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Numerical and experimental thermal performance of forced convection in metal foam heat sinks

Ban Hamed Abdullah, Abbas J. Jubear, Hussein R. Al-Bugharbee

Department of Mechanical Engineering College of Engineering-University of Wasit

Abstract

Open-pore metallic foams have been used in a variety of applications in recent years due to essential characteristics such as low weight compared to size, high permeability, and efficient thermal performance. The thermal performance of an open-celled copper finned heat sink was examined experimentally and numerically under forced convection in this study. The copper foam used in the two parts of the analysis (numerical and experimental) is open-celled, with 90 percent porosity and 15 pores per inch (PPI) . The heat sink fins are mounted on a (100 * 100 * 3) mm3 solid copper plate. The heat input range was (5, 10, 15, 20, 25, and 30) Watts. The flow is incompressible and turbulent in all cases of analysis.

For the numerical component, the thermal behaviour of the foam heat sink was simulated using ANSYS Fluent R19.0 software. The air velocity values evaluated in this analysis were (1, 2, 3, 4, and 5) m/s. Furthermore, the number of fins varies as (1, 2, 3, 4, 5, and 6) fin. This simulation takes into account the influence of the heat sink direction with regard to the air flow, both perpendicular and parallel.

The heat sink with six fins was installed within an air tunnel setup for the experimental part. This experimental setup comprises a (130 * 130) mm2 square test section with a length of 200 mm. There were three airflow velocities used: 1, 2, and 3 m/s.

The simulation study results include the difference between the base and ambient temperatures, the temperature distribution of the heat sink, and the Nusselt and Reynold numbers. The ideal air flow velocity is discovered to be 3 m/s. The optimal number of fins was discovered to be six, which resulted in an 82.2 percent increase in heat transfer coefficient when compared to a single fin case.

In comparison to the opposite direction, the parallel orientation was demonstrated to have a favourable affect on the heat sink thermal performance. This benefit grows as the number of fins rises, with an average 48.7 percent increase in temperature differential decrease.

The comparison of the simulated and experimental findings reveals a significant correlation of 95.9 percent.

1. Introduction

Thermal management represents one of the main issues in the electronic packaging processes because high heating causes many problems notably that it reduces the life of electronic components and weaken their reliability unless there are ways to control it. Although many advanced cooling methods are suggested as liquid cooling or cooling by two -phases, the thermal management of the electronic components depends largely on the heat sink with the extended surface, or so-called fins, through which the heat source is dispersed. The design of extended surfaces (fins) greatly determines the area of heat transfer and flow specifications within the structure. The perfect structure must have a high fluid-solid interfacial area as well as a high heat transfer coefficient, but low flow resistance. Anyway, the first two factors usually opposed to the last. Therefore, one should design the structure of extended surfaces ideally to balance between the conflicting factors to gain maximum heat transfer under any specific conditions. Typically, modern heat sinks forced convection heat transfer use in most applications specific fins structures as extended surfaces. There are a lot of studies interested in this regard and using the extended surfaces in various forms and different, such as parallel continuous fins [1, 2], radial continuous fins [3,4], tree-shaped continuous fins [5], pin-fins [6], pin-fins with single hole [7], etc. In addition to all the aforementioned structures of fins, open metal foams have been used as extended surfaces also due to their light weight, high thermal efficiency, high surface-to-volume ratio, excellent fluid mixing. Due to these important advantages, metal foams have become attractive to many researchers and have been increasingly used in cooling.

Ali Samir and Ihsan Yahiya (2019) [8], investigated six vertical tubes (16mm inner diameter and 140 mm length) filled by copper foam (10,20, 40, 60 and 80 PPI) with 95 % and 98% porosity for each under forced convection. The working fluid was (R-134a) because of its zero ozone depletion. It entered the domain at uniform velocity, 0.6 MPa pressure and 10°C as constant temperature while heat flux was (10-80 kW/m^2).

Results showed that the average Nusselt number was higher for high pore density at constant heat flux.

Christopher, et al. (2009) [9], studied an aluminium foam fin of 20 PPI and 90.05 porosity. Its dimensions were (62.5*62.5*56.25 mm) under forced convection heat transfer to find the total heat transfer from the base that effected by multiple extended surfaces. The thickness is constant for a total fin of 6.25 mm and divided into (1, 2, 4, 6 and 8 fins).

The researcher used Computational fluid dynamics (CFD) for calculation. The data showed that the addition of fins distributed the heat more steadily in the heat sink and improved effectiveness. The heat transfer enhancement was highest when added six fins were with the aluminium foam block.

Nicholas and Sanjeev (2012) [10], investigated forced convection heat transfer of copper and nickel foam heat exchanger tubes. The nickel and copper foams cross-section was (300 mm length, 20 mm wide and 20 mm height) with 10 and 40 PPI (pore densities) and 94% porosity. The channels heated by the outer heater wrapped around foam at uniform heat flux varying from 427 to 6846 W/m2, while the air compressed through the channels at varying flow rates of (5–80 L/min).

The results achieved an increase in heat transfer coefficients when increased the fluid flow velocity, while heat transfer coefficients and volumetric heat transfer coefficients for the copper foams were always higher than nickel foams.

Nihad and Krunal (2010) [11], Studied the length effect of aluminium foam samples on flow properties experimentally to obtain a minimum length necessary. The cross-sectional area of foam samples was 101.6 * 101.6 mm had different lengths, while their pore density was (10, 20, 40 PPI) and porosity of (92.9, 92.5, 93.5), respectively. The foam samples were subjected to airflow rates of more than 17 m3/min by powerful suction.

Results showed that the flow parameters have a strong relationship with the thickness of foam samples, but it became independent when the thickness of foam samples was equal to or more than a hundred cell sizes.

Tamayol and Hooman (2011) [12], Studied numerically forced convection heat transfer in rectangular and tubular heat exchangers filled by metal foam to compare results with another experimental study. The rectangular heat exchanger has three different heights (45, 65 and 90 mm) and its length 0.112 mm while the diameter of the tubular one was 180 mm. The pore density for the metal foam was (5, 10 and 40 PPI) with different porosity.

The results show that the Nusselt number increased when the height of the rectangular heat exchanger without linear relationship increased. This is due to the higher surface temperature of the ligaments near the heated surfacewhich contribute to the overall heat transfer more than those away from the wall.

The main objective of this study is to reduce the base temperature of the heat sink by Selection the best fins number and the heat sink oreintation. This changes contributed effectively to reducing the temperature of the base and achieved outstanding results in terms of thermal performance of the heat sink.

2. Problem Statement

The reliability of the performance and the life expectancy of the electronic components and equipment are important factors that are closely related to the temperature of any electronic device. The relationship between them is inverse. However, the reduction of the temperature is fully compatible with increasing the reliability and age of the electronic devices. So, control of the temperature of the device through free convection is of vital importance. Therefore, the focus of this study was to find the best fins number and its orientation , which proved the better ability to cool electronic components.

3. Numerical Analysis

The numerical solution comprises a discussion of the following sub-sections: governing equations, computational of the domain, the grid-independence analysis, setting model, setting the cell zone condition, defining boundary conditions.

3.1 Computational Domain

The different components of the model were created by using ANSYS R19. software. These components consist of fins, base plate, and a box. These components are individually built up and after then assembled. The fins were arranged in the middle as well as the base on which the fins were mounted and then the enclosure box was constructed. All these components are lastly assembled to be ready for the mesh as shown in figure 1. The foam fins were fixed on an aluminum base plate of dimension 100 * 100 * 3 mm. The geometrical dimensions and some of the properties of the metal foam and air that’s used in the present work are listed in Table 1 and Table 2.

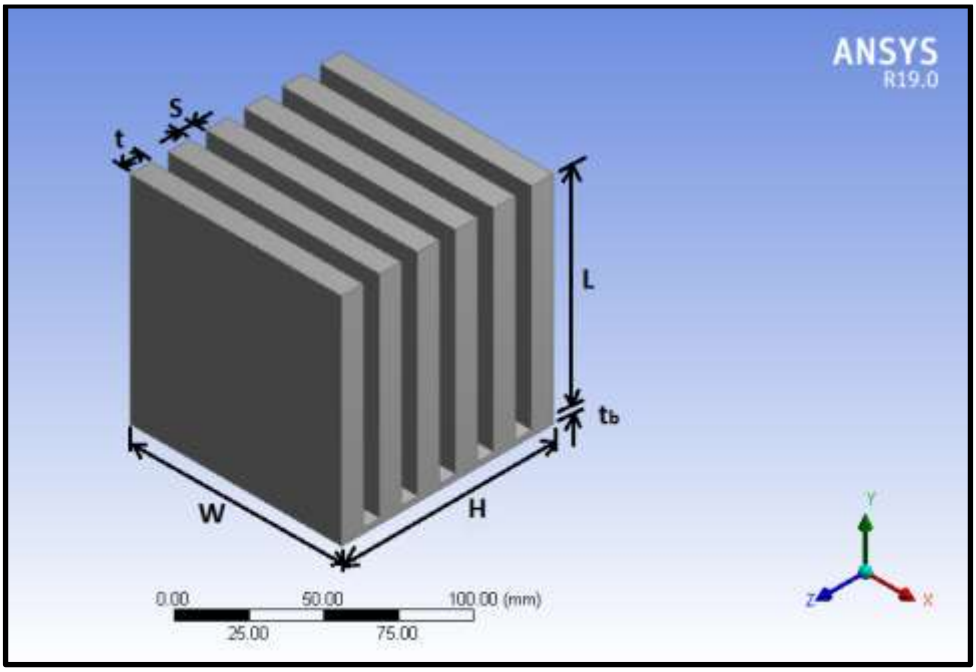


Fig. 1: Heat sink with same length

Table (1): Dimensions of the copper foam fin samples

Number of fins (N)	Length (L) mm	Width (W) mm	Thickness (t) mm	Fin spacing (S) mm
6	100	100	10	8
5	100	100	10	12.5
4	100	100	10	20
3	100	100	10	35
2	100	100	10	80
1	100	100	10	-

Table (2): Copper properties

Types of materials	Density $\rho[kg/m^3]$	Heat capacity Cp [J/kg.K]	Thermal conductivity k [W/m.K]	Porosity ε	PPI
Copper foam	8978	381	387.6	90 %	15

3.2 Assumptions

In order to solve the equations of the flow and heat transfer (continuity, momentum and energy equations), sets of assumptions must be made to simplify the problem.

- Turbulent flow.
- Steady state.
- Incompressible fluid.
- There is no internal heat generation and neglecting the radiation
- The energy equations are written based on a local thermal non-equilibrium (LTNE) model in which the temperature of the solid matrix and the passing fluid are solved separately.

3.3 Governing Equations

The governing equations used in the ANSYS simulation include the momentum equations as well as the continuity and energy equations that belong to porous mediums, especially metal foams [13].

Conservation of Mass

The principle of continuity is that the input mass of any system must be equal to the mass outside it through the surface of the control volume. The expression of this is mathematically for non-compressible fluid as follows:

$$\frac{\partial \mathrm{Vx}}{\partial x}+\frac{\partial Vy}{\partial y}+\frac{\partial Vz}{\partial z}=0 \tag{1}$$

Momentum equations

The other principles governing fluid flow are derived from Newton's second law, called the Navier-Stokes equations (the conservation of momentum). And with respect to incompressible fluids the full instantaneous equations are taken in this form: when the gravity is in −y direction, the body force vector g = [0, -g, 0]. The momentum equations are:

In X-direction

$$\frac{1}{\varepsilon^2}\left(V_x\frac{\partial V_x}{\partial x}+V_y\frac{\partial V_x}{\partial y}+V_z\frac{\partial V_x}{\partial z}\right)=-\frac{\partial p}{\partial x}+\frac{1}{\varepsilon\,Re}\left(\frac{\partial^2 V_x}{\partial x^2}+\frac{\partial^2 V_x}{\partial y^2}+\frac{\partial^2 V_x}{\partial z^2}\right)-\frac{1}{Da\,Re}V_x-\frac{F}{\sqrt{Da}}V_x\left|\vec{V}\right| \tag{2}$$

In Y-direction

$$\frac{1}{\varepsilon^2}\left(V_x\frac{\partial V_y}{\partial x}+V_y\frac{\partial V_y}{\partial y}+V_z\frac{\partial V_y}{\partial z}\right)=-\frac{\partial p}{\partial y}+\frac{1}{\varepsilon\,Re}\left(\frac{\partial^2 V_y}{\partial x^2}+\frac{\partial^2 V_y}{\partial y^2}+\frac{\partial^2 V_y}{\partial z^2}\right)-\frac{1}{Da\,Re}V_y-\frac{F}{\sqrt{Da}}V_y\left|\vec{V}\right| \tag{3}$$

In Z-direction

$$\frac{1}{\varepsilon^2}\left(V_x\frac{\partial V_z}{\partial x}+V_y\frac{\partial V_z}{\partial y}+V_z\frac{\partial V_z}{\partial z}\right)=-\frac{\partial p}{\partial z}+\frac{1}{\varepsilon\,Re}\left(\frac{\partial^2 V_z}{\partial x^2}+\frac{\partial^2 V_z}{\partial y^2}+\frac{\partial^2 V_z}{\partial z^2}\right)-\frac{1}{Da\,Re}V_z-\frac{F}{\sqrt{Da}}V_z\left|\vec{V}\right| \tag{4}$$

Energy equation of the fluid in the porous medium (LTNE)

$$\left(V_x\frac{\partial \mathrm{T}_f}{\partial x}+V_y\frac{\partial \mathrm{T}_f}{\partial y}+V_z\frac{\partial \mathrm{T}_f}{\partial z}\right)=\frac{(1+k_d)}{\mathrm{Pr}\,Re}\left(\frac{\partial^2 T_f}{\partial y^2}+\frac{\partial^2 T_f}{\partial z^2}\right)+\frac{h_{sf}a_{sf}}{K_{fe}\,\mathrm{Pr}\,Re}\left(T_s-T_f\right) \tag{5}$$

3.4 Mesh Generation

FLUENT software is a finite volume numerical method. The solution by numerical methods requires meshing for the geometry. The accuracy of the solution depends on the size of the meshes (cells). The small mesh size introduces more accuracy solution, but whenever the mesh size decreases the number of meshes increase. The result is that the capacity of the computer is large in addition to the time factor, where increasing the number of cells the researcher needs a lot of time. Therefore, the researcher should provide a compromise for this problem because reducing the size of cells requires a large memory of the computer that is used. In FLUENT software there is two types of grids consist of the tetrahedron or hexahedron cells (or a combination between them) in 3-D. The selection of the mesh depends on the type of the application. For applications that involve the complex geometry, the creation of structure or block-structure grids consisting of hexahedron cells, it can be a waste of time. If not impossible, the time of the setup is considered the main motivation for using unstructured- grids that it contains tetrahedron cells.

3.5 The Optimum Mesh Size Procedure

For to investigate the spatial accuracy of mesh, five various meshes were constructed for three-dimensional simulation using ANSYS software (R19.0), as explained in Table 2. After simulation of the models for all above-mentioned meshing sizes the maximum temperature is observed. For the two last meshes mesh 4 and mesh 5, the prediction of the temperatures is observed to be very close in contrary to the other three kinds. Thereby the mesh M4 has been chosen for the simulation, which it is suitable, adequate for capturing the flow properties inside the domain and the fins as shown in figure (2).

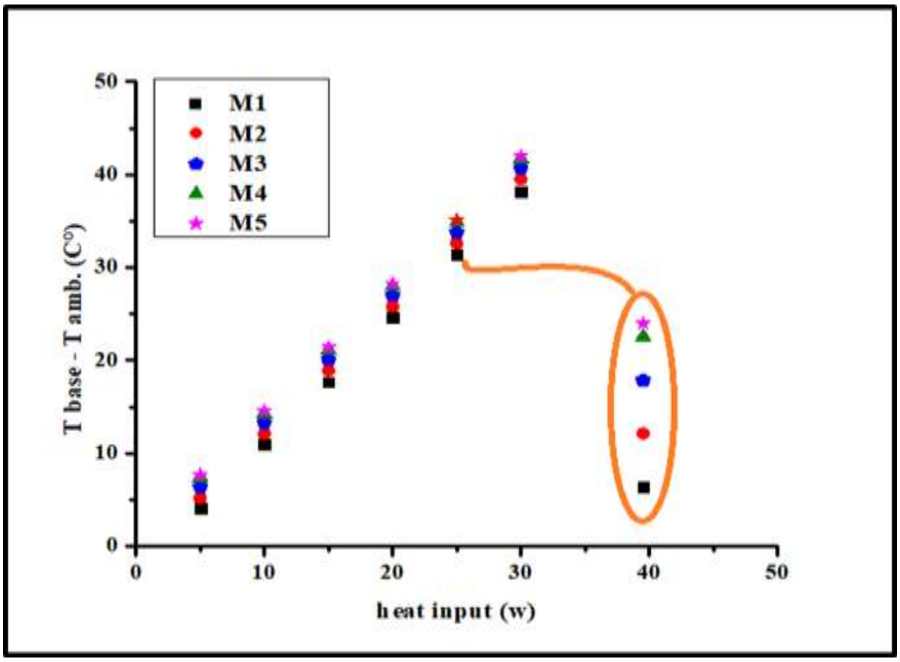


Fig. 2: Mesh independency of numerical model.

Table (2): Overview of mesh models.

Mesh	**M 1**	**M2**	**M3**	**M4**	**M5**
Total Mesh Cells	884,155	1,585,513	3,682,834	6,332,955	8,430,959

3.6 Boundary Conditions

In the beginning, flow conditions and other thermal variables are determined on all aspects of the mathematical model being simulated. The identification of these conditions is an important part of the simulation process so it is necessary to identify them appropriately as shown in figure (3a,3b). The boundary conditions that it is applied in this study, the boundary condition applied :

1. At the entrance of the enclosure is the inlet velocity as (1, 2, 3, 4 and 5) m/s
2. With respect to the top of the enclosure, the pressure outlet is applied at the outlet of the fin array as a boundary condition.
3. The setting of other sides of an enclosure is set as pressure outlet also, this meaning the foam fins are exposed to the effect of the atmospheric pressure
4. For the type of flow has been applied as a turbulent flow in numerical simulation of this study.
5. There are various values of power input (5, 10, 15, 20, 25 and 30 W) have been used in the present work.
6. A very soft mesh is used especially in areas near the fin surface to capture flow behavior with extreme accuracy.

7. The major condition of the fins model at the base plate is a constant heat flux.

8. There are two kinds of heat transfer occur, the first one: in the fin base is transferred by conduction and the second at the surface of the fins is carried out by convection.

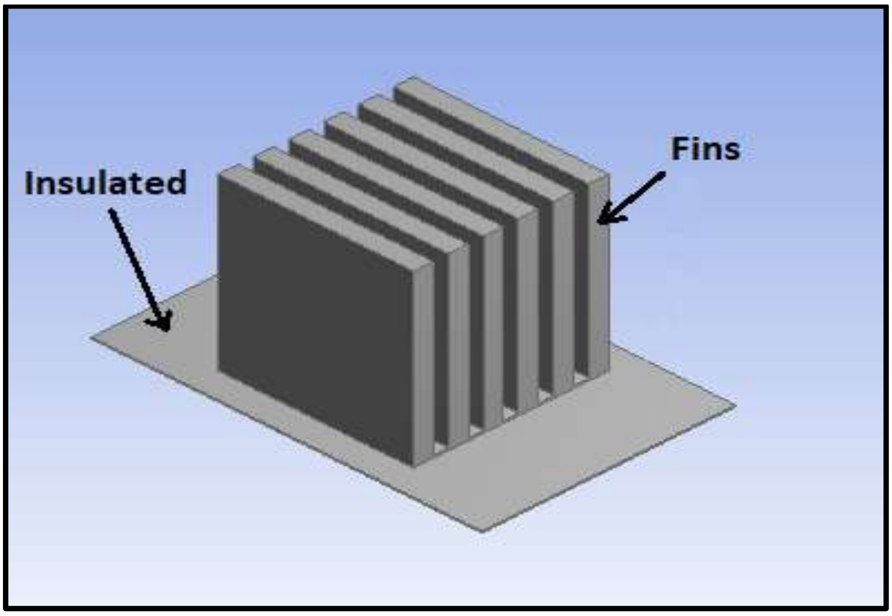


Fig. 3.b: Boundary condition for the domain

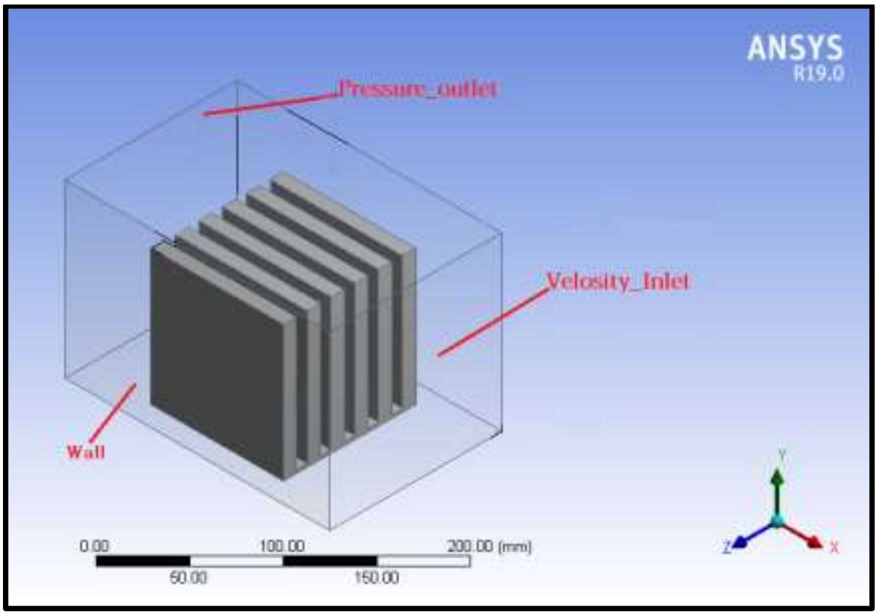


Fig. 3.b: Boundary condition for the domain

Results and Discussions

4 Numerical Results

Determining the number of metal foam fins has an important effect to improving the heat transfer and raising the efficiency of the heat sink, thus, it is necessary to choose best number of fins for the application used while considering all factors of economic, ease of operation, cost, lightweight, and availability. In this section, the effect of varying fins number on the heat sink performance is introduced in aditional to change the heat sink orientation.

4.1 Effect of fins number on base temperature: The influence of fins number variation on temperature difference is shown in figure 4 at various heat input values. It is shown from the figure that the metal foam fins number has an influential role on the heat transfer performance. As generally observed in the figure, as the number of fins rose, the base temperature decreased. This can be attributed to the increase of heat sink surface area.

The influence of increasing fins number becomes clearer when the heat input increases.

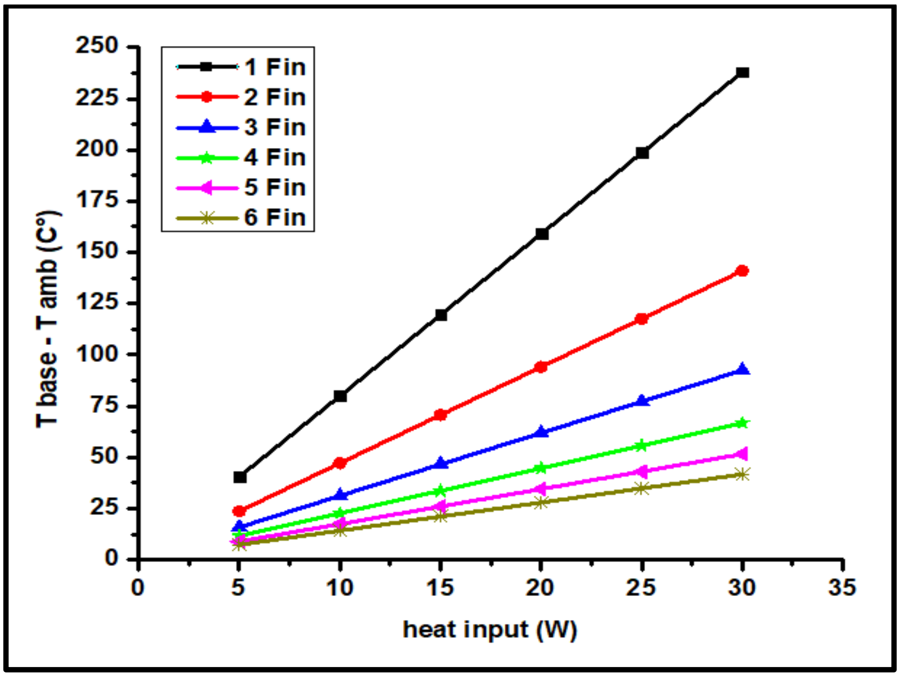


Fig. 4: Effect the fins number on base to ambient temperature different oriented parallel to the airflow

The effect of fins number oriented perpendicularly to the airflow was also studied and showed a same behavior as in figure (5). Generally, the base temperature in the case of parallel orientation is smaller than in the case of opposite orientation with a ratio acceded (95 %).

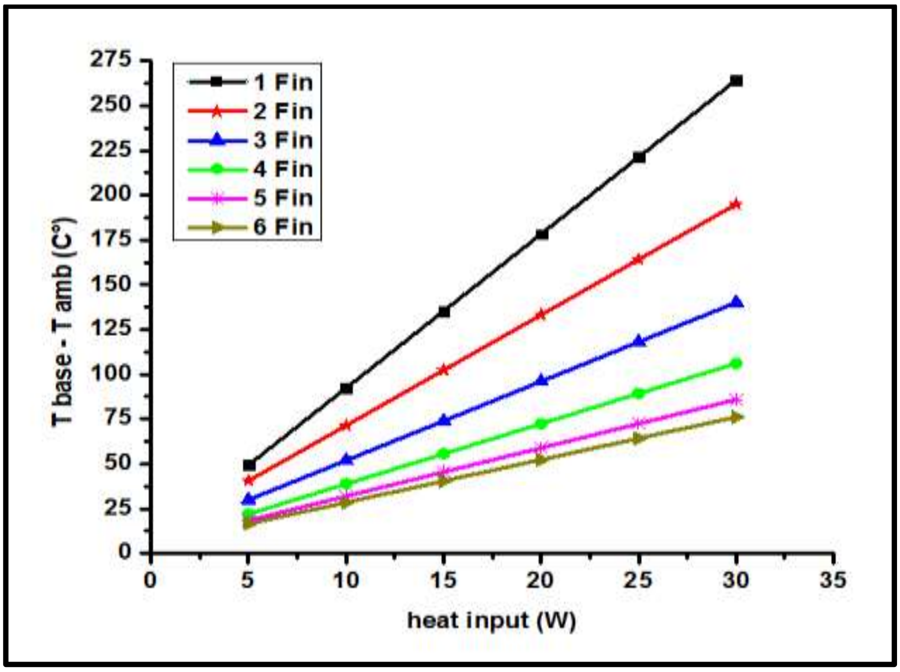
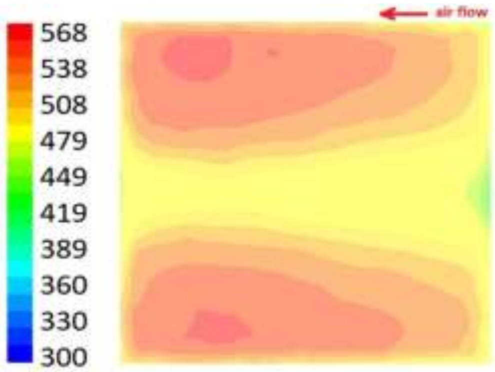


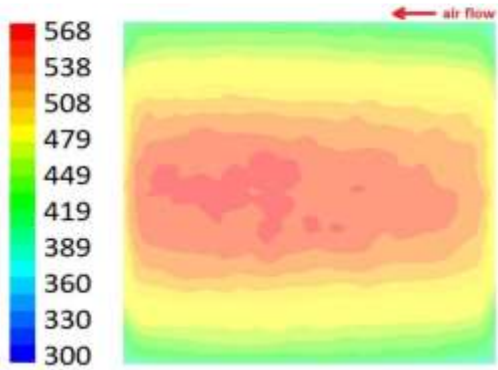
Fig. 5: Effect the fins number on base to ambient temperature different oriented perpendicularly to the airflow

For a further clarification for the influence of fins number on the base temperature, the visualization of temperature contours are introduced. As a result, six arrangements of fin number were analyzed in order to determine which is suitable for this investigation, as illustrated in figure (6). Generally, the increase of fins number leads to reduce the concentration of temperature in the base plate and causes for significant decreasing in base temperature.

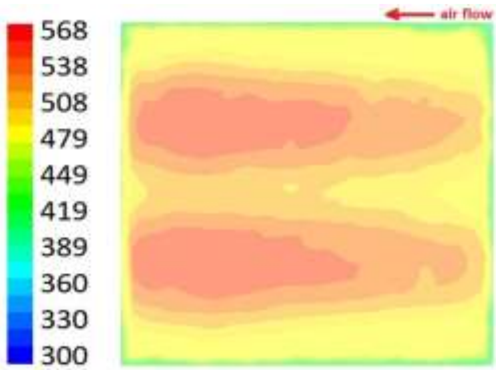
Overall, the optimum number of metal foam fins in terms of best base temperature reduction, was six fins.



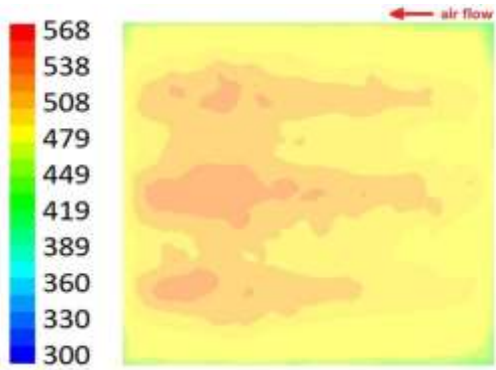
(a)
ase with one fin



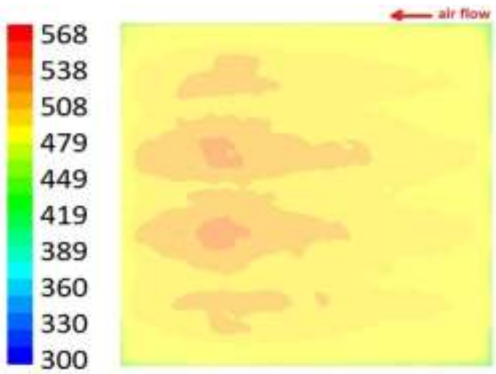
(b)
ase with two fin



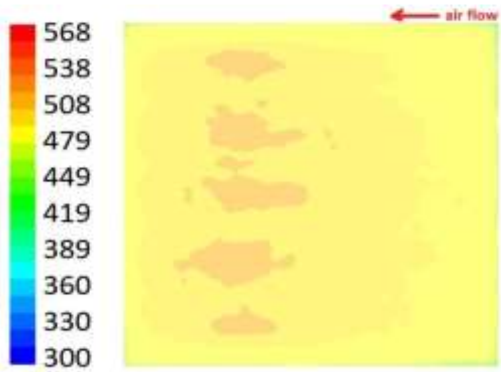
(c) base with three fin



(d) base with four fin



(e) base with five fin



(f) base with six fin

Fig. 5-6: Temperature contours (k) of heat sink base with 30 W various Fins number (a), (b), (c), (d), (e) , and (f) in the parallel orientation for 3 m/s airflow velocity

4.3 Effect of Fins number on Nusselt number: The Nusselt number is computed at several heat input as seen in figure (7). It is noticed that the Nusselt number rises as the number of the fins increases. This is because the increase of fins number lead to the reduction of base temperature and consequently to the improvement of heat transfer coefficient. In comparison, Nusselt number increased by 4.8 times when the fins number increased from one to six. The effect of fins number oriented perpendicularly to the airflow was also studied and showed a same behavior as in figure (8).

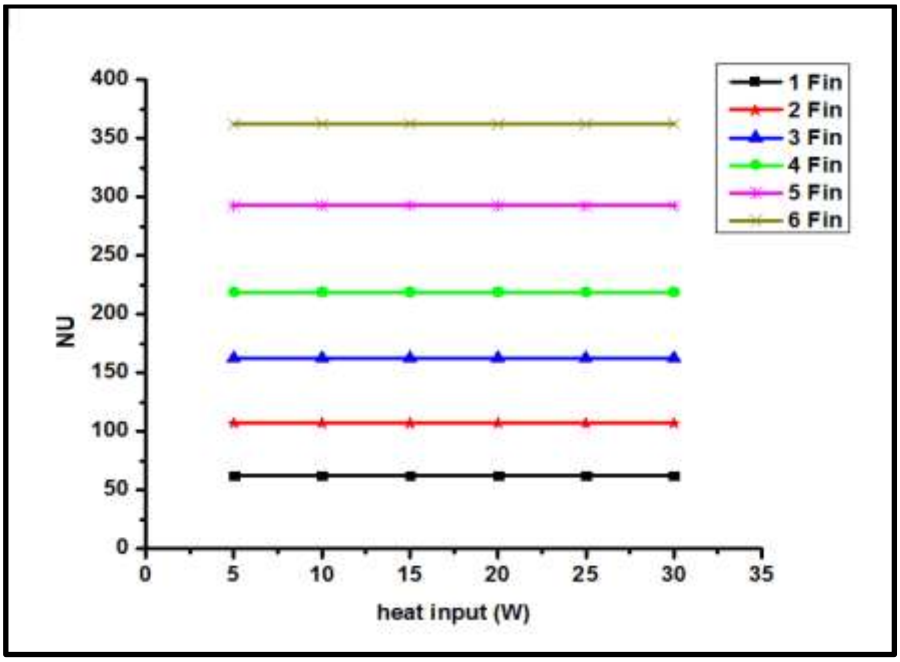


Fig. 7: Effect the fins number on Nusselt number for 3 m/s airflow velocity the heat, sink oriented parallel to the airflow.

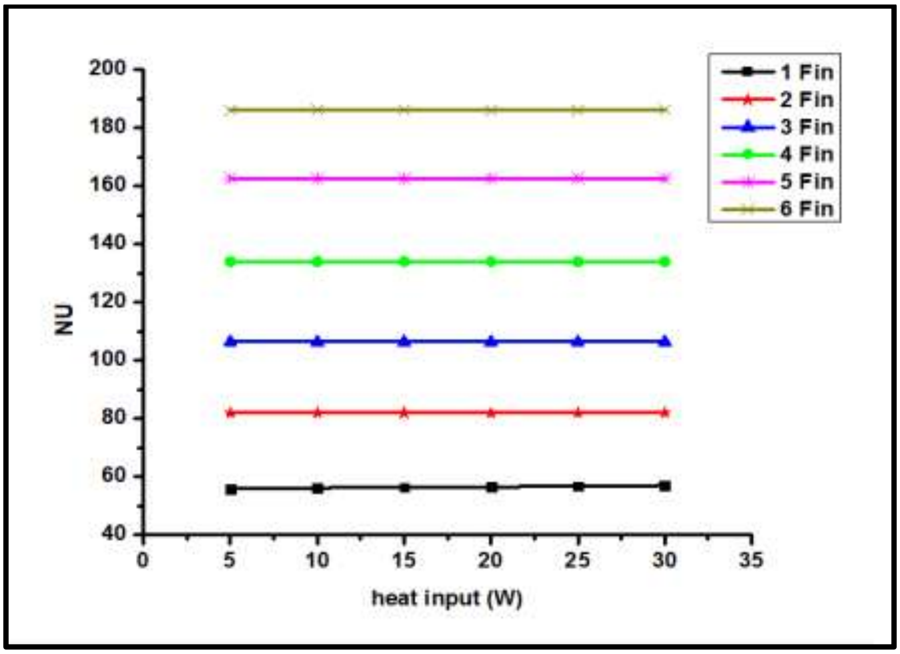


Fig. 8: Effect the fins number on Nusselt number for 3 m/s airflow velocity the heat, sink oriented opposite to the airflow.

4.3 Effect of Fins number and Reynolds number on base temperature: The influence of Reynolds number on temperature difference was evaluated at different fins number (1, 2, 3, 4, 5, and 6) oriented parallel and at 30 W. Figure (9) visualize the variation in temperature difference values at different Reynolds number. The figure shows that temperature difference declines with the increase of Reynolds number especially few number of fins. However, this reduction of temperature difference becomes slight at greater fins number.

This action might be driven by the fact that the ligaments nearest to the outer surface will be increased with increase the fin number and it help to make the overall heat transfer more efficiently than ligaments further away from the outer surface. So the enhancement of the base temperature difference reach about (82.3 %) when used six fins. The effect of fins number oriented perpendicularly to the airflow was also studied and showed a same behavior as in figure (10).

For the same heat sink, the Nusselt number was larger in parallel orientation than in opposite. The primary reason for this behavior that parallel orientation offers lowest flow resistance which allowing air to flow more easily, and contacts with a larger surface area result in improved the performance of the heat sink.

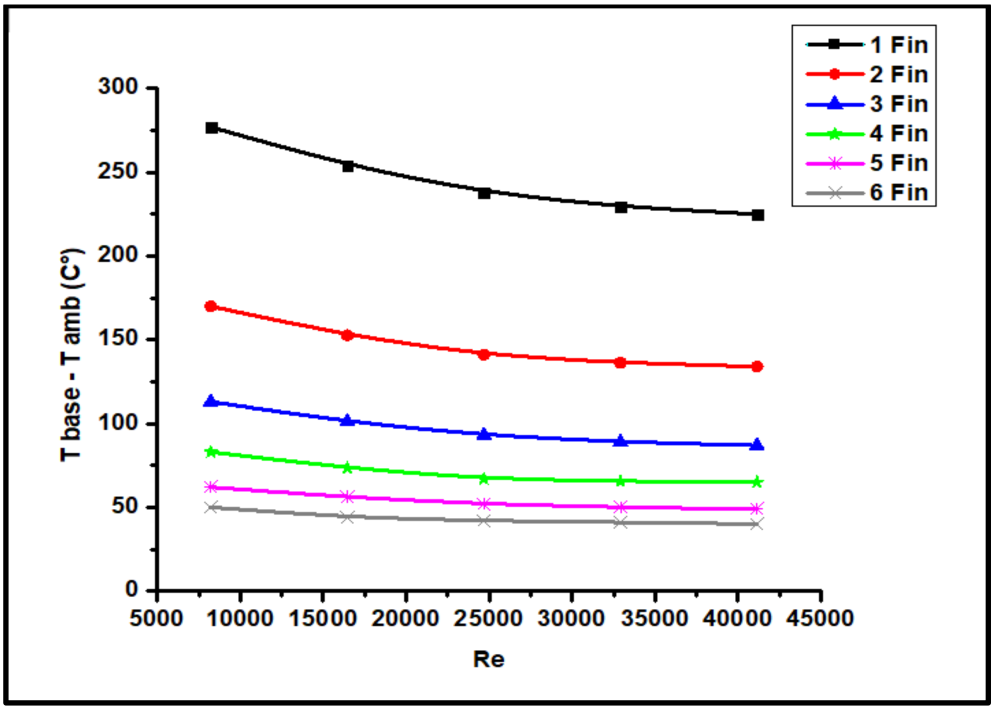


Fig. 9: Effect of Reynolds number on base to ambient temperature difference for different fins number placed in the parallel orientation

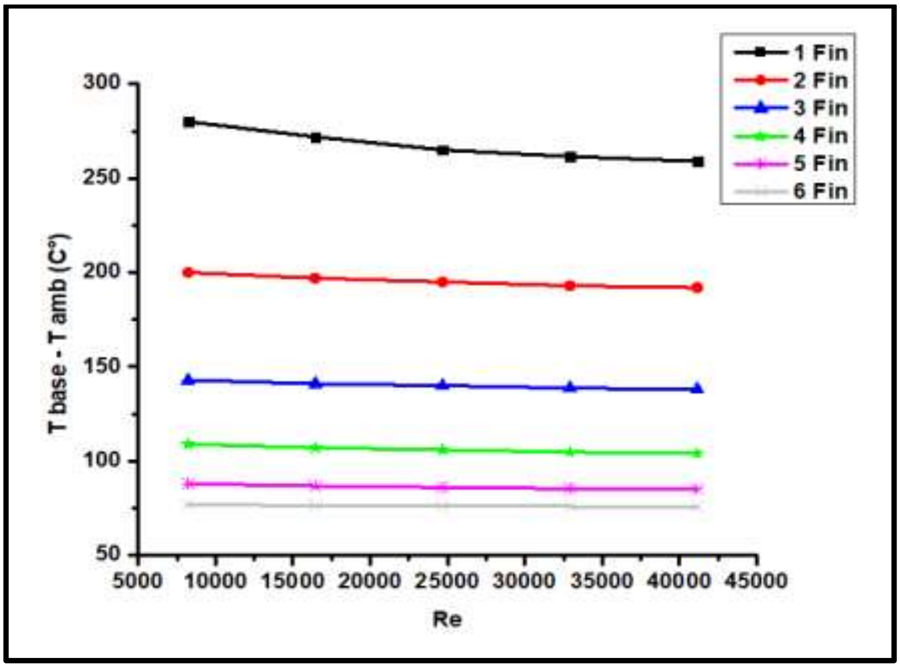


Fig. 10: Effect of Reynolds number on base to ambient temperature difference for different fins number placed in the opposite orientation

4.4 Effect of Fins number and Reynolds number on Nusselt number:

Figure (11) illustrates the variation of Nusselt number the number of fins and Reynolds's number (i.e. ranges from (8.2 103 to 4.1*103) and constant heat input of 30 W. It appears that increasing the Reynolds number causes an increase in the Nusselt number, because the inertia force effect is greater than viscous force. it is also observed that the Nusselt number increases as the number of fins increase, resulting in a decreasing temperature difference. This is because increasing the fins number leads to the growing of dissipation surface area and consequently increasing the heat transfer coefficients. Thus, an improvement ratio exceeding

(4.6) time when using six fins compared to one fin. The effect of fins number oriented perpendicularly to the airflow was also studied and showed a same behavior as in figure (12).

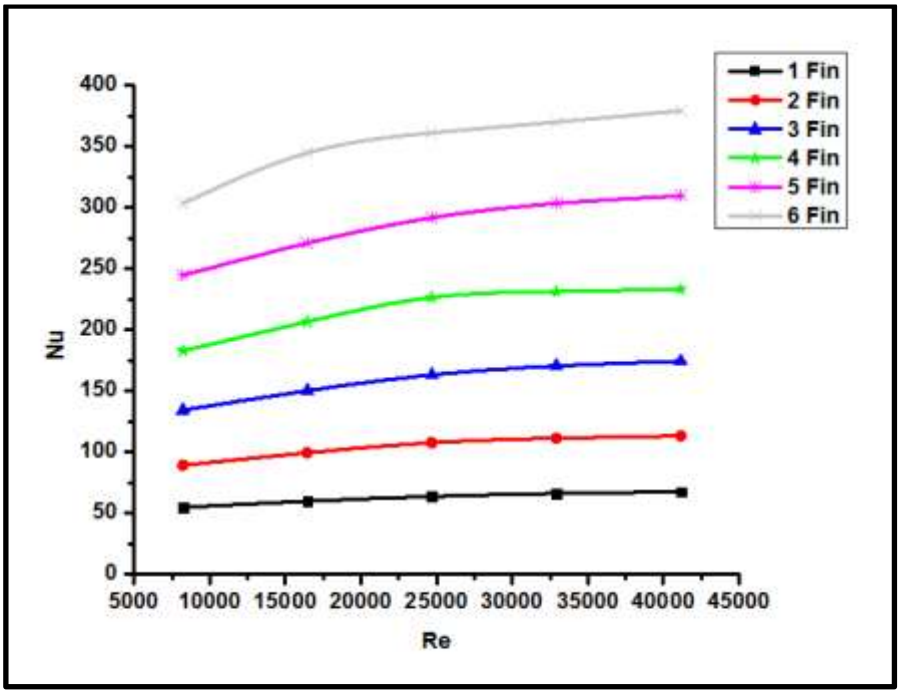


Fig. 11: Effect of Reynolds number on Nusselt number for different number of fins placed in the parallel orientation

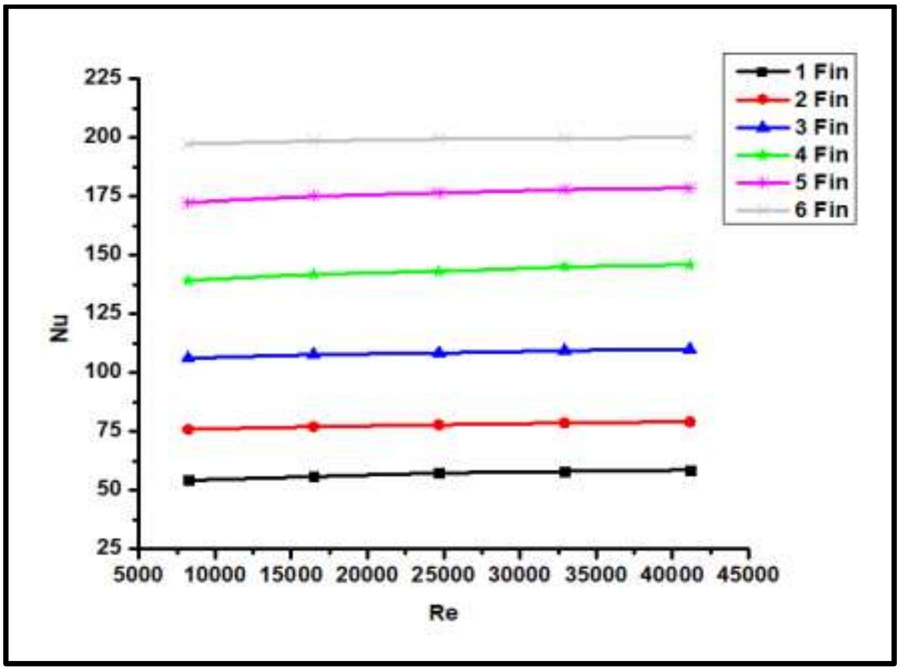


Fig. 12: Effect of Reynolds number on Nusselt number for different number of fins placed in the opposite orientation

4 Experimental Results

This section presents and discusses the results obtained from the experimental work. The experimental results covered the influence of airflow velocity variation on the temperature difference, heat transfer coefficient and Nusselt number at various heat input. For obtaining these results, a copper foam heat sink of six fins, 15 PPI, and porosity of 90% in a parallel orientation is utilized.

5.1 Effect of airflow velocity on base temperature: The influence of airflow velocity variation on the temperature difference was calculated. A velocity range of 1, 2 , and 3 m/s was used.

Figure (13) illustrates the variation of temperature difference as a function of various heat inputs and airflow velocities. According to this figure, the temperature difference and airflow velocity had an opposite relationship. In other words, as airflow velocity increased, the temperature difference between the base and ambient decreased at all heat inputs. So this s rate of heat transfer rise as increases the airflow because of increasing the turbulence of the air flowing through the foam and allowing the air to make contact with a larger surface area result in improved the performance of the heat sink. The maximum rate of reduction in base temperature with 3 m/s is 19.2 % compare with the 1 m/s of the airflow.

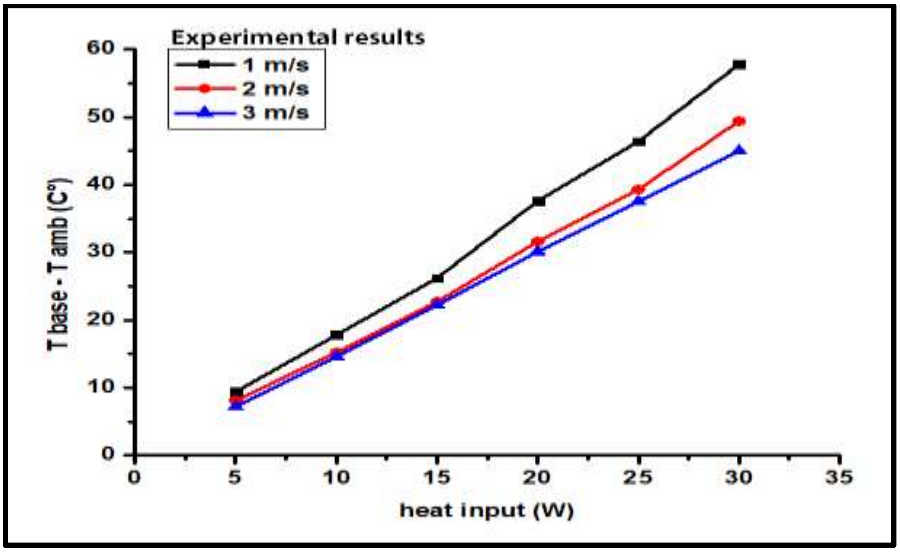


Fig. 13: Variation of average base temperature to ambient with heat Input and different airflow velocity (1, 2 and 3 m/s)

5.2 Effect of airflow velocity on heat transfer coefficient: The variation of heat transfer coefficient was also investigated by applying a variety of heat inputs ranging from (5 to 30) W and three different airflow velocities (1, 2 and 3 m/s) to a set of six cooper foam fins oriented parallel to the airflow. As shown in Figure (14), increasing airflow velocity tends to increase heat transfer coefficients, while the heat input has no effect on heat transfer coefficients for all velocities. Basically, when increase the airflow then flow resistance will reduce, resulting in better flow mixing, which improve the heat transfer of the heat sink and rise the heat transfer coefficient. Whereas the greatest value of improvement achieved with the best airflow was 27.6 %.

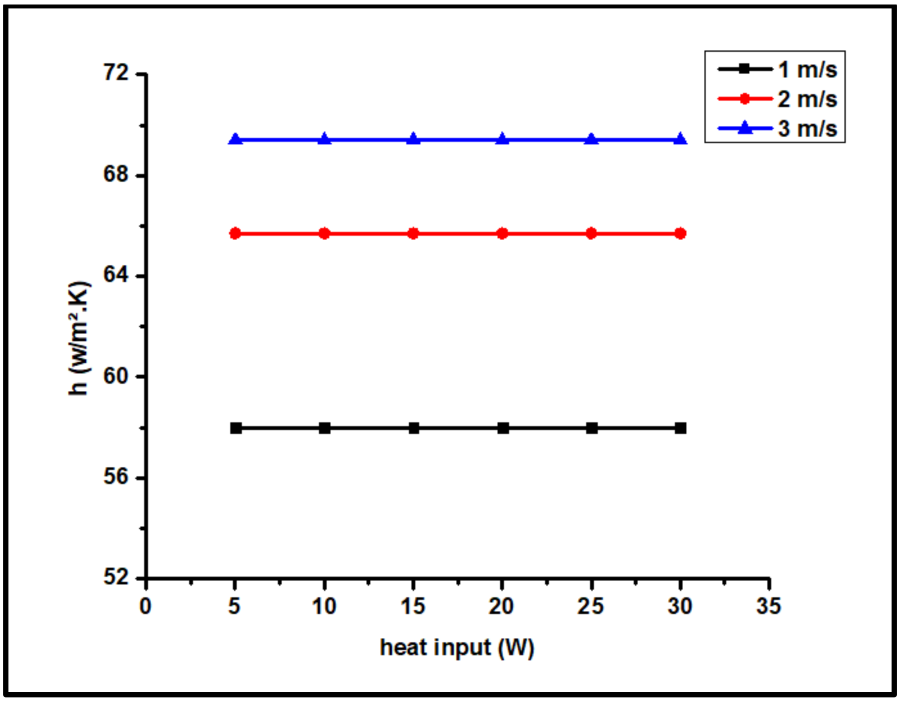


Fig. 14: Variation of heat transfer coefficient with heat Input and different airflow velocity (1, 2 and 3 m/s)

5.3 Effect of airflow velocity on Nusselt number: Figure (15) shows the correlation between the heat input and the Nusselt number when the airflow velocities varied from (1to 3 m/s) at a range of heat inputs of (5, 10, 15, 20, 25 and 30 W) for a six fins heat sink in parallel orientation. The highest value of the Nusselt was found to be at 3 m/s. Because the Nusselt number is directly proportional to the airflow velocity, as the airflow velocity increases, so does the Nusselt number. Furthermore, the metal foam with a velocity of 3 m/s has a higher heat transfer coefficient among the others and the lowest base to ambient temperature.

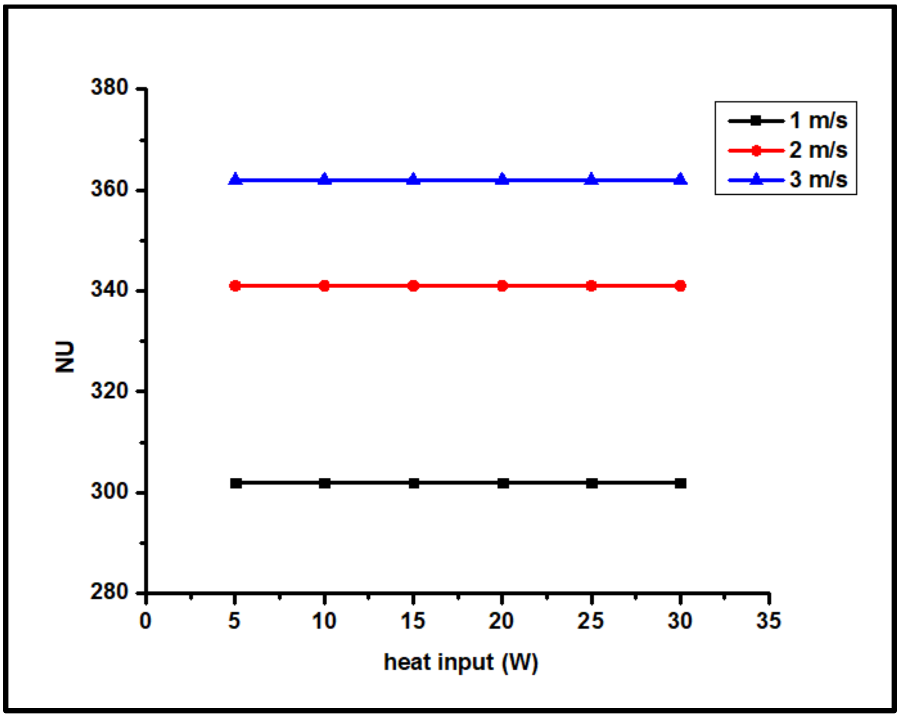


Fig. 15: Variation of Nusselt number with heat Input and different airflow velocity (1, 2 and 3 m/s)

5. Comparison of the numerical and experimental results:

In this section, the numerical results were compared to the experimental data. For varied values of heat input, the numerical average base temperature of the metal foam heat sink was evaluated and verified with the experimental results at the same geometry and conditions of external ambient temperature and airflow velocity. The average base temperature agrees very well with the experimental average temperature determined in the real heat sinks, and the agreement was about 4.1 % for 3 m/s, as seen in figure (16).

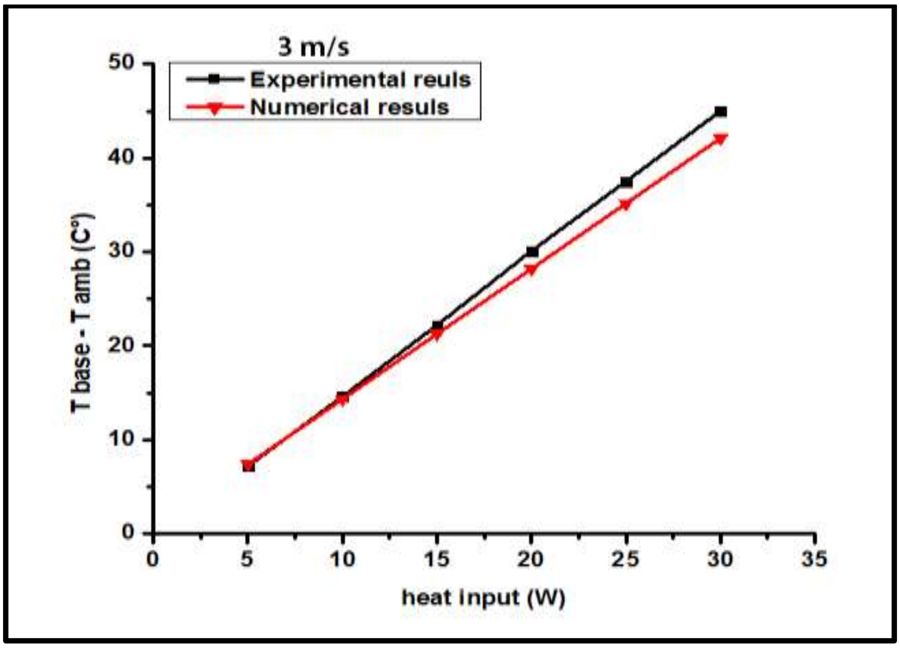


Fig. 16: Comparison between numerical and experimental results for the airflow velocity of 3 m/s

5 Conclusions

The purpose of this chapter is to evaluate to which extent of the current study's aims were carried out, based on the numerical and experimental results acquired for this work. The following points are the main findings of the current study:

Effect of airflow velocity:

The base temperature decreases as the airflow velocity increases. As a result, the heat sink with a velocity of 5 m/s recorded higher improvement in the reduction of average base temperature. This improvement is close to those achieved at 3 and 4 m/s airflow velocity. On the other hand, the improvement was minimum at 1 m/s. The enhancement of heat transfer coefficient at 5 m/s is estimated by 23.7 % higher than that at 3 m/s which is only 18.6 % compared to the heat sink with 1 m/s while 2 m/s had an average improvement of 11.8 %.

Effect of Fins number:

The number of fins plays an important role in enhancing the heat sink performance throughout reducing the base temperature and improving the heat transfer coefficient. When the fins number of heat sink increased from one to six, the base temperature improves by 82.2 %. The highest improvement in Nusselt number achieved by six fins is approximately 464 % larger than a single fin.

Effect of heat sink orientations

For the same heat sink dimensions, the optimum orientation for the heat sink is to be parallel to the airflow. For this orientation, the temperature difference becomes smaller in comparison to those in the opposite direction. The largest rate of decreasing in base temperature with parallel orientation and six fins is 48.7 % when compared to the opposite orientation, while the lowest rate of decreasing with one fin in parallel orientation is around 12.3 % when compared to the opposite direction. However, only one fin of 40 mm thickness in the opposite direction has a rate of decrease that is around 1.3% greater than in the parallel orientation.

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