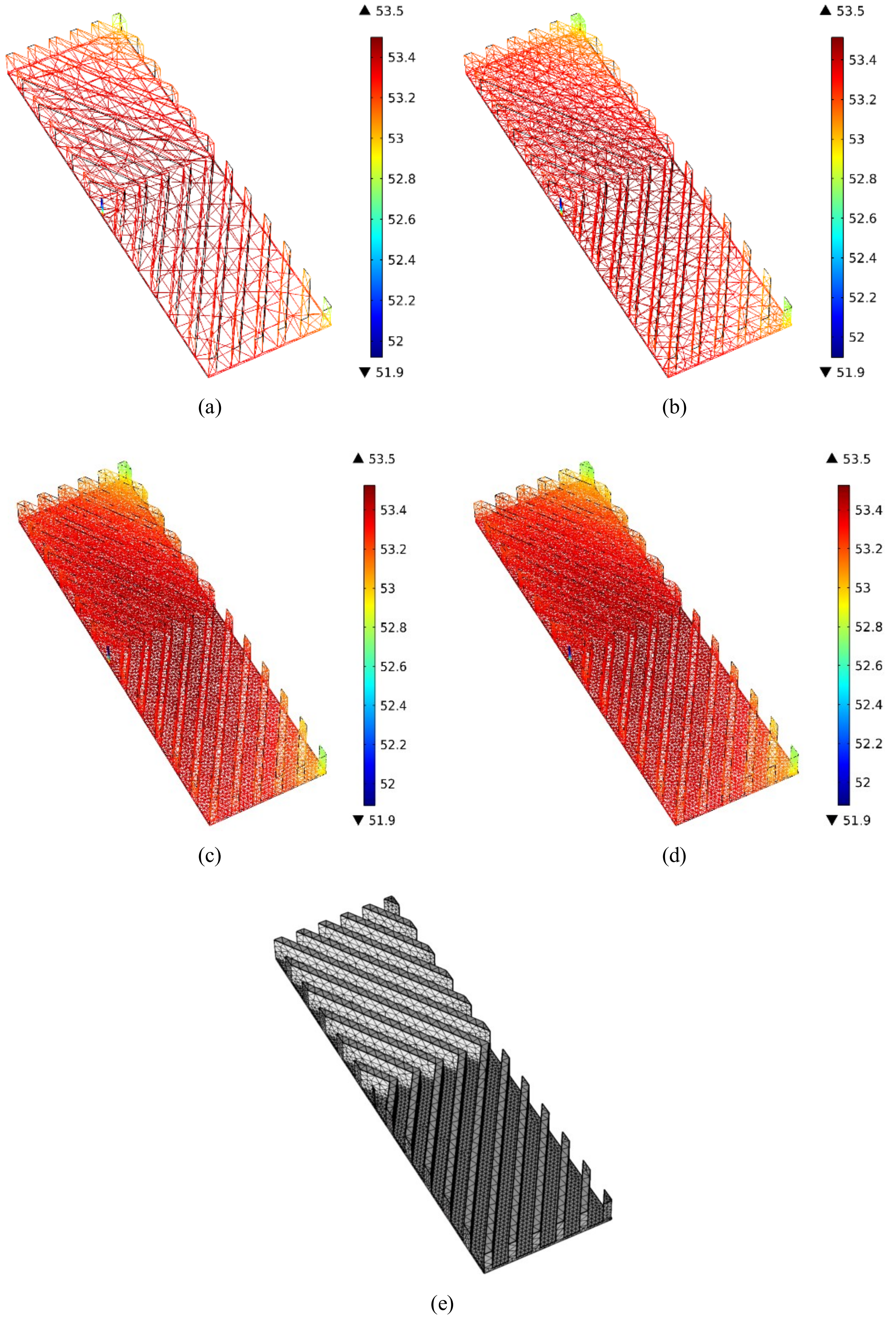


Figure 11. Mesh independence of v-fins

Table 1. Mesh independence

Type of fins	Figure	Mesh element
Continuous fins	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
1-Interrupted fins	a	1331 domain elements, 844 boundary elements, and 425 edge elements
	b	4091 domain elements, 2790 boundary elements, and 829 edge elements
	c	31607 domain elements, 18540 boundary elements, and 1962 edge elements
	d	42903 domain elements, 23572 boundary elements, and 2272 edge elements
	e	Complete mesh consists of 31607 domain elements, 18540 boundary elements, and 1962 edge elements.
4-Interrupted fins	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
Inclined fins	a	1566 domain elements, 982 boundary elements, and 473 edge elements
	b	5058 domain elements, 3432 boundary elements, and 1028 edge elements
	c	36248 domain elements, 20810 boundary elements, and 2507 edge element
	d	40648 domain elements, 23444 boundary elements, and 2717 edge elements
	e	Complete mesh consists of 40648 domain elements, 23444 boundary elements, and 2717 edge elements
v-fins	a	1592 domain elements, 1038 boundary elements, and 529 edge elements
	b	4636 domain elements, 3228 boundary elements, and 1026 edge elements
	c	36294 domain elements, 21206 boundary elements, and 2622 edge elements
	d	40069 domain elements, 22718 boundary elements, and 2740 edge elements
	e	Complete mesh consists of 36294 domain elements, 21206 boundary elements, and 2622 edge elements

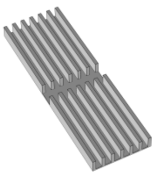
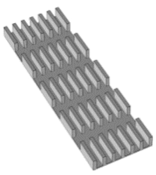
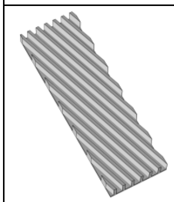
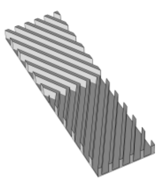
6. Validation

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 2).

7. Result and discussion

Figure 12 indicates the variation between temperature different (base plate temperature minus from ambient temperature) in Celsius and heat input in watt for five models of fins with deferent geometry (continuous fins, 1-inerrupted fins, 4-interrupted fins, inclined fins and V-fins). As a rule when increasing the heat input, that will increase(ΔT) because of the increasing in the convection and radiation heat transfer. As well as, note (ΔT) of 4-interrupted fins more of than the rest cases because of the small surface area, where the surface area effect on heat transfer, in case of 4-interrupted fins have smallest surface area, that leads to weak capacity to carry the heat and cause high base temperature, This agreed with SenolBaskaya et al [11] and Salila Ranjan Dixit and Tarinicharana Panda [22] where the Figures 13-17 shows the base temperature distribution for continuous fins,1-inerrupted fins, 4-interrupted fins, inclined fins and V-fins respectively, for different environmental temperature at (25 watt) where the heated area marked with red color and cold area marked with blue color.

Table 2. Comparison between CFD model and experimental results

Heat sink type	Power = 5 W		Power = 10 W		Power = 15 W		Power = 20 W		Power = 25 W	
	T_{ext} = 20.2 C		T_{ext} = 19.9 C		T_{ext} = 17.6 C		T_{ext} = 20.6 C		T_{ext} = 24.5 C	
	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)
	26.25C	26.5C	34.6C	33.8C	36.25C	35.6C	44.8C	43.76C	55.9C	54.86C
	T_{ext} = 20.2 C		T_{ext} = 20.8 C		T_{ext} = 22.4 C		T_{ext} = 23.6 C		T_{ext} = 24.8 C	
	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)
	26.62C	25.12C	34.07C	33.37C	44.13C	41.6C	53.21C	52.64	58.24C	56.6C
	T_{ext} = 22.5 C		T_{ext} = 23.3 C		T_{ext} = 27.6 C		T_{ext} = 27.3 C		T_{ext} = 26 C	
	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)
	30.20C	29.31	38.4C	36.86	48.82	46.66	54.75	53.26C	62.36	61.63
	T_{ext} = 37 C		T_{ext} = 38 C		T_{ext} = 38 C		T_{ext} = 38 C		T_{ext} = 38 C	
	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)
	42.33C	40.88	48.51	47.36	53.35	52.64C	59.11	57.31	68.52C	67.02
	T_{ext} = 36.8 C		T_{ext} = 37.7 C		T_{ext} = 38.4 C		T_{ext} = 38.3 C		T_{ext} = 39.1 C	
	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)
	44.25	43.42C	50.22	48.84C	53.45	52.54	59.75	57.87C	66.21	65.65

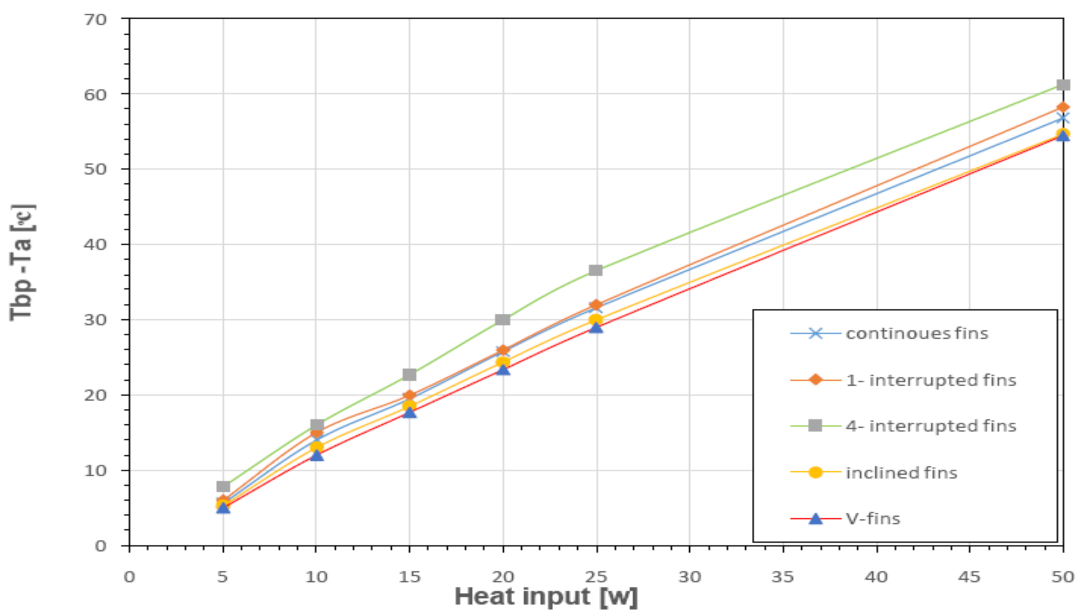


Figure 12. The variation between temperature different in calicoes and heat input

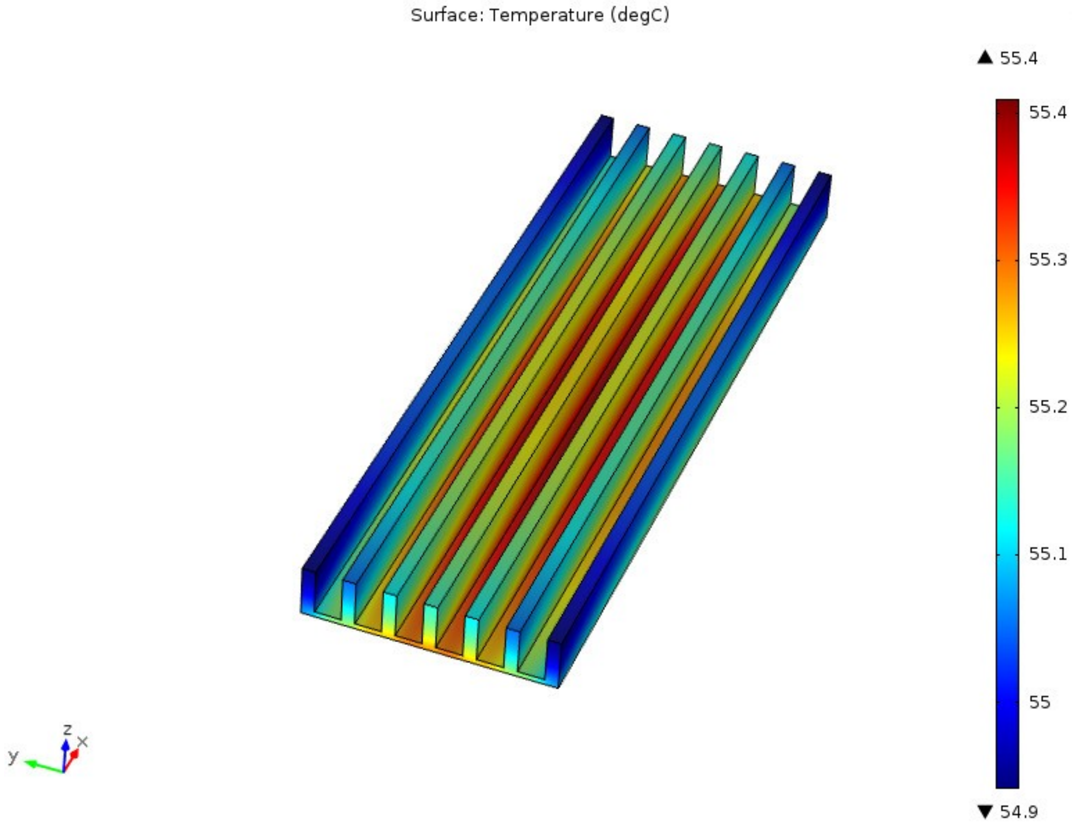


Figure 13. Base temperature distribution of continuous fins

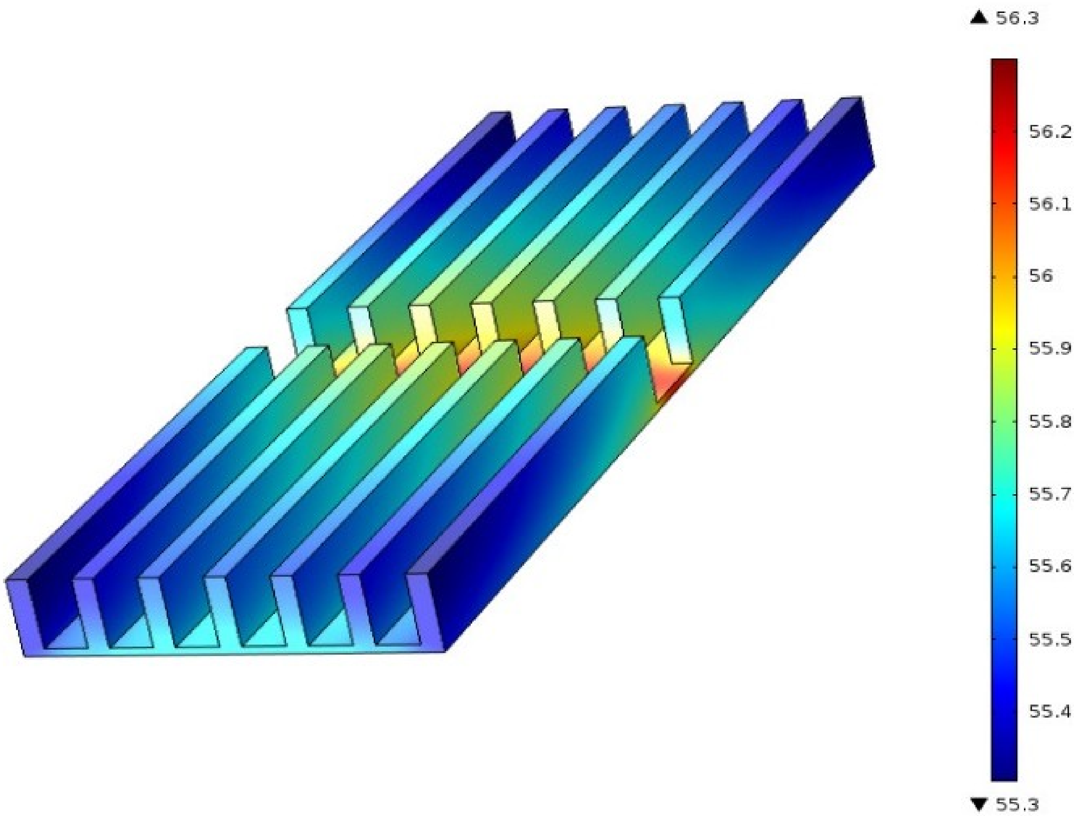


Figure 14. Base temperature distribution of 1- interrupted fins

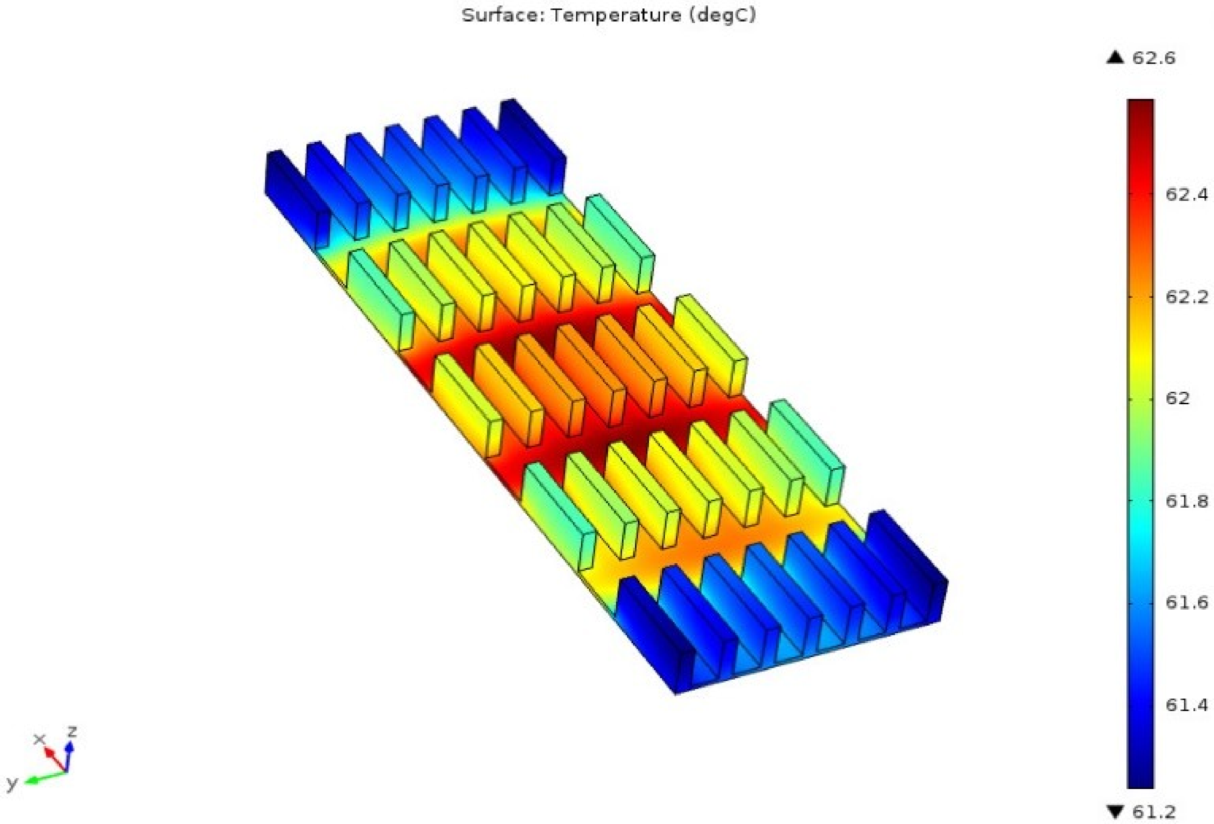


Figure 15. Base temperature distribution of 4-interrupted fins

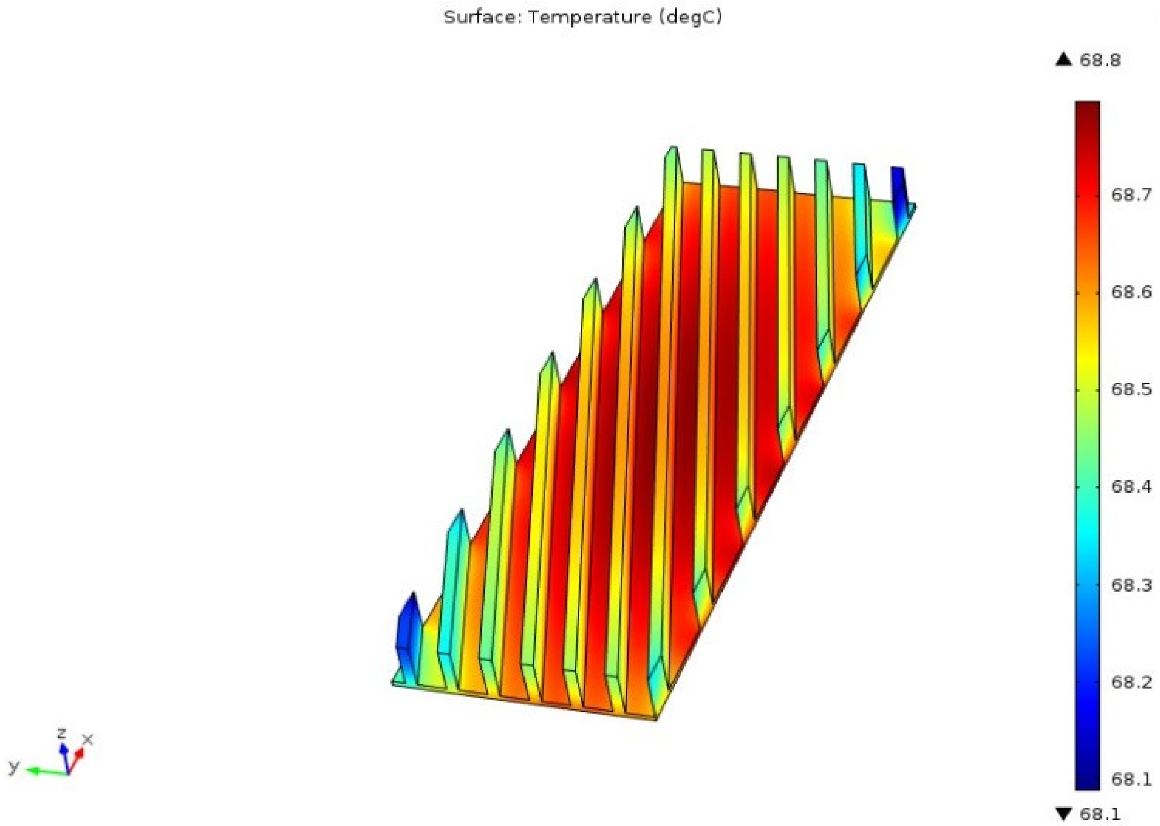


Figure 16. Base temperature distribution of inclined fins

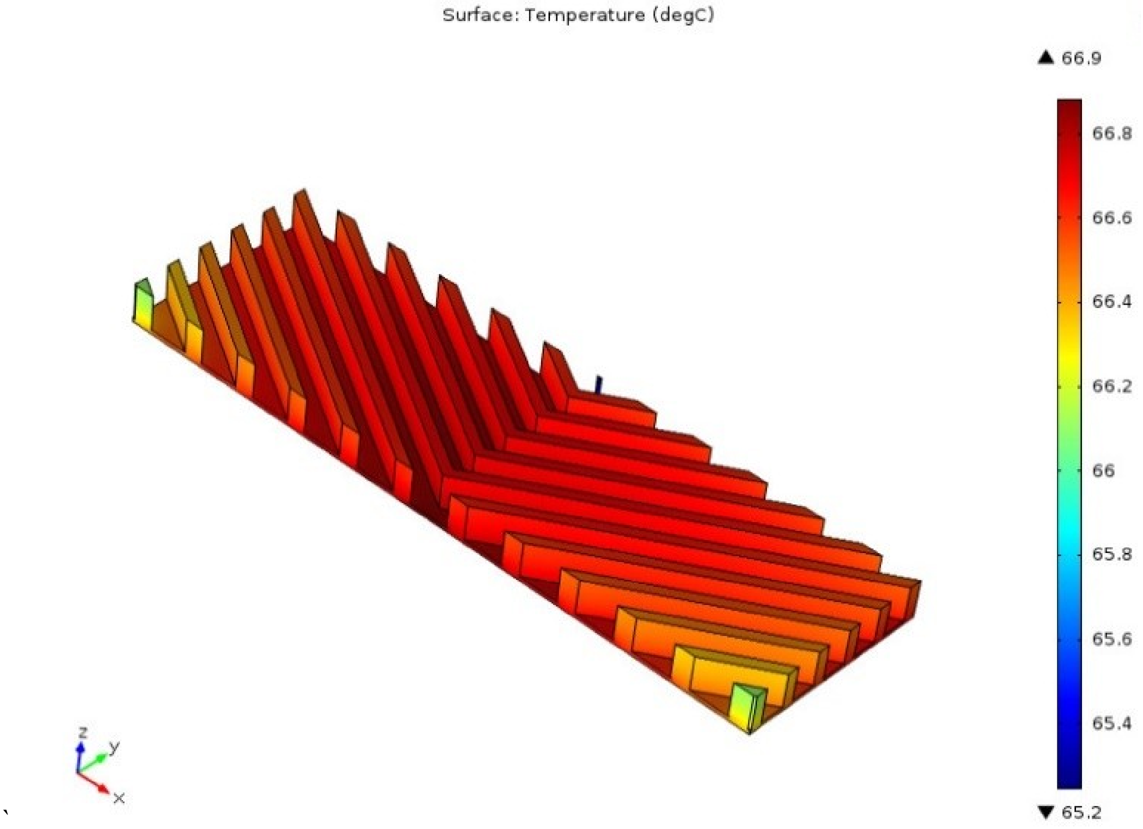


Figure 17. Base temperature distribution of v- fins

Figure 18 indicates the variation between the heat transfer coefficient and length of fins for continuous fins at (25 watts) and constant value of space (5mm) and height (18mm). When there is an increase in length, the heat transfer coefficient will decrease because that the length of fins reduces the velocity of air, that can be explicate by generating boundary layer at length of fins. Golnoosh Mostafavi and Majid Bahrami [26]. Figure 19 indicates the dissimilarity between heat transfer coefficient and space between fins at (25 watts), constant value of length (300mm) and constant height (18mm). The spacing between fins affect on velocity of air on heat sink, well the velocity direct proportional with heat transfer coefficient. When the spaces between fins are small then that will reduce the velocity. If the space between fins is large then the intersection of boundary layer won't happen and will cause reduce in heat transfer coefficient. The spacing between fins must be at a distance where the intersection of boundary layer that let the air enter the channel and out from its end at acceptable velocity that cause increase in heat transfer coefficient and this result agreed with S.G. Taji et al. [10].

The variation of heat transfer coefficient with fins height for same fins spacing (5mm) and length (300mm) for five type of fins shown in Figure 20, average heat transfer coefficient increases with fins height. An increase in the fins spacing causes an increase in the average heat transfer coefficient. The increase of fins height causes an increase in the pressure gradient in the z-direction along the fins length. This causes an increase in the flow rate of air entering the channel, therefore the average heat transfer coefficient increases and this agreed with result of H.G. Yalcin et al [16].

Figure 21 shows the effect of thickness of base plat on heat transfer coefficient for five different geometry of fins, increasing the thickness of base plat leads to decrease the heat transfer coefficient because of the losses in the heat input by conduction.

The variation between Rayliegh number and nusselt number indicated in Figure 22 for five type of heat sink and the length of plate take as characters length. Generally, increasing heat input will increase Rayliegh number and nusselt number because the increase in thermal properties by film temperature. Observe that nusselt number of heat sink with 4-interrupted fins higher because both of Rayliegh number and nusselt number depend on difference between base temperature and ambient temperature.

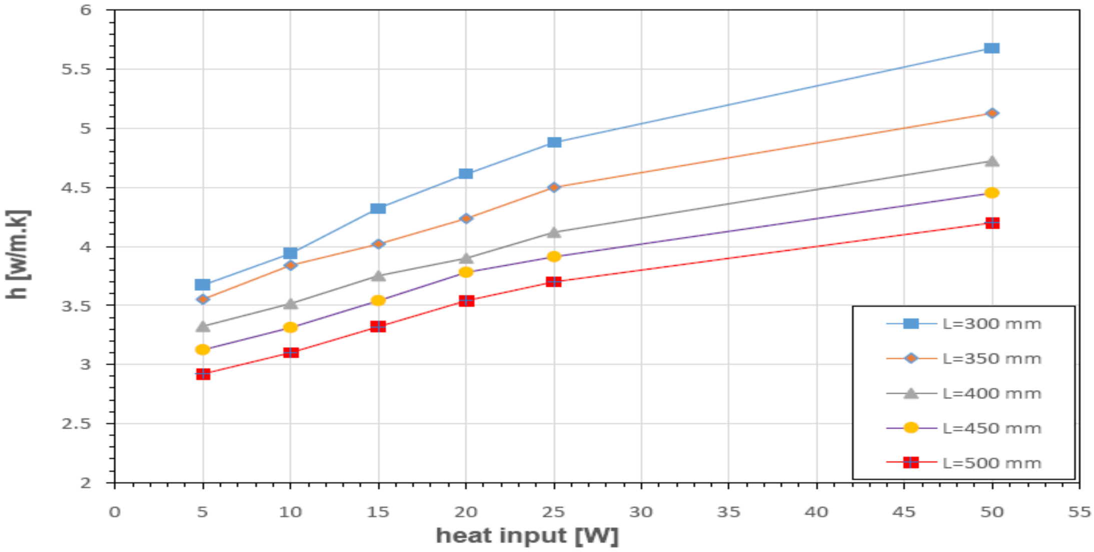


Figure 18. Variation between the heat transfer coefficient and length of fins for continuous fins

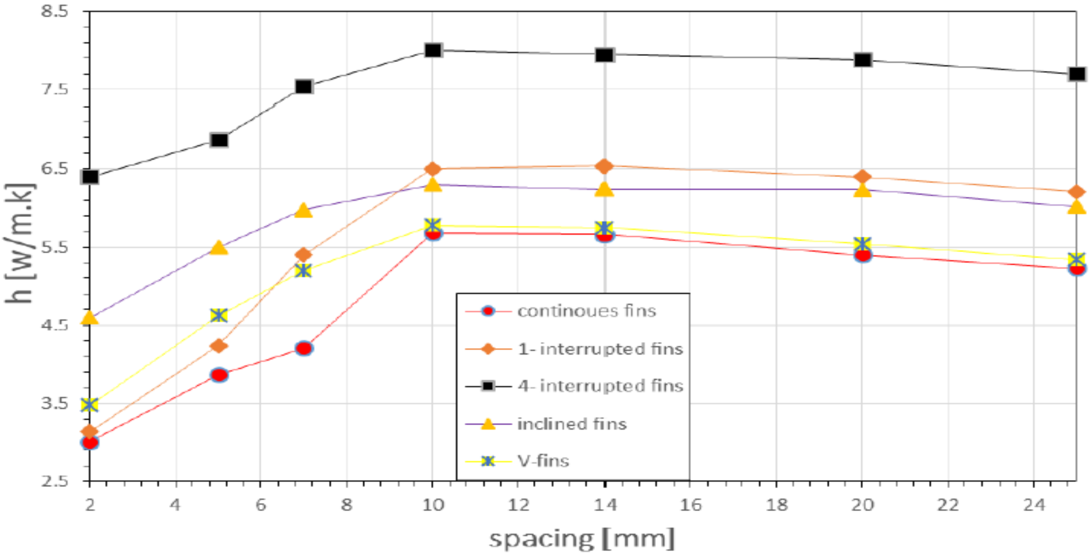


Figure 19. Indicate the dissimilarity between heat transfer coefficient and space between fins

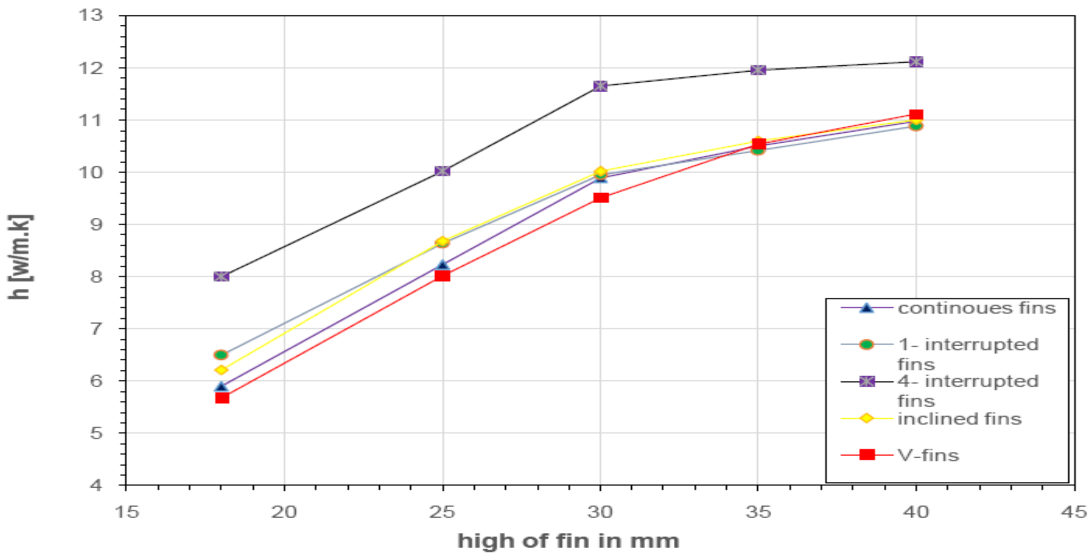


Figure 20. The variation of heat transfer coefficient with fins height

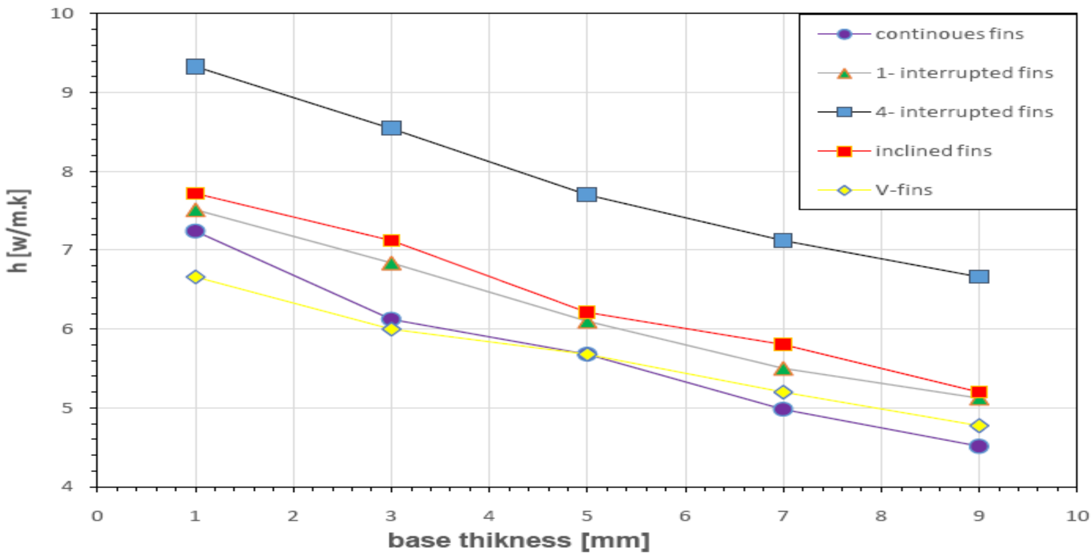


Figure 21. The effect of thickness of base plat on heat transfer coefficient for five different geometry of fins

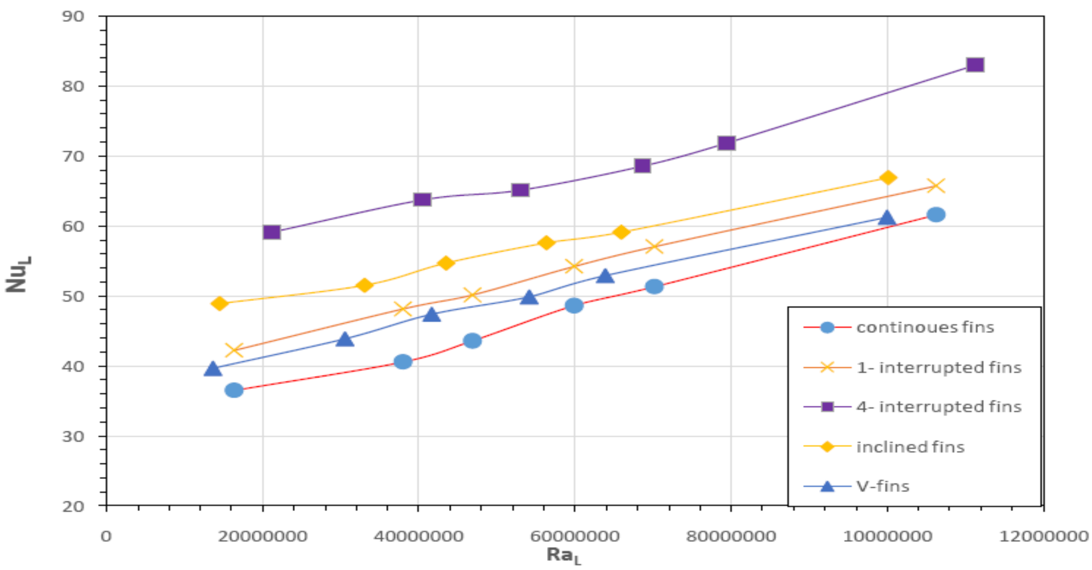


Figure 22. The variation between Rayliegh number and nusselt number for five type of fins

8. Conclusion

In this study, natural convection heat transfer for heat sink with rectangular fins for five different geometries (continuous fins,1-interrupted fins, 4-iterrupted fins, inclined fins and V-fins) at different heat flux values were investigated experimentally. The effect of geometric parameters (length of fins, thickness of base plate, fins high and the space between fins) on heat transfer coefficient and base to ambient temperature difference on the heat transfer performance of fins arrays was discussed. The following conclusions may be drawn from this study:

- The temperature difference between the base plate and surrounding air, ΔT at the same heat input rate, was found that the heat sink with rectangular fins type 4- interrupted fins have higher difference temperature with deviation (12.5%) between nearest case (1-interrupted fins).
- The results obtained show that the average heat transfer coefficient increases with the increase in the value of the fins height. According to this experimental study, the optimum configuration for fins arrays has been determined as the one having a maximum heat transfer coefficient at height 40mm by constant fins length 300mm and spacing 5mm.

- From results, heat transfer coefficient increases by increasing spacing (1-10) mm and it starts to decrease with the increasing in spacing more than 10mm. Where the result shows the maximum heat transfer coefficient at 10mm spacing with fixed value of length 300mm and height 18mm.
- The result shows increased length of fins for continuous causes a decrease in heat transfer coefficient. Where the heat transfer coefficient reaching its lowest value when length equal to 500mm at same heat flux, height of fins 18mm and same space between fins and other 5mm.
- When increasing the base plate temperature, decreasing heat transfer coefficient. Result shows heat transfer coefficient reaching its lowest value when base thickness equal to 9mm at constant height 18mm and spacing 5mm.

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