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Research Article

Effect of air fan position on heat transfer performance of elliptical pin fin heat sink subjected to impinging air flow

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ABSTRACT

Heat rejection from electronic components by heat sink is still a viable cooling solution. The optimal heat sink design enables higher heat transfer performance. The purpose of the present study is to predict the effectiveness of heat sink elliptical closely spaced fins subjected to impinging air cooling. The air fan is the main source of impinging air, then its position and direction with the heat sink take the main role in present work. Two positions of fan location are studied. The first position where the fan is outside the heat sink and the second case where the fan is existed in a cut out template. So there are one impinging air inlet with four transverse outlets and one axial exit opposite to the air flow inlet. Reynolds number were taken at a range 3400-16000, the flow was turbulent so k-ϵ model turbulence model was used as our choice to simulate mean flow characteristics for turbulent flow conditions. The heat sink base was subjected to constant heat flux condition and proposed with range between 10000–40000 kW/m^2 to keep the base temperature at a temperature around 100 °C. The Results of temperature contour lines depicted a variation from the base to the extended surfaces tips. The comparison between the two cases results showed high temperature difference in the case with the cut out template. Nusselts numbers indicated that the second case performed better in heat transfer than the first case. The experimental and numerical results showed a good agreement with a difference not exceeding 2%.

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INTRODUCTION

The proper working of semiconductors that make electronic circuit's components took a wide attention topic from designers. There is a correlation between cooling and the super work of electronic components. The high heat rejection that accompanies the electronic components work represents an obstacle to evolution of super computers. So the looking for cooling solutions aspect lead to the progress of cooling facilities. These efforts influence on both performance and reliability have been a respective

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importance in the overall design of electronic devices. One of the things that counts on electronic components performance is the dissipating adequate heat flux. The progress in electronics industries has enabled it to serve severe processing so overheating can be attended. Excessive working temperature effects on performance, reduces life time, and can cause failure [1]. The use of impinging flow enhances heat and mass flow have made them a focusable alternative in industrial applications, the thermal management of electronics is one of them [2]. The heatsinks using the air as a working fluid forced by a small fan with nominal speed attached to the top of a heat sink induces drawn or impingement axial air flow are employed as the solution with high power components [3]. It is indicated that with increase of chips working temperature may cause a reduction of reliability and stability. Therefore, the cooling means of disposal of extreme heat is worth to study. The dissipation of heat needs efficient way, the heat sink is one of the best solutions. The performance of the heat sink depend on the geometry of the fins so the fins can be existed in many shapes, the number of the fins and the fin thickness take the main role in heat dissipation. The heat source is closeness of the heat sink base center. For a permanent protection a continuous heat transfer system must be work effectively. A heat sink with the impinging air cooling one of the appropriate options of larger heat removal [4]. Air cooling heat sink low-priced, effective and reliable so it is a widespread in computers cooling. The generated energy by electronics component is transferred to the heat sink by conduction as a first step. In the second step is dissipated from heat sink to the ambient air by natural or forced convection depending on the requirement of the power dissipation. The heat sink has multiple pins with different geometries, elliptical pin-fin is one of multiple choices. The pins extend from the base and aligned at a computed pitch. The heat transfer and pressure drop characteristics along the computational domain can be presented for performance assessment. The cooling air flows between pin fins and contacts the hot surfaces obey to a technique that organizes its inlet and outlet. Impingement cooling can be selected in the applications which require high convective heat transfer rate. The coolant is forced and directed toward the finned heat sink may effectively reject high heat fluxes. Furthermore, the system can be worked at lower pressure compared to the other methods. The output of the cooling processes are largely affected by the fin shapes. The shape can impact on the heat transfer. So for the longer fins or fins with smaller thickness they became relatively more effective with heat transfer and important on pressure drop [5]. The dimensions of the extended surfaces, the fin height and a limitation on aspect ratio have a significant effect on heat sink performance. Fins that are too short show modest performance and may show overheating of the sink surface on the compromise the too long fins are more efficient since the efficiency of the fin is related with the fin height. There are a number of geometric parameters share in the design of optimal heat sink concern the fin characteristics such as fin material, height, length, thickness, fin shape or profile and the number of fins also the base plate thickness and the space between fins [6], [7]. The elliptical pin fin geometry having major axis and minor axis. To promote heat transfer performance the two axis must be varied to reach the optimal measurements. [8] They simulated the heat sink of elliptical pin fin. Many different major and minor axis were used under different velocities at constant heat input. When they changed the major axis, they obtained different values of thermal resistance in exchange for changing the minor axis. They deduced the value of the elliptical pin fin minor axis at all air velocity appearing better thermal resistance and pressure drop. The heat transfer depend on the fins or extended surfaces with minimal material spending, subjecting to the design and the ease of manufacturing of the fin shape. Fin performance can be examined by using the effectiveness of the fin, thermal resistance and fin efficiency to get maximum heat removal. Studies have been conducted on improving fin shapes while others have provided shape adjustments. To increase one or both the heat transfer and the heat transfer coefficient, by removing some material from fins to make cavities, holes, slots, grooves, or channels through the fin body [9]. Several experimental studies have been carried out and customized to the cooling of the desktop central processing unit by means of a heat sink that is exposed to an air impinging jet. The important parameter in the performance characterization of heat sinks is the jet height, jet diameter and the influence of the high to the diameter ratio [10]. Increasing flow rate can effectively improve the cooling efficiency but not absolutely. At both too small and too large distances impinging flow may increase the thermal resistance, which denotes that the design is optimal for the distance of the impinging flow [4]. The most important parameters that influencing on the thermal resistance are fin geometry height and width also the effect of fluid flow velocity by these parameters control which can be successfully get the thermal resistance improvement. So it means that pin fin heat sink is better than flat plate heat sink as thermal performance is considered [11]. The increase in the cooling fluid velocity provides better results in the convective heat transfer coefficient, which improves thermal performance [12]. There is a significant decrease in pressure drop with the increase in diameter ratios and velocity ratios in case of in-line fin compared with staggered arrangement [13]. More turbulences can be observed at higher Reynolds number which helps to enhance heat transfer. At higher flow velocity the Nusselts number increases but on other side flow become wavy and diverts from the target impingement. [14]. The thermal performance of the heat sinks with confined impingement cooling could be investigated by using infrared thermography. Increasing the impinging air velocity consistently diminishes the thermal resistance. Increasing the exposure surface by increasing the width of the fins increases, which basically enhances heat convection. Increasing the height of the fins beyond a critical value might also impede the penetrating ability of the impinging jets. The thermal resistance could be improved by increasing the impinging distance and it was significant for a small Reynolds number [15]. The Nusselts profile decrements gradually as the radial increasing away from the stagnant point. Increasing of the power takes place in near and far jet region, the impinging air causes more turbulence in flow which causes more heat transfer rate. As the radial distance is away from stagnation region the power increases and the power magnitude is the least at stagnant region. The generation of swirl and back flow may cause to restrict some heat transfer [16]. The high pin fin heat sink efficiency is an indicator of heat transfer increment. The parameters and techniques can share in the increasing of efficiency. In materials of fins aluminum were proved to be more efficient. Adding spacing and interruptions creating notches and slots can also be useful to improve the efficiency of pin fins. Elliptical geometry as compared to other geometries can also enhance heat transfer. The design supplementations like adding coating to fins and the pin fin array with dimples are another means to enhance heat transfer through pin fins. [17]. Motivated by aforementioned studies we formulate the problem to handle the cases with elliptical fins, aluminum made. One model with cut out fan template has enough seize to occupy the air fan and the second model is planned to work under fan fixed outside heat sink. As far as to our knowledge this case with this specific geometry, dimensions and the working heat transfer parameters here in after has not discussed before in the literatures.

MATHEMATICAL MODEL

The present study assumes that the impinging flow is steady, turbulent and 3D. Air is the cooling fluid it is assumed to be Newtonian and incompressible. Fluid thermophysical properties are assumed uniform except for density in the buoyancy term, which is modeled using the Boussinesq approximation. The governing equations of conservation of mass, momentum and energy are,

Continuity equation.

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{1}$$

Momentum Equation.

The momentum equations in x, y and z directions can be written as:

$$\rho u\frac{\partial u}{\partial x}+\rho v\frac{\partial u}{\partial y}+\rho w\frac{\partial u}{\partial z}=-\frac{\partial p}{\partial x}+\mu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right] \tag{2}$$

$$\rho u\frac{\partial v}{\partial x}+\rho v\frac{\partial v}{\partial y}+\rho w\frac{\partial v}{\partial z}=-\frac{\partial p}{\partial y}+\mu\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right] \tag{3}$$

$$\rho u\frac{\partial w}{\partial x}+\rho v\frac{\partial w}{\partial y}+\rho w\frac{\partial w}{\partial z}=-\frac{\partial p}{\partial z}+\mu\left[\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right]-\rho g_z \tag{4}$$

Energy Equation.

The energy equations for fluid are as following:

$$\rho C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=k_f\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right] \tag{5}$$

For Solid

$$k_s\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right]=0 \qquad \text{[12] or [13]} \tag{6}$$

Heat-Transfer Coefficient

For calculating the heat-transfer coefficient and Nusselt number:-

$$q''_w=-k\frac{dT}{dy} \tag{7}$$

$$q=q''_w A \tag{8}$$

$$q=hA(T_w-T_\infty) \tag{9}$$

$$h=\frac{q}{A(T_w-T_\infty)} \tag{10}$$

$$Nu=\frac{hl}{k} \tag{11}$$

To accomplish numerical analysis on the mathematical model, a package Fluent (Version 19) was used. The Navier-Stokes governing equations combined with the constitutive properties of the working fluid are solved by Fluent. The cooling air then passes through 7 rows each row with 6 inline elliptical pin-fin and after runs out through exit smooth section design to atmospheric pressure to prevent reverse or feedback flow boundary condition into the test section. To get the optimum configuration giving highest heat transfer, the resent geometry can be extended easily with staying the main surface area by changing the other.

Problem Geometry Specification

The perspective geometry design of the heatsink as shown in Fig. 1 was suggested after surveying many researches. It consists of 42 elliptical fins aligned in (6 × 7) parallel lines and attached on the upper side of the common

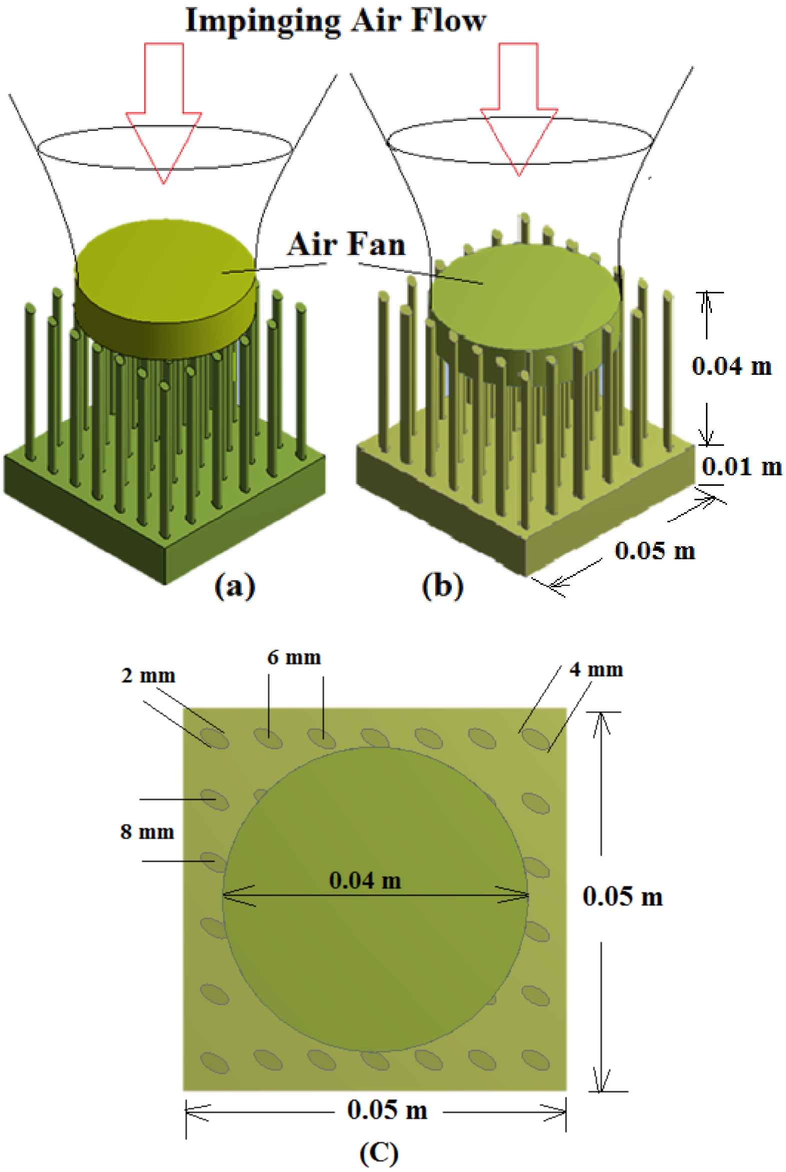

Figure 1. Heat sink geometry.

heated base plate in an inline manner. The base works as a virtual heated electronic component. The dimension of the square base plate is 0.05 m with 0.01 m thickness. Elliptic pin fin has minor and major axis equal 0.002 m and 0.004 m and height 0.04 m.The fins are arranged in (6 × 7) lines as shown in Fig. 2, the transversal and longitudinal pitches are 0.006 m and 0.008 m respectively. Impinging air passes through the heat sink as a cooling fluid by air fan. There are many virtual positions of the cooling air fan. Taking into account the available space, the first position proposed in this study is attached above the heat sink. The fan occupy space volume with 0.04m diameter and 0.01m depth. The second position is proposed by creating a enclose cut out template inside the longitudinal heat sink axis between the fins enough to fit the fan with the dimensions are mentioned

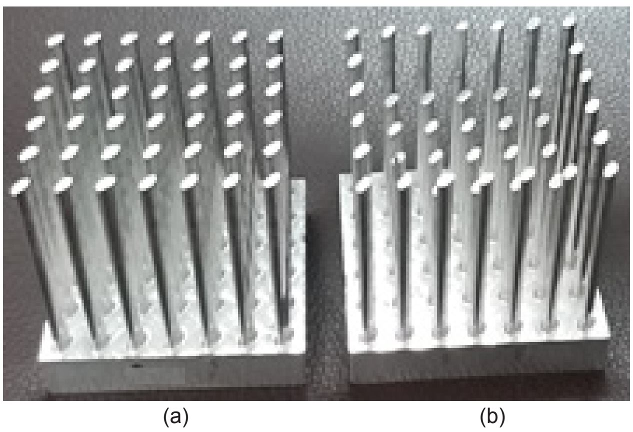

Figure 2. Machined geometry.

before. The work imperatives are pushing to capture the available size, therefore, the fan location must be within this size [18]. Impinging axial air passing through the fins as a coolant transport the heat generated in electronic component made from Aluminum with the thickness 0.01m attached to the lower side of the heat sink and generates a quantity of heat between (25–100W). The design is done in solid works 16, so that we can import this model easily in ANSYS.19. The machined geometries of the two types are depicted in Fig. 2.

Numerical Prediction

This study aims to simulate heat transfer in a heat sink with elliptical pin fins and validate the simulated results with actual experimental result using ANSYS-FLUENT (version19) software package. Various turbulence models have been developed using CFD software for predicting heat sink with elliptical fin pin under various impinging air velocity. Reynolds number were taken at a range (4000–16000) at temperature 300 K. The air is swept in by cooling fan. The outlet pressure is considered atmospheric. A constant heat flux ranged from (10000–50000 W/m^2) supplies energy under the heat sink generating a temperature below 100°C. Turbulent flow model is selected to predict the thermal analysis performance of elliptical pin fin arrays with constant physical properties. They were designed with the aid of a Solid works program version 16.

CFD Simulation Approach

The simulation procedure by the ANSYS FLUENT 19 CFD code was started with pre-processing after the computational mesh which generated using mixed elements with the number of elements 1309364 and nodes 489921 Fig. 3,a b, and c, mesh validation is clear in the table 1 and table 2.

A SIMPLE algorithm was used with the discretization scheme used coarse meshing at first order upwind scheme. The flow domain and heat transfer were solved by iterative determination for the set of continuity, momentum and energy governing equations. The k-ϵ turbulence model with two transport equations to represent the turbulent characteristics of the flow. The under-relaxation factors were set to stabilize the calculation process and the solution convergence at the proper time. The normalized residuals were set at 10^{-6} for the components of velocity and at 10^{-7} for the equation of energy, which proved to be adequate [19]. During the pre-processing steps the material must be defined, air and Aluminum have been chosen, so their properties were be taken in account.

Table 1. The validation of mesh generation

	Number of Nodes	Number of Elements	Temperature Difference T_h°C	T_l °C	DT °C
1	587277	1648343	315.4	312.5	2.9
2	516025	516025	315.6	312.6	3
3	489921	1309364	315.6	312.6	3
4	357483	1032771	315.4	312.5	2.9
5	339065	921439	315.6	312.7	2.9
6	249764	651472	315.9	313.3	2.6

Table 2. The validation of mesh generation for heat sink with cut out template

	Number of Nodes	Number of Elements	T_h	T_l	DT
1	587277	1648343	344.4	322.5	21.9
2	516025	516025	344.5	322.5	22.0
3	5433535	4628440	344.7	322.6	22.1
4	357483	1032771	344.6	322.5	22.1
5	339065	921439	344.6	322.6	22.0
6	249764	651472	344.6	322.6	22.0

Boundary Conditions

The analysis is done with atmospheric impinging air flow at temperature of 300.0 K. The bottom base surface at uniform heat flux so the heat is transferring from the base surface to the top surface. The four sided walls of the bottom are assumed to be adiabatic. Fins are aligned and welded to the base. The heat is transferring from the solid heat sink to the surrounding atmosphere by convection. Air enters into the Heat sink with Reynolds number 4000–15000 in the Z-direction according to the geometry. The heat sink gets rid of the cooling air to the atmospheric pressure so an enclosure around the heat sink is created to study the flow and heat transfer of the coolant flowing over and through the heat sink as shown in the Fig. 3.a. The geometry mesh generated is shown in Fig. 3.b and c). A SIMPLE solution method solved the energy equation with the two viscous turbulent (k-epsilon) model. Open loop wind tunnel with dimensions 0.4 × 0.4 × 0.2 m is used to do the experiments. Manometer with pitot is used to measure the blower air velocity. Blower air velocity can be varied by electric regulator. The impinging air is 0.06 m far from the above side of the test section. The inlet air temperature is kept at 300 K by controlled air conditioner. Temperature degrees are monitored by thermocouples k-type inserted in the sample at the three more importance sections, at the base bottom, base upper side and the fins upper side. Temperature reading is done by temperature recorder. The heater temperature is controlled by variac. The rig was assembled as shown in Fig. 4.

RESULTS AND DISCUSSION

The current study aimed to delineate the influences of utilizing elliptical pin fin heat sinks. The effects of different

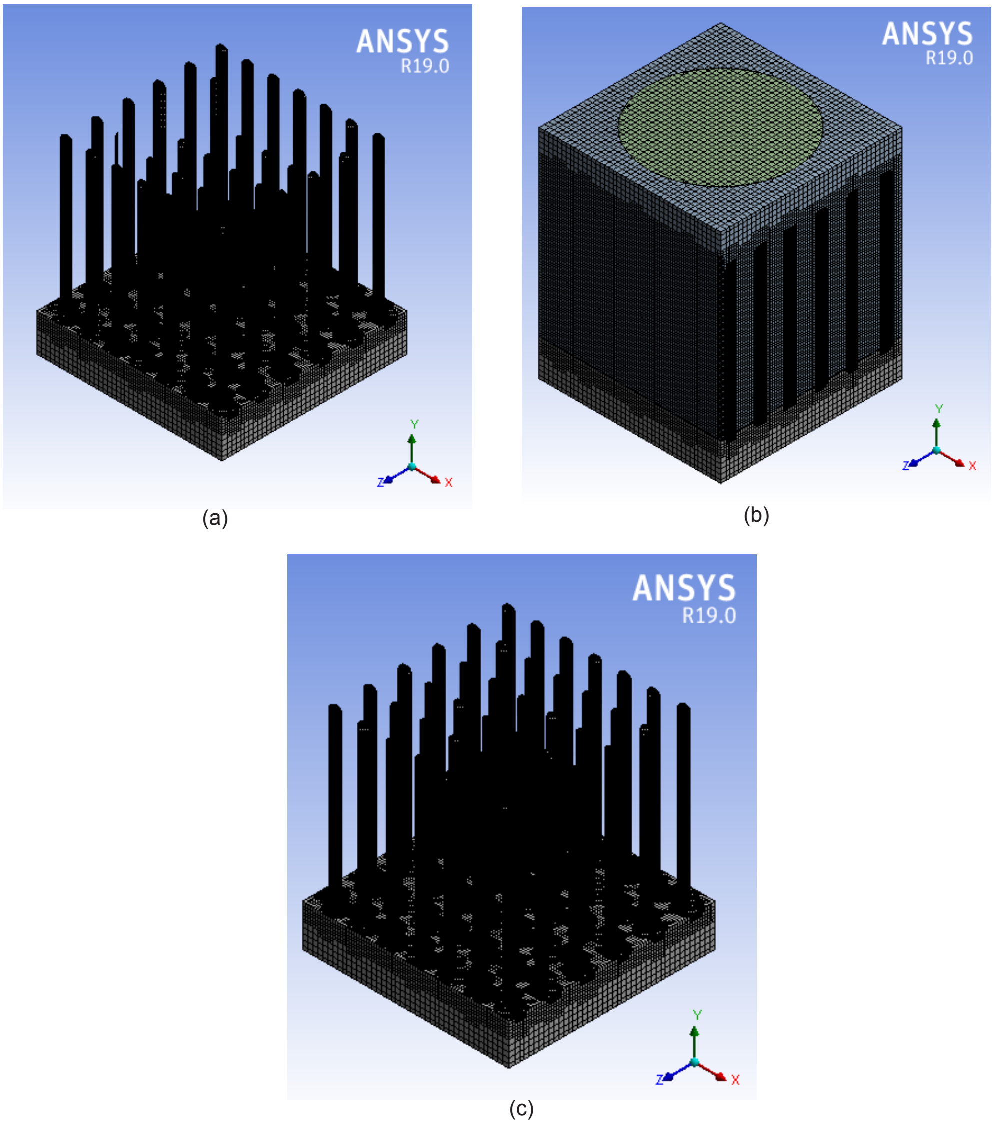

Figure 3. Enclosure and mesh generation.

parameters such as impinging air flow rate and heat flux on the temperature reduction and standard deviation of the temperature field alongside the heat sink surfaces are studied. There are wide use of heat sink, one of these applications is the cooling of electronic components which limit the working temperature below 100°C [20]. The processor is represented by the Aluminum heat sink base with thermal conductivity of 202 W/m K. The 42 fins with the geometry above mentioned are under heat transfer with conduction and convection along all the boundaries except the bottom from which heat flow toward air flow domain. Impinging ambient air with controlled temperature at 300K enters from the four sides and exits the four sides. Many studies were done to decrease the maximum temperature and increase the heat flux in the electronic component by varying fins geometry and the fan position. Referring to the temperature variation contours (Fig.5, Fig.6) for the two cases at high Reynold number, the fins that aligned in a circle under the cooling fan, can be said the heat sink core about 20 units there are no significant temperature gradients. The cooling air entered from a main one inlet flow and there are four out flow, but the air forced to flow between narrow passages till reach the outlets. This caused the air temperature to be increased. This because their positions are in touch with the high velocity cooling air and the temperature gradient lines overlapped. On the contrary during low Reynold number (Re = 3400) there are a significant temperature gradients in all extended surfaces. Generally the overall impression of the matter encourages to reduce their length and cutout enclosure template. It is axiomatic in the present case that energy dissipated by conduction and convection and the length is important in the heat transfer. When there

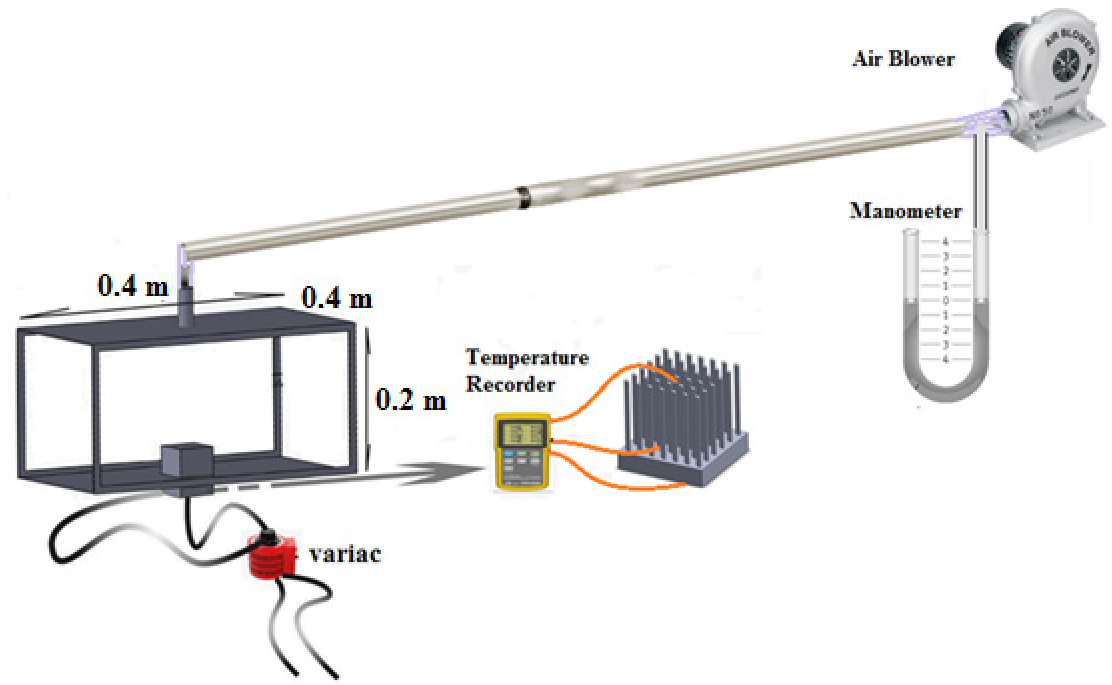

Figure 4. Experimental setup.

is a shortage in conduction in other side it is compensated by convection. The air flow jet has a flow diameter equal to the fan diameter and this diameter not necessary cover all the heat sink area depending on the design imperatives. So when removing part of the fins as much as the size of the fan lets other passages to the fins aligned surrounding. The fins aligned below the fan depicted more effective than the fins around this because that the impinging cooling air passes through the internal fins, which are facing the cooled air. The surrounding fins heated air passes through them. Although the mass of the fins is limited and affects during conduction heat transfer, it is compensated by convection heat transfer for the surrounding fins. Nusselts number variation with Reynolds number is depicted in Fig. 7. The curve trend of Nusselts number showed that the fins with cutout enclosure template are the best in heat transfer and the solid fins more effective in heat transfer than the hollow fins. The differences in temperature degrees between the heat sink base and the fins tip represent the overall heat transfer by conduction and convection. Fig. 8 illustrated that the variatin of heat transfer in (Watt) with temperature difference, for the fins with cut out tempelate is more effective than that fins using cooling fan out side the heat sink assembly Fig. 9 depicts that total thermal resistance for the heat sink decreases with increasing Reynolds number. The minimum thermal resistance occurs for heat sink with cut out fins. At a same heat flux the fins with cut out produces low temperature difference. The reduction in thermal resistance with increased Reynolds number is attributed to the enhancement in the heat transfer coefficient due to the generating flow turbulence by the high speed air flow through the multi exit in opposite transverse and axial directions. [21]. There is a good agreement between the experimental and predicted results. As the increasing in Reynolds number there is a reduction in the thermal resistance [22]. This attributes to the heat transfer coefficient enhancement by the flow turbulence which is generated by the air flow in opposite axial or transverse directions. The error deviation between the experimental and predicted results is clear in the error bar and it showed a value not exceeded 2% as shown in Fig. 10. From the many cooling systems applied to the electronic components, the best selected one depends on the amount of heat generated. There are many additional factors help in choosing the appropriate system are the conditions of the environment, the cooling system cost and the reliability applications for the applied cooling method [23].

CONCLUSIONS

From the fore mentioned investigations the following conclusions that have been inferred.

- The impinging air cooling flow strikes the heat sink from the coolest part along the heat sink base which is the hottest part.
- The cooling process two main temperature different part at the fin tip and the fin base which increases the heat conduction.

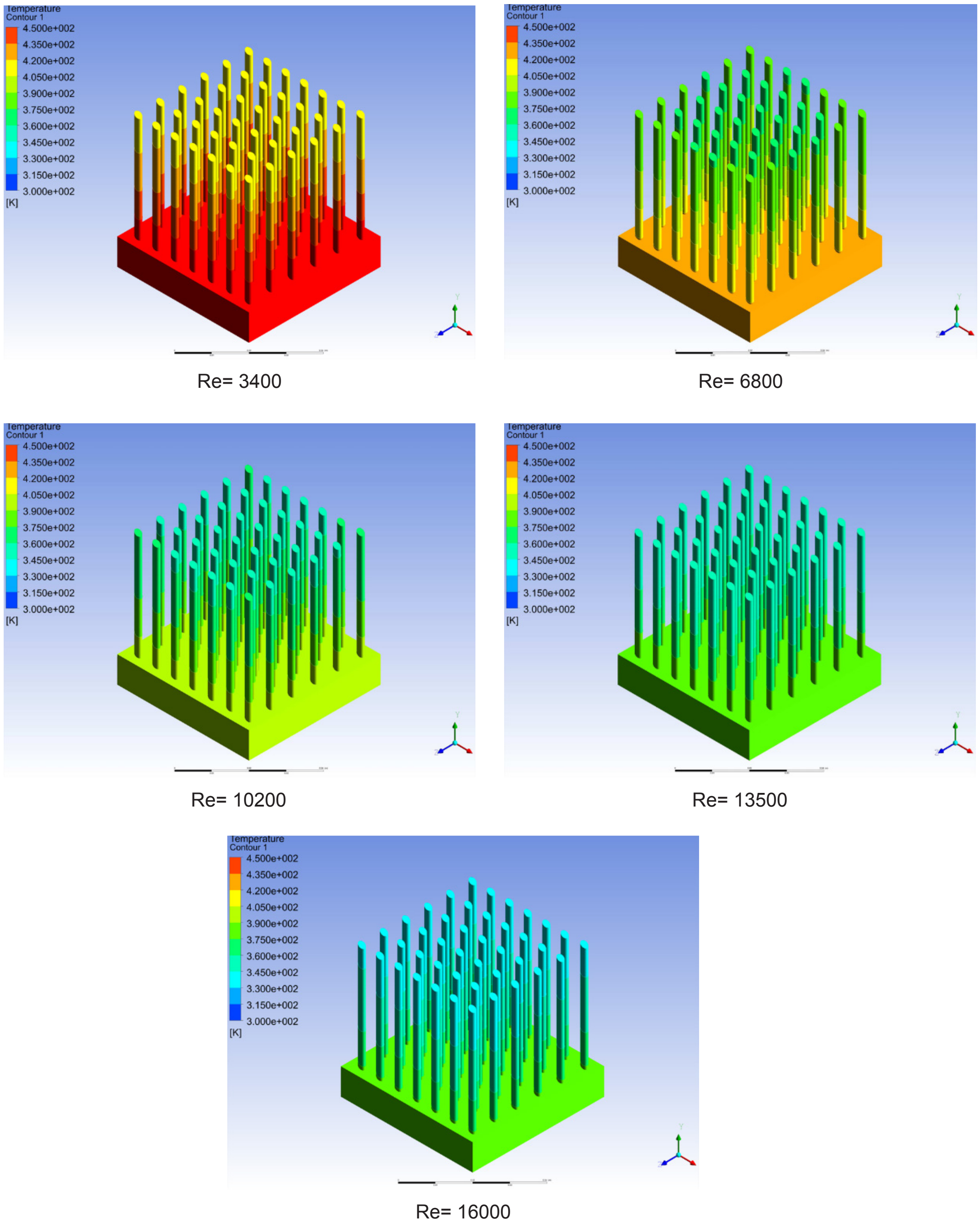

Figure 5. Temperature contour lines of heat sink.

- The fins under the cut out template are shorter than the other fins around, but this air gap allowed the air to cool the surrounding fins easily.
- The drop in temperature degrees along the shortest fins is consistently greater than the temperature difference recorded along the surrounding fins.
- Nusselts number values depicted an increasing as the Reynolds number increased, also heat sink with cut out template showed higher Nusselts number with the same Reynolds number values.
- Thermal resistance values decreases as the Reynolds number increases, and heat sink with cutout template depicted higher thermal resistance with the same Reynolds number values.
- The lost heat is directly proportional to the impinging air speed, and the heat sink with cut out fan template depicted more effective in heat rejection.
- Nusselts number values increases as the combined increasing of Reynolds number and heat flux.

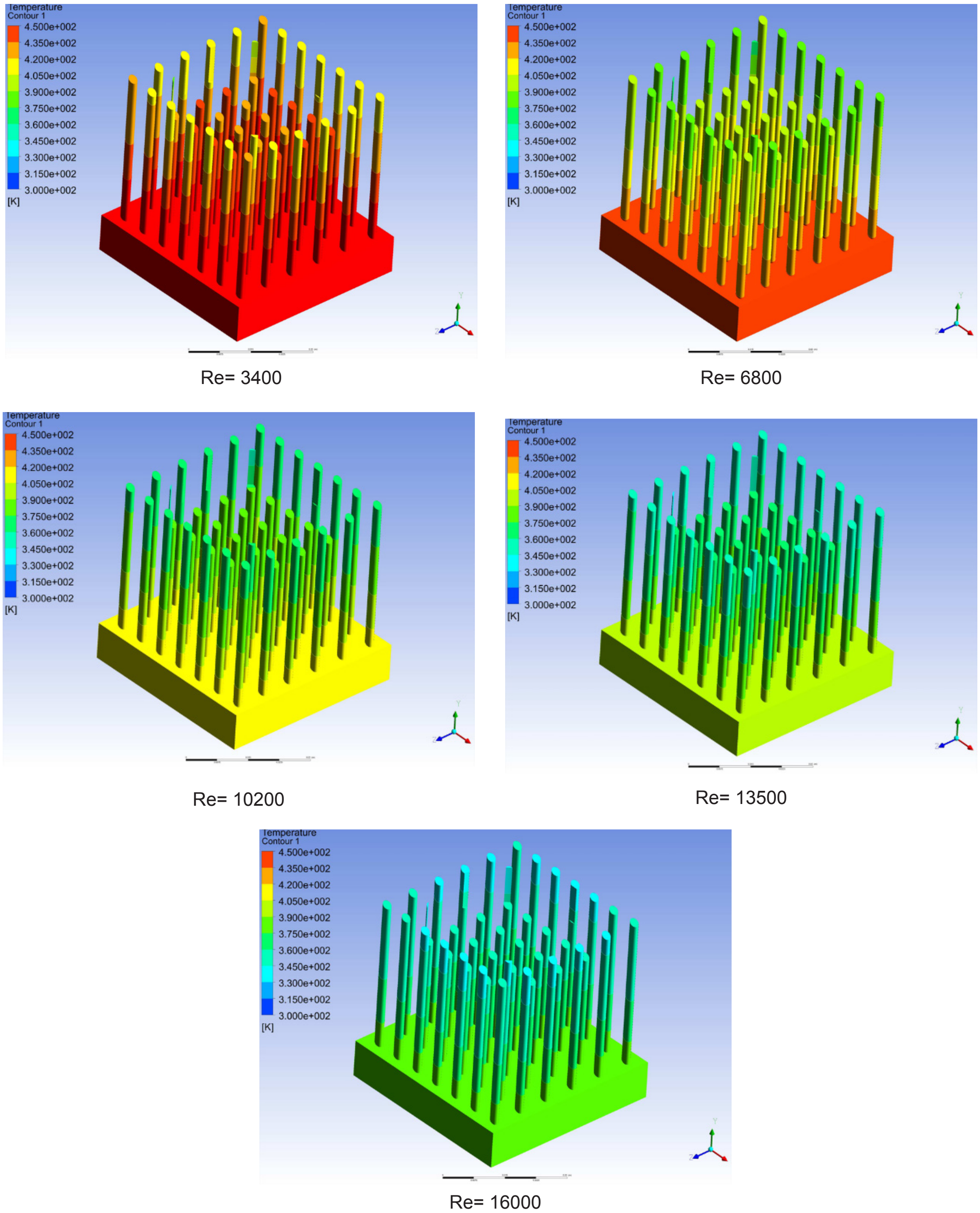

Figure 6. Temperature contour lines of heat sink with cut out.

- The temperature difference which can be calculated between the higher and lowest heat sink temperature degrees depends on the twice heat flux and Reynolds number.
- From mesh validation it can be concluded that the increasing mesh elements is not necessarily lead to improve the experiment results.

NOMENCLATURES

A	Surface area [m^2]
DT	Temperature difference [^oC]
D	Diameter [m]
g	Gravity acceleration [m/s^2]
h	Heat transfer coefficient [$W/m^2.K$]
k	Thermal conductivity [W/m.K]

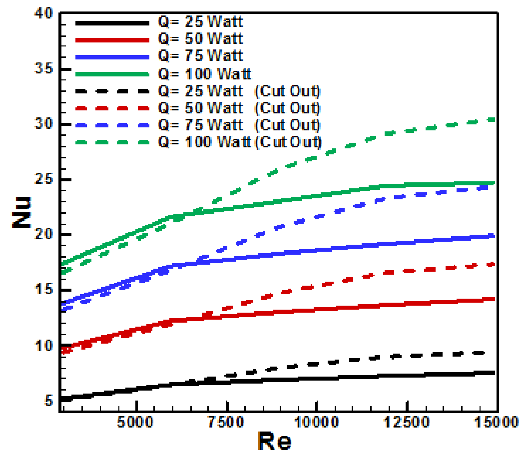

Figure 7. Nusselts number variation with Reynolds number.

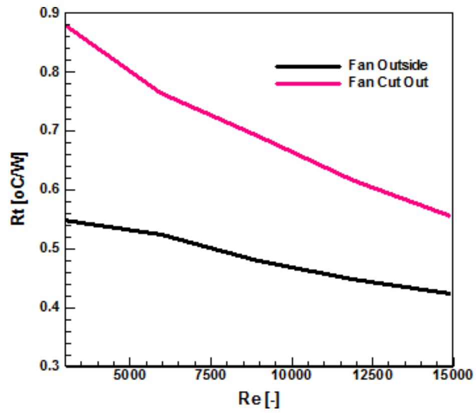

Figure 9. Thermal resistance as a function of Reynolds number.

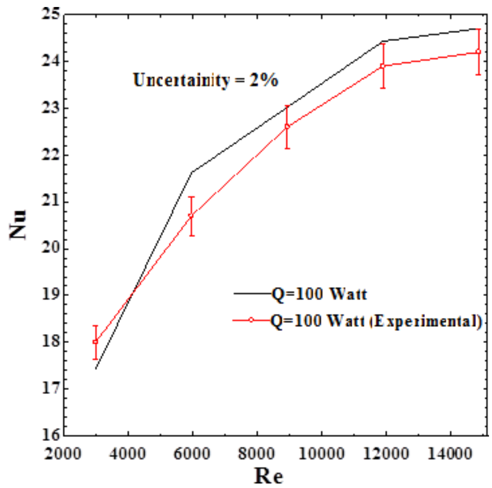

Figure 8. Temperature difference with heat flux variation.

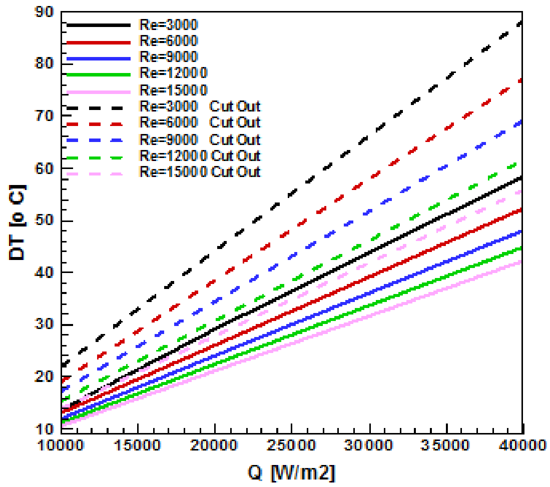

Figure 10. Comparison of experimental results with numerical results

l	length [m]
Nu	Nusselts number
P	Pressure [N/m^2]
q''_w	Heat flux [W/m^2]
Q	Heat [W]
Re	Reynolds number
Rt	Thermal Resistance [°C/W]
x	The horizontal coordinate, [m]
T	Temperature [K]
u	The horizontal velocity component in x direction, [m/s]
v	The vertical velocity component in y direction, [m/s]
W	The horizontal velocity component in z direction, [m/s]

Greek Symbols

μ	Dynamic viscosity, [N.s/m^2]
ρ	Density [kg/m^3]

Subscript

w	Wall.
∞	The value at atmospheric condition.
f	Fluid
s	Solid

DATA AVAILABILITY STATEMENT

No new data were created in this study. The published publication includes all graphics collected or developed during the study.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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