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An Investigation of the Effect of interrupted fin arrangements on the Thermal Performance of a Heat Sink under a Free Convection Condition

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ABSTRACT: The heat sink with vertically rectangular interrupted fins was investigated numerically in a natural convection field, with steady-state heat transfer. A numerical study has been conducted using ANSYS Fluent software (R16.1) in order to develop a 3-D numerical model. The dimensions of the fins are (305 mm length, 100 mm width, 17 mm height, and 9.5 mm space between fins. The number of fins used on the surface is eight. In this study, the heat input was used as follows: 20, 40, 60, 80, 100, and 120 watts. This study focused on interrupted rectangular fins with a different arrangement and angle of the fins. Results show that the addition of interruption in fins in various arrangements will improve the thermal performance of the heat sink, and through the results, a better interruption rate as an equation can be obtained.

***Key words**— Heat sink, Heat transfer, free convection, interrupted fins, CFD*

التحقق من تأثير ترتيب الزعانف المتقطعة على الاداء الحراري لمشتت الحرارة تحت حالة الحمل الحر

عباس جاسم جبير، علي حميد عبد

الخلاصة:تم فحص بالوعة الحرارة مع الزعانف المستطيلة المتقطعة عمودياً في حقل الحمل الحراري الطبيعي، مع الحالة المستقرة لانتقال الحرارة. وقد أجريت دراسة عددية باستخدام برنامج أنسيس فلونت المستخدم من أجل تطوير نموذج رقمي ثلاثي الأبعاد. أبعاد (R16.1) الزعانف هي (طول 305 ملم، عرض 100 ملم، ارتفاع 17 ملم، وفراغ 9.5 ملم بين الزعانف). عدد الزعانف المستخدمة على السطح ثمانية. في هذه الدراسة، كانت القدرة الداخلة هي على النحو التالي (20، 40، 60، 80، 100، و120 واط). وركزت هذه الدراسة على الزعانف المستطيلة المتقطعة بترتيبات وزوايا مختلفة للزعانف. وتظهر النتائج أن إضافة زعانف انقطاع في ترتيبات مختلفة من شأنها تحسين الأداء الحراري لبالوعة الحرارة، ومن خلال النتائج، حصلنا على معدل انقطاع أفضل كمعادلة.
الكلمات الرئيسية: بالوعة الحرارة، انتقال الحرارة، الحمل الحر، الزعانف المتقطعة، سي اف دي

1. INTRODUCTION

Heat sinks are used in various engineering applications for the cooling of electronic and electrical appliances, communications equipment and vehicle components. These components can be either semiconductor devices or integrated circuits, and more accurately, use passive cooling extensively in CPU cooling, amplifiers, and LED light bulbs. Natural convection heat transfer from vertical rectangular fins is a settled subject in literature. Here it has been investigated numerically. Previous studies include numerical and experimental works, and further studies can be found elsewhere [1].

[2], studied the numerical model by using the ANSYS program in this project. The aim was to reduce the materials used, and make the fins highly efficient, at a relatively low cost. The fin form used was a rectangular fin made of a metallic element in a horizontal position, and exposed to the effect of natural convection. The results showed that when the heat input increased the heat transfer coefficient would increase and the Nusselt number would also increase. [3], investigated the numerical and experimental study of vertically rectangular interrupted fins in a natural convection field. In order to create a two-dimensional digital model, ANSYS fluent was used to study the effect of the interruptions numerically. The aim of this study is to add different interruptions with a variable dimension to the vertical fins and then to find the optimum value of the interruption ratio. The results showed that the optimal

interrupted length of the fin was obtained and related**.** [4], studied several models of interrupted fins with different interruption ratios under the natural convection effect. The governing equations (continuity, momentum, and energy) of the two-dimensional system, with a steady state, laminar flow, and Boussinesq approximation were solved by using the ANSYS-Fluent V15 software. The results showed that addition of the interruption to the fins led to the re-growth of the thermal boundary layer and thus enhanced thermal performance and reduced the total weight. [5], studied different models of the heat sink, which contained continuous, inline-interrupted, and staggered fins, by using the fluent software. The study concluded that interrupted fins were more effective than continuous fins and that staggered fins were better than the rest. The results showed that staggered fins could be used to improve the thermal performance of the attachments for a variety of electronic, electrical, and communication applications. [6], explained the heat transfer by free convection for two types of fins, one-interrupted and four-interrupted rectangular fins, by placing them horizontally, vertically, and in an inclined manner. The base of the fin was susceptible to constant heat flux. The COMSOL 5.0 package was used to find the mesh generation and locate the results. The results showed that convection heat transfer of the fin in a horizontal position was less, when compared with the vertical and inclined cases.

The main purpose of this study was to obtain the best heat transfer coefficient and reduce the total weight. A numerical study was carried out by using ANSYS-Fluent R16.1, starting from the continuous fin, with dimensions of 101 mm width, 305 mm length, 2.5 mm thickness, and 9.5 mm space between the fins. Following that, the effect of interrupting the fin with different lengths of interruption (G = 10 mm, 20 mm, and 30 mm) and numbers of interruption (N = 1, 2, 3, 4, and 5) were studied, in order to obtain the best ratio of the interruption (γ). After getting the best number of interruptions, the arrangement was changed from inline interruption to staggered interruption. Finally, the angle of inclination of the fins was changed.

2. PROBLEM STATEMENT

When the fin base plate is heated from the bottom, the heat passes through conduction to the vertical fins and the thermal boundary layers begin to evolve from the lower edges of the fins. If the fins are long enough, the thermal boundary layers will be fully integrated, Bejan, 1984. The interruptions of the fin disrupt the growth of the thermal boundary layer, even as they maintain a thermal flow system, leading to a higher natural heat transfer coefficient. Three-dimensional (CFD) models with dissimilar patterns of heat sinks have been presented in this study.

3. NUMERICAL ANALYSES

This section includes discussion of the following sub-sections, computational domain, the governing equations, mesh generation, and precondition. In addition, validation of the present numerical study and another previous numerical study has been discussed.

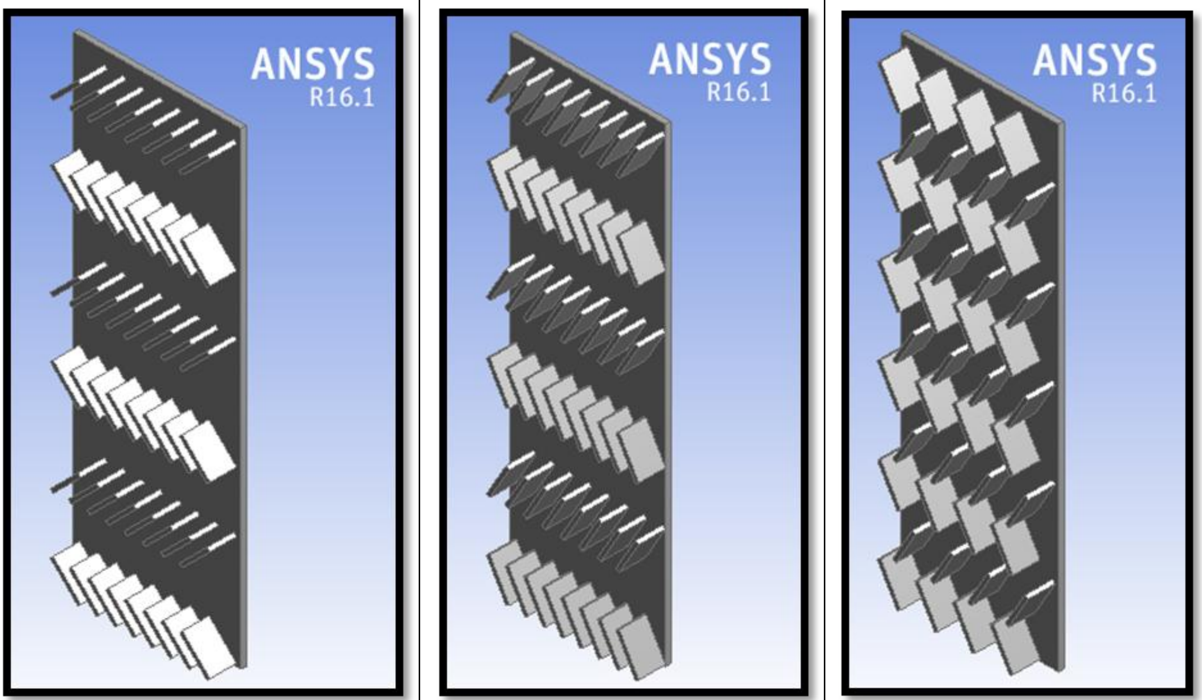
3.1 Computational Domain

Different samples of fins have been modeled as shown in Table 1. The main species of fins (heat sinks) geometries are shown in Figure 1.

Table 1: Dimensions of finned plate samples; interrupted and staggered fins

Sample name	N (number of interruptions)	G (mm)	*l* (mm)	n (number of fins)
Int.-1-10	1	10	147.5	16
Int.-2-10	2		95	24
Int.-3-10	3		68.75	32
Int.-4-10	4		53	40
Int.-5-10	5		42.5	48
Int.-1-20	1	20	142.5	16
Int.-2-20	2		88.33	24
Int.-3-20	3		61.25	32
Int.-4-20	4		45	40
Int.-5-20	5		34.2	48
Int.-1-30	1	30	137.5	16
Int.-2-30	2		81.6	24
Int.-3-30	3		53.75	32
Int.-4-30	4		37	40
Int.-5-30	5		25.83	48
Stag.-5-10	5	10	42.5	44
Stag.-5-20	5	20	34.2	44
Stag.-5-30	5	30	25.83	44

Int.-5-30 zigzag (45^o)	5	30	25.83	48
Int.-5-30 zigzag (60^o)	5	30	25.83	48
Stag.-5-30 zigzag (60^o)	5	30	25.83	44



Int.-5-30 angle (45^o) fin arrangement	Int.-5-30 angle (60^o) fin arrangement	Stag.-5-30 angle (60^o) fin arrangement

Figure1: The considered heat sink Geometries

3.2 Governing Equations

The dominant equations for the continuity, momentum, and energy have been resolved by using the FLUENT R16.1 software. The processors of numerical study are divided into two parts. The preprocessing begins with the program structure that creates the geometry and grid by using the FLUENT R16.1 software. Post-processing comprises of resolving governing equations, which include conservation of mass, momentum, and energy for the purpose of obtaining the results [7].

The governing equations for this study are as follows:

Mass conservation equation:

$$\nabla . V = 0 \tag{1}$$

Momentum:

$$(V . \nabla) V = -\left(\frac{1}{\rho}\right) \nabla P + \upsilon \nabla^2 V + g\beta(T - T_o) j \tag{2}$$

Energy:

$$(V \cdot \nabla) T = \alpha \nabla . (\nabla T) \tag{3}$$

The heat in the fins is dissipated by three methods, conduction, convection, and radiation. The temperature field is gained by settling the energy equation.

3.3 Mesh Generation

For the choice of meshing size, proximity and curvature size operations are taken into account. For higher meshing at the fluid–metal interface, the inflation layer is taken into account. The mesh independence study is performed for this study. Figure 2 shows two different meshing models of a numerical model, which is created using ANSYS Fluent R16.1 software. The first model is for meshing with a minimum cell size of 1 mm, even as the second model shows mesh generation using 0.5 mm minimum cell size. After numerical simulation of the fins for both grid sizes, the maximum temperature is determined. Our result shows that selection of a meshing size less than 1 mm has very less effect on temperature variation. For both grid sizes the maximum temperature is about 159.4°C. Finer meshing only increases the computational cost without further changes in maximum temperature, therefore, 1 mm meshing size is selected for all the ANSYS simulations in this study.

(a) (b)

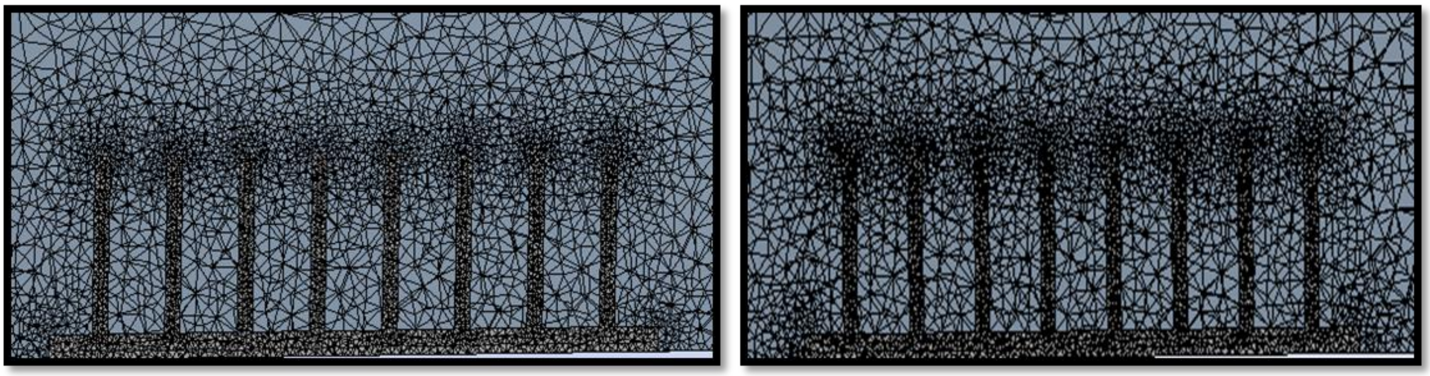


Figure2 Mesh independency of numerical model to thermal performance: (a) meshing with 1 mm minimum cell size; (b) meshing with 0.5 mm minimum cell size.

3.4 Boundary Condition

The flow type in this numerical study is laminar. The effect of density variation on temperature has been estimated by the Boussinesq approximation. In the current study, different values of heat flux (649, 1298, 1948, 2597, 3246, and 3896 W/m^2) have been used. A very soft border layer mesh was used for areas close to the fin surface, to capture the flow behavior with high accuracy. The primary condition of the fin model at the base is in constant heat flux, the heat in the fin base is transmitted by conduction. At the surface of the fins, the heat transfer is carried out by convection and radiation as shown in Figure 3, where the emissivity of aluminum (0.5) has been used.

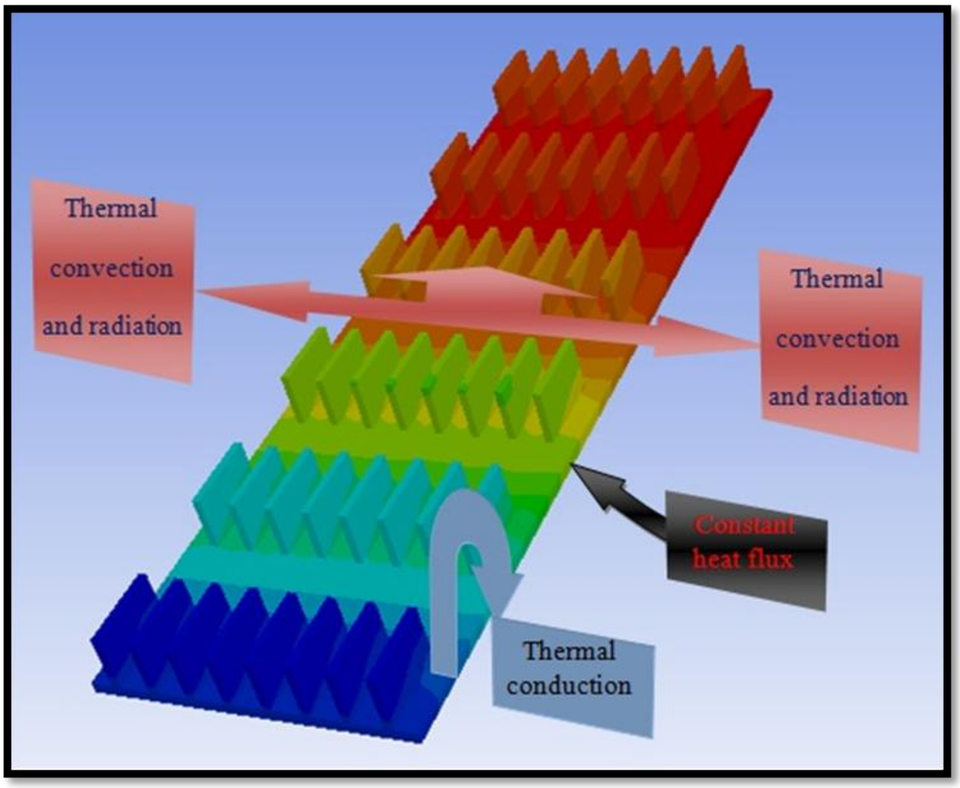


Figure 3 Boundary condition

4. CFD MODELING

In the present study, the fins were placed on a base plate of dimension 305 * 100 mm^2. The fin arrangement was placed in the middle, and thus, the bounding box was created, as shown in Figure 4. The boundaries of this process domain were placed sufficiently far from the fins to avoid any flow-reversal throughout the numerical simulations. The CFD meshing, process domain discretization was performed using the mistreatment ANSYS software, a processor. Combos of tetrahedral and prism parts were used. On account of the high-temperature distinction between the fin surfaces, and therefore, the surroundings, the buoyancy currents were going to be dominant. The prism layers generated close to the fin surfaces would facilitate to numerically resolve the thermal boundary layers and this led to the improved accuracy of the general heat transfer rate.

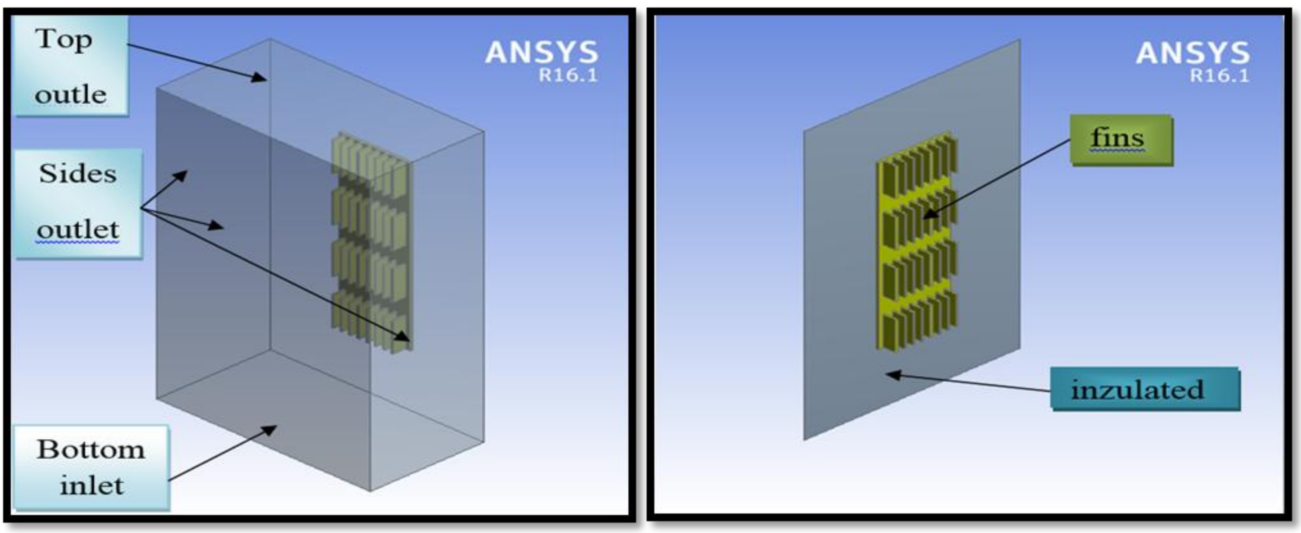


Figure4 CFD modeling.

5. SIMULATION RESULTS

In this section, the most important results for the fins with different arrangements were reviewed.

5.1 Validation of the Present Study:

For the purpose of starting our current numerical study, the comparison has been done with [8]. Through comparison between two studies, it was found that there was a good agreement between the results, and the error rate did not exceed 2.7%. This validation for continuous fins is shown in Figure5.

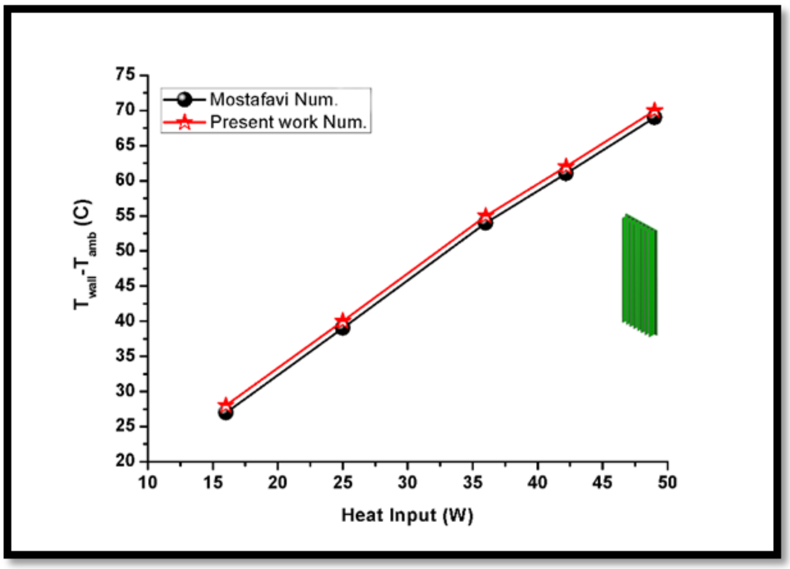
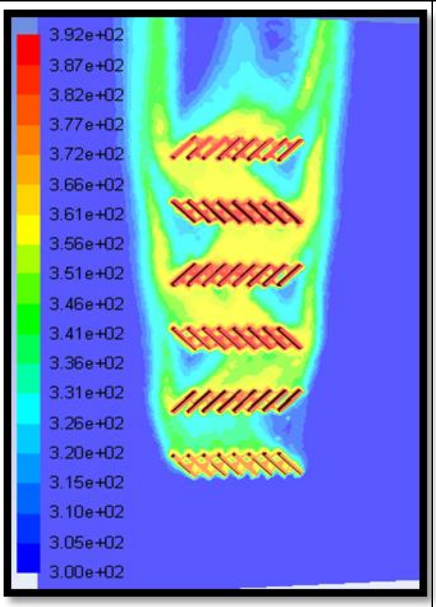
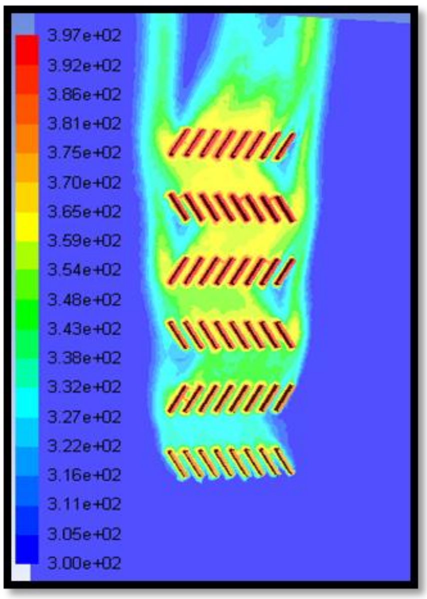


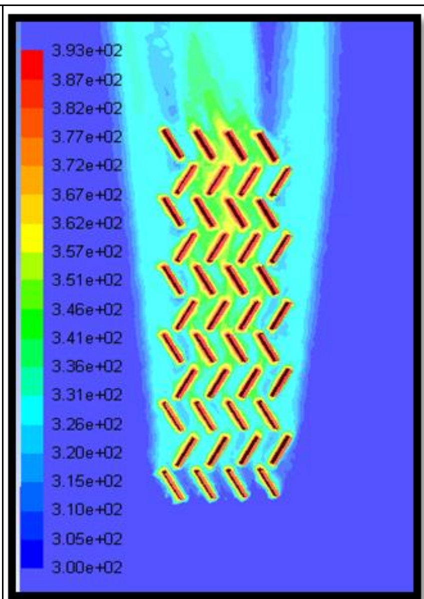
Figure5 Validation of numerical analysis graph.

5.2 Velocity and Temperature Contours:

The main species of a heat sink has been chosen in order to show the effect of changing the arrangement of temperature and velocity through the contour. Figures 6 and 7 show the temperature and velocity flow contours for three models. These contours clearly show the effect of interruption length on the thermal boundary layer. The length of the interruption prevents the growth of the adjacent thermal layer that causes resistance to heat transfer to the ambient. The development of flow in the fin can be seen through the regions of argument. The contour of the temperature and the velocity flow are created at the level of the mid-height of the fin, and at the value of heat input (60 watts).

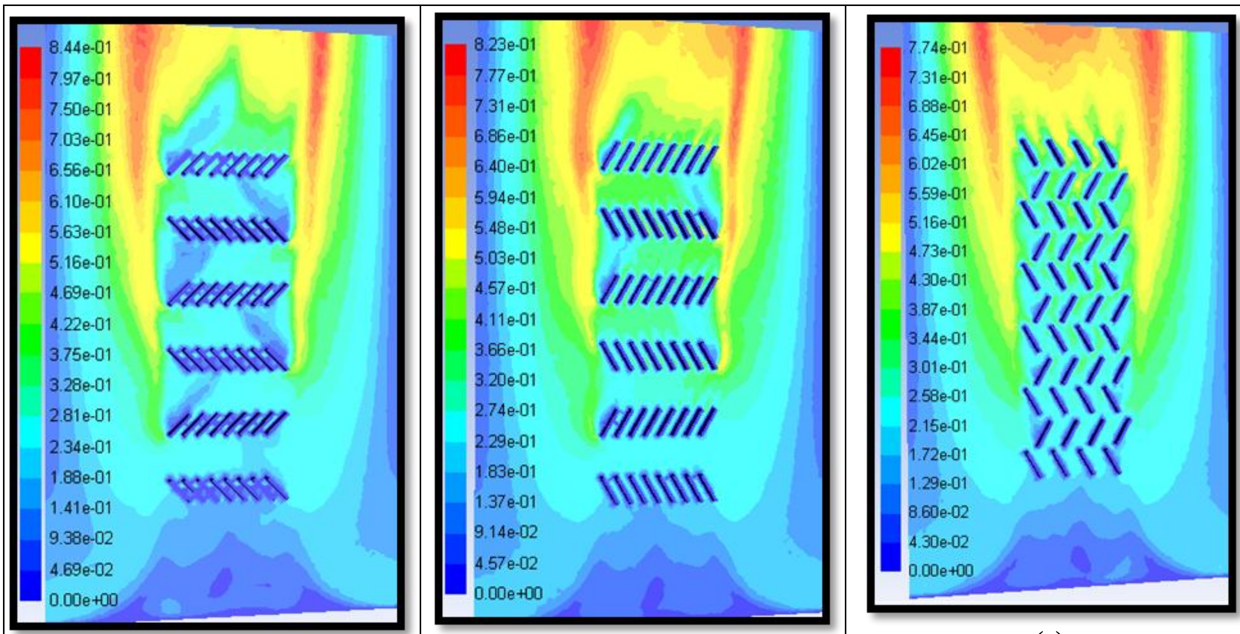






(a)	(b)	(c)

Figure6: Temperature contours; (a) Int.-5-30 zigzag (45°) fins; (b) Int.-5-30 zigzag (60°) fins; (c) Stag.-5-30 zigzag (60°) fins.



(a)	(b)	(c)

Figure7: Velocity contours; (a) Int.-5-30 zigzag (45°) fins; (b) Int.-5-30 zigzag (60°) fins; (c) Stag.-5-30 zigzag (60°) fins

5.3 The Variation between Interruption Ratio and Heat Transfer Coefficient:

The relation between heat transfer coefficient and interruptin ratio (**γ**) with different heat inputs can be seen in Figure 8. The reason for increasing the heat transfer coefficient is because, when the interruption ratio increases, the surface area decreases and the flow velocity increases at the base of the fin, thereby increasing the heat transfer coefficient through natural convection.

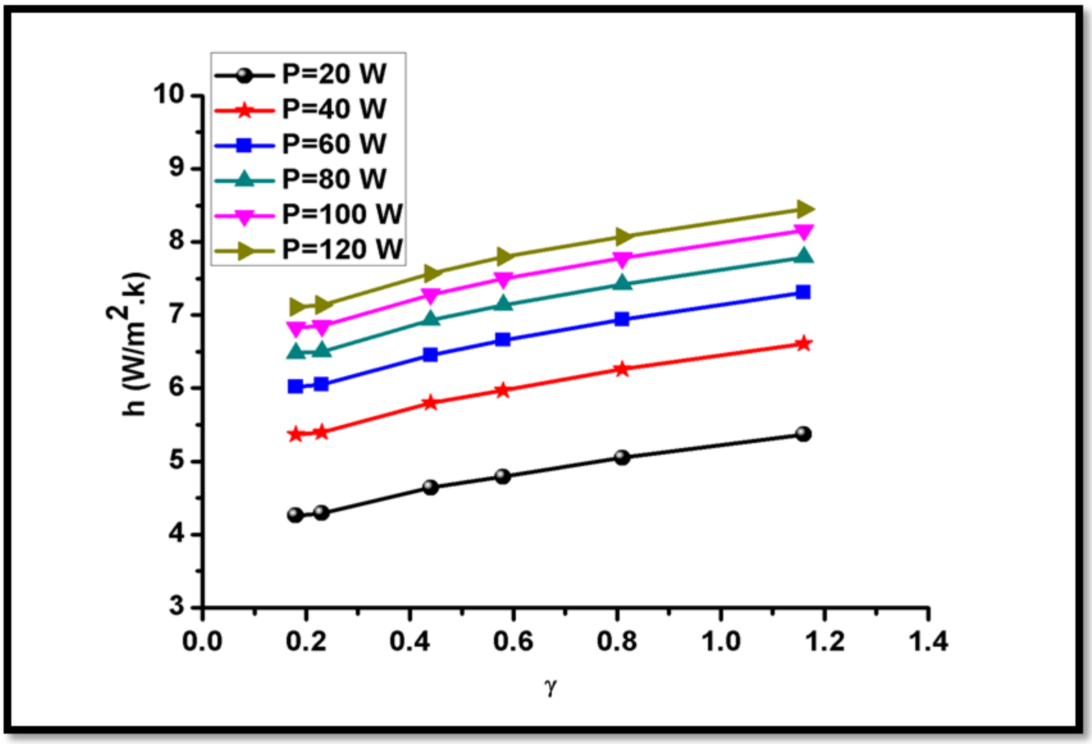


Figure8: The effect of interruption ratio on heat transfer coefficient with different heat inputs

Wasit Journal of Engineering Sciences
VOLUME: (7), NO. : (1) 2019

5.4 The Effect of Different Heat Input on Temperature and Nusselt Number

Using the numerical results obtained, a relationship between the heat flux and the difference in temperature was plotted for the interrupted fins (with different N and G) as shown in Figures 9 to 12. It can be noted from these figures that when increasing the length of the interruption (G) and the number of interruptions (N) lead to increased temperature due to the decrease surface area of a heat sink. This is clear from Figures 13 to 16, which depict the relation between different heat inputs and the Nusselt number with a different G. They exhibit that the heat transfer coefficient enhances when the heat inputs, G and N, increase, which means the interruption length leads to a higher thermal performance, as interrupted fins prevent growth of the thermal boundary layer and lead to an increased heat transfer coefficient and thus increase the Nusselt number with natural convection.

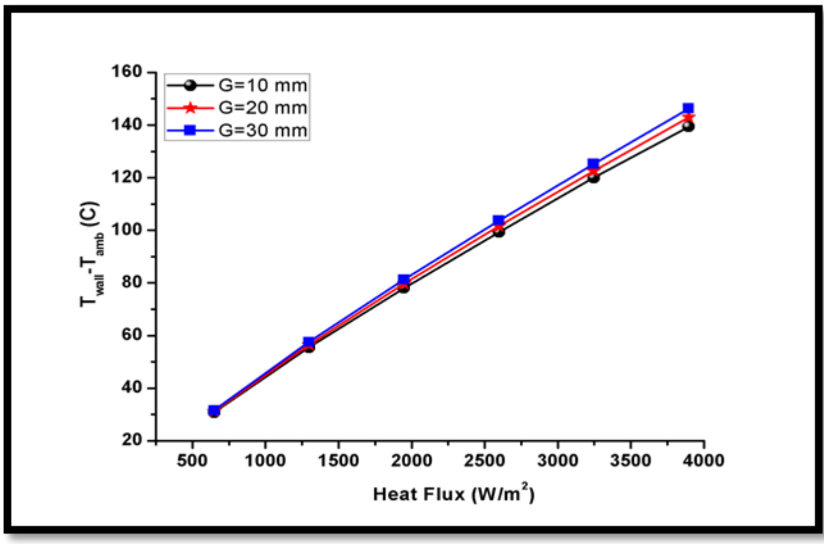


Figure 9: The effect heat flux on temperature is different with different G at N = 2.

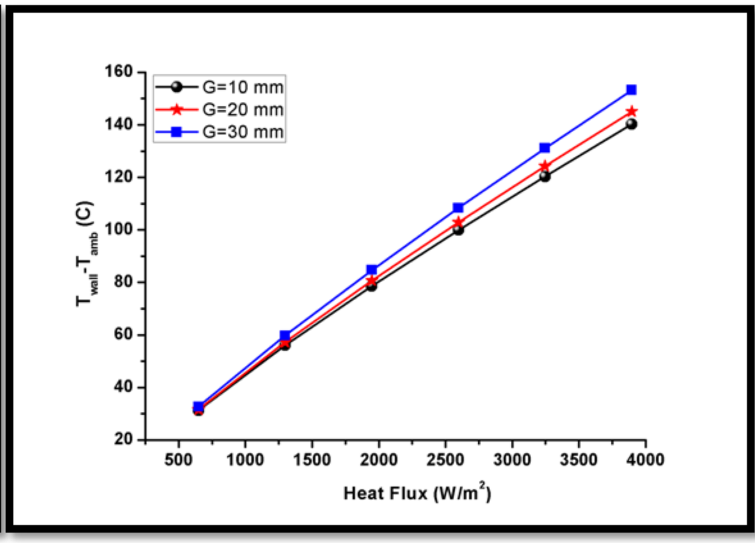


Figure10: The effect of heat flux on temperature is different with different G at N = 3.

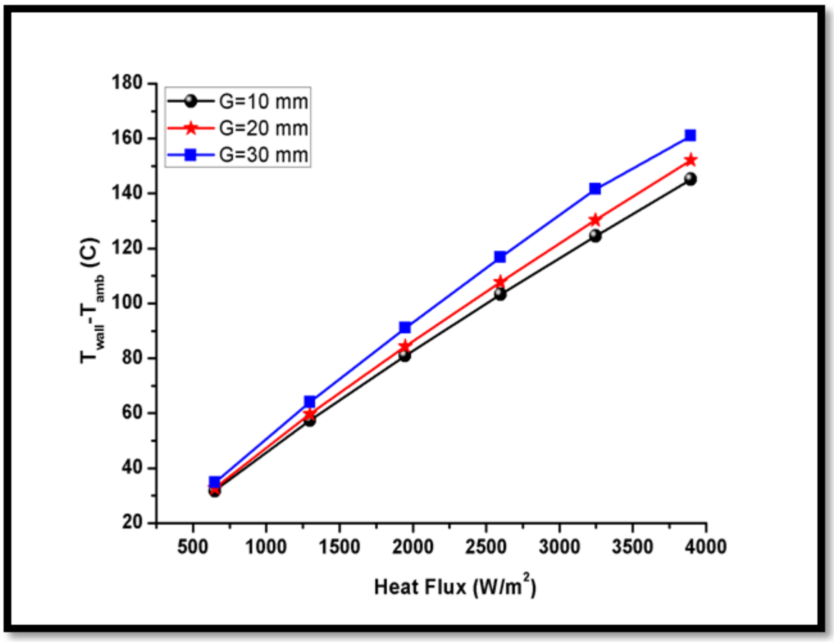


Figure11: The effet of heat flux on temperature is different with different G at N = 4.

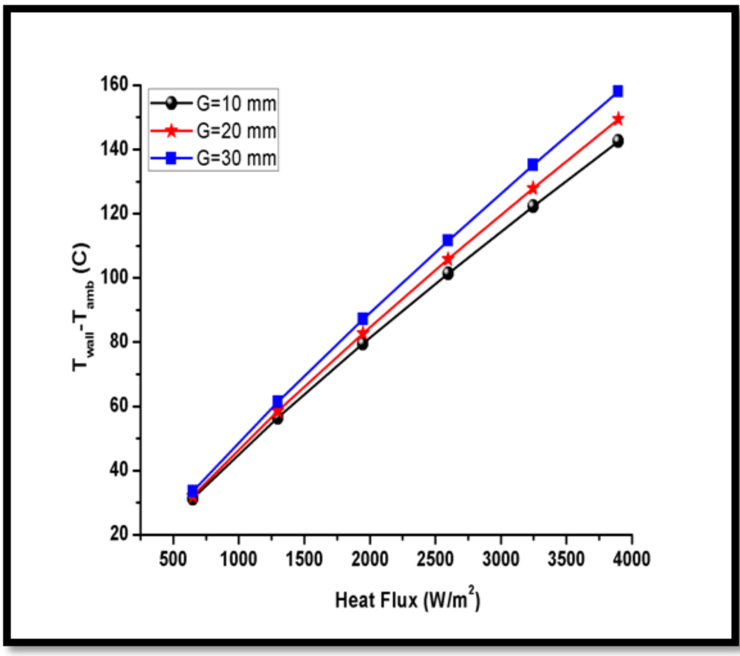


Fig.12: The effect of heat flux on temperature different with different G at N=5.

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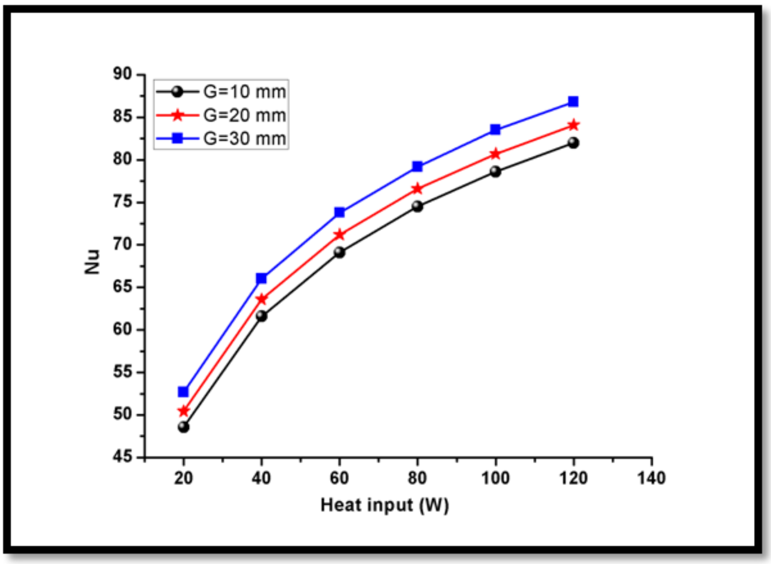

Figure13**:** The effect of heat input on Nusselt number with different G at N=2.

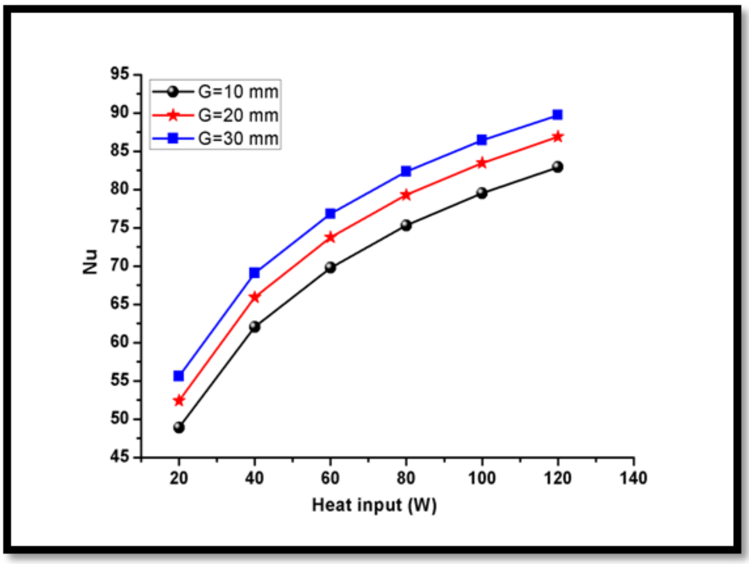

Figure14**:** The effect of heat input on Nusselt number with different G at N=3.

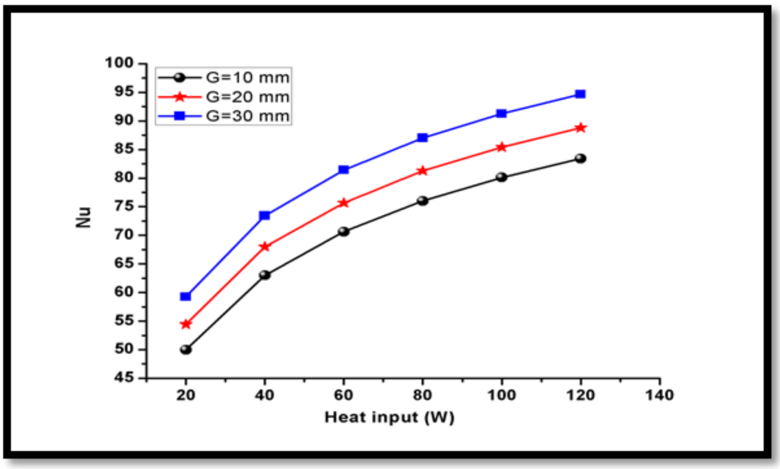

Figure 15: The effect of heat input on the Nusselt number with different G at N = 4.

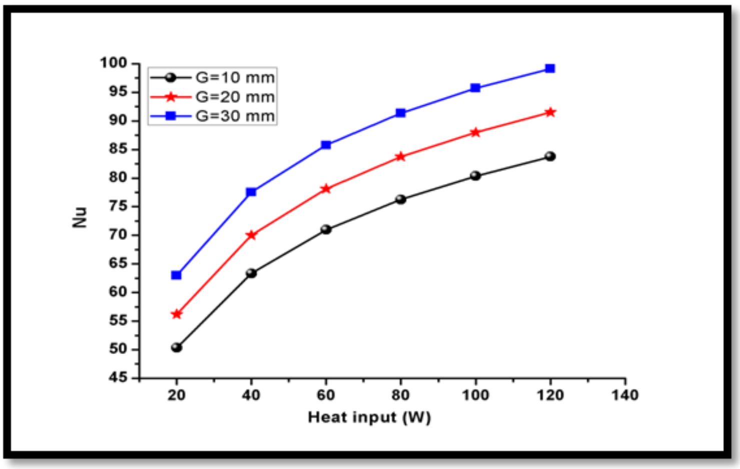

Figure16: The effect of heat input on the Nusselt number with different G at N = 5.

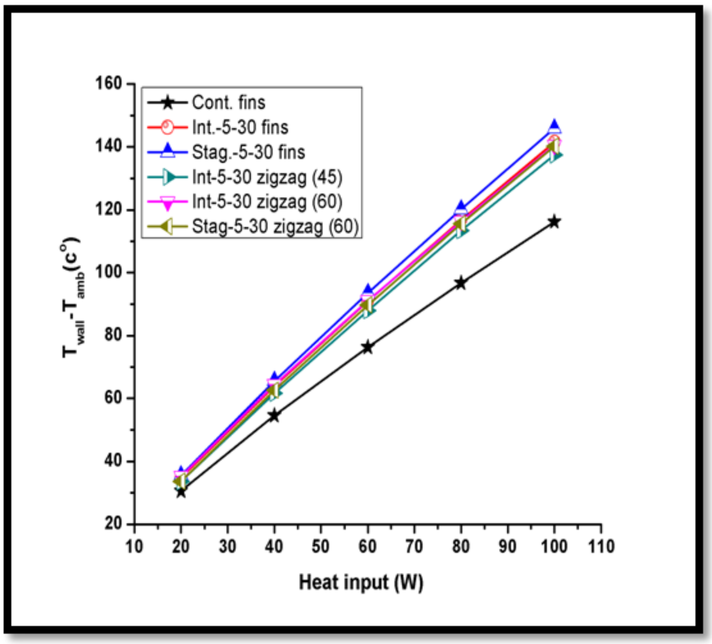

Figure 17: Effect of interruption on temperature difference.

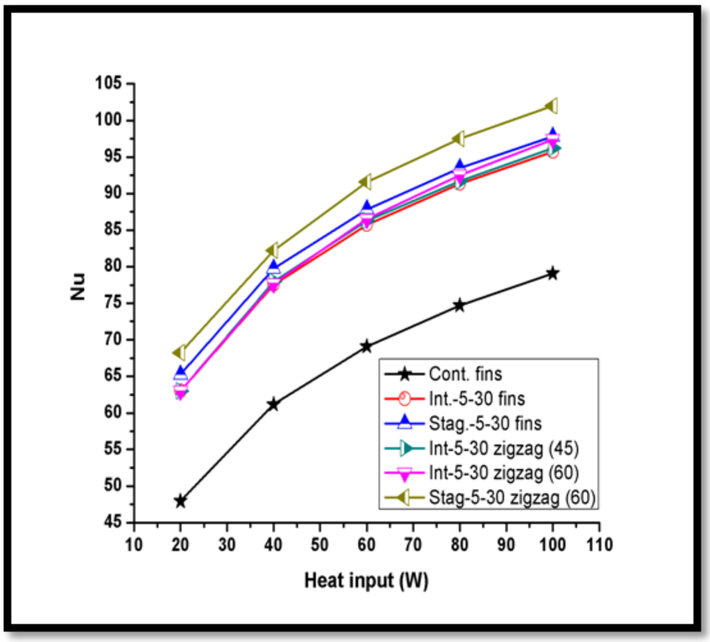

Figure18: Effect of interruption on the Nusselt number.

5.5 The Effect of Fin Arrangement on Temperature and Nusselt number

Through the numerical study and comparison between the fins with different arrangements, Figures 17 and 18 show that staggered interrupted fins, with an angle of 60^o, are better compared to other species, because the surface area subjected to heat transfer is larger, thus giving greater heat exchange as well as an improvement in the Nusselt number.

6. CONCLUSIONS:

Numerical studies were performed in order to establish optimum geometrical heat sink parameters for natural convection heat transfer from interrupted rectangular fin arrays with different arrangements. The most important results of the numerical study can be listed as follows:

- The results are based on the fact that when the fin length is increased, the heat transfer coefficient is low, where the heat transfer coefficient reaches the lowest value at a length of 305 mm (continuous fin) at the same heat flux.
- When the interrupting length is increased, the heat transfer coefficient increases, and increasing the number of interruptions leads to a clear improvement in the heat transfer coefficient. The model in which the number of interruptions is five and the interrupting length is 30 mm is better than the one with continuous fins by 22.3%, in terms of the heat transfer coefficient.
- The optimum interrupted ratio (γ_{opt}), which is a function of surface temperature for different lengths within this range (12 mm < l < 25 mm), has been obtained as an equation:

$$(\gamma_{opt}) = 8112\ (T_{wall} - T_{amb.})^{-2.2}$$

- Staggered interrupted fin arrangement with an angle of 60^o provided a better heat transfer rate in comparison with the continuous fin arrangement. The ratio of improvement is 42.4%.

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NOMENCLATURE

Symbol	Description	Unit
G	Interruption length	mm
G	Gravitational acceleration	m/s^2
H	Heat transfer coefficient	W/m^2k
L	Total length	mm
L	Fin length	mm
Nu	Nusselt number	
P	Pressure	N/m^2
T	Temperature	C°
V	Velocity	m/s
α	Thermal diffusivity	
β	Thermal expansion	K^{-1}
∇	Divergence	
γ	Ratio of the interrupted length, G, to the fin length, *l*	
ϑ	Kinematic viscosity	m^2/s
ρ	Fluid density	kg/m^3