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Development Of Natural Convection Heat Transfer In Heat Sink Using A New Fin Design

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ABSTRACT: Improve of heat sink performance suffers from diverse levels of difficulties due to the complexities of the work environments. For that, researchers try always to develop the performance of heat sink using many ways. Therefore, the enhancement of heat sink performance by using the combination between two types of fin was investigated in this study. All models have constant parameters (size and weight) of heat sink to test the level of enhancing achieved, compared with the original model (longitudinal-fin). Converting part of the original size into pin fins, led to the emergence of a hybrid design (longitudinal – pins fin) that was studied numerically using Finite Element Method and validated by computational simulation (Ansys). The results were conducted for natural convection and Ra=10^7. While, range of surface area was expanded from 1 to 1.8 times. The strong agreement of the validation results (0.31% - 0.52%) showed the reliability of the presented model. Furthermore, the results demonstrated that the hybrid designs have discrimination in several aspects. Consequently, reduced of fin temperatures by (2.7% to 8.8%), lower thermal resistance by (24% to 46%), and augmentation of heat transferred by (31% -80%), compared with longitudinal fin. Meanwhile, entropy generation was minimized with an increase in area ratio for constant heat flux according to the decrease of thermal resistance. Improvement of thermal parameters does not behave in the same approach due to the an overlapping between the effects of miscellaneous parameters, especially at an area ratio greater than 1.5. This led to the approximate steady state of entropy generation starting from the optimum point at area ratio 1.5.

KEYWORDS: Natural Convection, Finite Element Method, Combined Fin-Heat Sink, Entropy Generation Minimization.

INTRODUCTION

Electronic systems become more complex with high power levels to cover the different requirements of modern devices, which led to emerge the group of thermal problems. Therefore, it's necessary to develop the scientific researches that are concerned with the efficient transfer of the heat generation. Available space is limited for electronic units as a result of continuities making the devices in more integrated shape. This challenge should be added to other considerations. Fin- heat sinks are widely used to dissipate the heat using many geometric shapes and varying thermal properties. On the other hand, many electronic devices are adopted on the Heat sink to remove the heat by passive cooling. Mostly, air is used as a working substance for cooling techniques of such devices. The advantages of air cooling and the adoption of innovative designs of heat sinks, led to improve the effectiveness of thermal performance [1,2]. Pin–fin heat sink subject in a magnetic field was investigated experimentally using two arrangements (in-line and staggered) under Forced convection. The result, show a higher value of the heat transfer coefficient associated with in-line pin fins at all operating conditions, compared with the staggered arrangement [3]. While, hexagonal pin fins were carried out at varying heights and different operation conditions. Varying results (Nu-Re and f-Re) proved that the friction factor decreases and Nusselt number enhance with increasing Reynolds number [4]. In other hand, phase change materials integral with square and circular pin-fin heat sinks were studied at two set temperatures, best results of the thermal management in electronic devices can achieve with use the circular pin-fins [5]. Thermal performance of a longitudinal fin heat sink was investigated for different mounting angles. It is found that, remove of the stagnation zone and mounted the heat sink in angle 90^o led to highest improve [6].

Another group of investigators relied on the using the nanofluid in the microchannel heat sink to achieve the desired improvement. Laminar flow at Re (500-2000) and range of volume fraction of nanoparticles (0, 0.04 and 0.08) were used to test the performance of the double layered microchannel. Results of thermal resistance decreased and overall performance improved due to increased of the nanoparticles volume fraction [7]. Thermal behavior of cylindrical micro-fin heat sink was investigated numerically by assuming surface roughness. According to the results, it is established that improved of thermal performance can be occur by use the surface roughness as a result to affected of thermo-geometric parameters [8].

One of the adopted way to improve the performance of the heat sink is merge of two types of fins [7-10]. Plate fins and plate cubic pin-fins heat sinks were combined in horizontal form to find the thermal parameters. Under natural convection, presented heat sinks was enhanced by 10–41.6%, compared with non-combine fins. Also, this design led to lower the thermal resistance [9]. Another work deal with plate-circular pin fin heat sink to test both of the heat transfer characteristics and effects of the configurations. The performance of the combined fins heat sink was preferable than the plate fin heat sink [10]. As the same way, The thermal performance of the a Splitter Plate Pin-Fin Heat Sink was better than Plate Pin Fin Heat Sink according to the results that show the advantages of new distribution on the flow streamlines. Thus, decrement of the thermal resistance and increase heat transfer area [11]. To find the optimum shapes of the fins, comparison between longitudinal and spine fully wet was considered. Besides, calculation process used the efficiency and volume of the fin as parameters in conjugate gradient method (CGM). So that, results show the optimum case has a significant relation to Biot number and relative humidity [12].Has been found an additional improvement about 11% when using a hybrid cross-fin heat sink (series of long fins separated by a series of perpendicularly fins). This enhancement was achieved based on the same weight (size and type of material) of heat sink [13].

The available scientific possibilities of the researcher make the solution procedures are widely miscellaneous [14-17]. Two methods for modeling the internal natural convection heat transfer of a laterally heated square enclosure were studied to show the advantages can be gotten. These methods (finite difference method and finite volume method) used to discretize the equations of Boltzmann method and the results refer to more accurate data and less usage time associated with FVM [14]. The governing equations of compound heat sink (a plate fin heat sink and circular pins) and circular pins were solved using a control-volume-based finite-difference method. Meanwhile, velocity and the pressure terms were coupled in the calculation process. Best performance of compound heat sink led to well reliability, compared with another type of heat sink [15]. Likewise, The governing equations of the elliptical pin fin heat sink and a plate fin heat sink were solved to examine the configuration effects. By using CFD as a solution method, the results illustrated the use of pin fin gives the better performance into the flow and heat transfer [16]. Furthermore, the finite element technique was used to solve the governing equations of a horizontal rectangular fin with perforate technique. The comparison between the results of perforated and solid fin, refers to increases the heat dissipation of the perforated fin and it has high performance in thermal approach [17].

The literature review shows that although many articles presented the different considerations to improve the thermal performance of heat sinks, still the effect of shapes is more parameter has the ability to achieve the desired goal. Consequently, the vertical merge between longitudinal and pin fins is presented in this study. Based on the this new configuration, the same material and size were considered to show the level of improvement can be achieved. The governing equation solves by using finite element techniques based on the operating assumptions to discuss the thermal approach of presenting a model. Then, process of optimization is applied to determine the best model using the method of entropy generation minimization.

METHODOLOGY AND THERMAL ANALYSIS

To achieve the current research goal, should be changed part of the original model to another shape on a requirement of improving the surface area. Accordingly, part of original volume was switching to spin fins with varying length based on the remaining length (L1) of longitudinal part. Meanwhile, a size value of the longitudinal –fin was installed. Figures (1-a) and (1-b) shows the original and hybrid design, respectively.

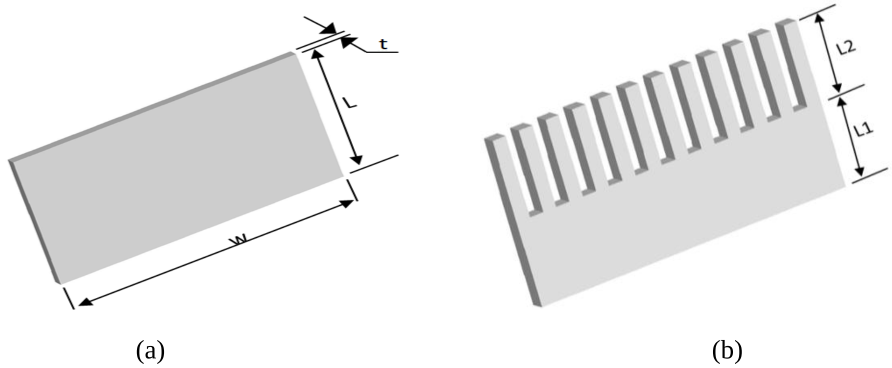


Figure 1. a- original heat sink . b– hybrid heat sink

The number and cross section area of the spin fin can be determined depending on the thickness (t) and width (w) of the original design. According to constant size of heat sink, length of spin fin can be calculated as follows:

$$\mathrm{L2} = \frac{w(\mathrm{L}-\mathrm{L1})}{t * NO.} \tag{1}$$

Where NO.= number of spin fins. In this work, the fins were identified with 12 spins agreeing with the space between the fins.

The thermal analysis of any fin depended on the assumptions that are employed to define and simplify the problem to get the general solution. The heat transfer analysis conducted in this study relied on the following assumptions:

1. Steady heat conduction with no heat generation.

2. 1-D heat transfer analysis of fin height.

3. Constant conductivity (k = 200 W / m.K) and constant base temperature.

5. Insulation of the fin tip at a corrected length, and the radiation effects are neglected.

6. Uniform ambient temperature and uniform convection heat transfer coefficient.

The energy differential equation eq.(2) was formulated by applying the steady-state heat balance over the differential element of high (dx) and parallel to the fin base (w. t) [18,19].

$$\frac{\mathrm{d}}{\mathrm{dx}}\left(\mathrm{A}_{\mathrm{cond.}}\frac{\mathrm{d}\theta}{\mathrm{dx}}\right)\mathrm{dx} = \frac{\mathrm{h.P}}{\mathrm{k}}\theta.\mathrm{dx} \tag{2}$$

To solve the energy differential equation, boundary conditions were considered as follows:

B.C:

$$\left.\begin{aligned} &\theta(\mathrm{x}=0) = \theta_{\mathrm{b}} \\ &\frac{\mathrm{d}\theta(\mathrm{x}=\mathrm{LT_c})}{\mathrm{dx}} = 0 \end{aligned}\right\} \tag{3}$$

$\frac{\mathrm{d}\theta}{\mathrm{dx}}$ and $\theta(\mathrm{x})$ are homogenous at x=L1

Where: $\theta = T(x) - T_{air}; \theta_b = T_b - T_{air}$

In this step, finite element techniques was used to solved the governing equation numerically based on the varitional approach [20]. However, The variational characteristic of the elements adjacent to the longitudinal and pin fins is described in eq.(4).

$$Ih = \frac{1}{2}\iiint_V k(\frac{d\theta}{dx})^2 dV + \frac{1}{2}\iint_{A_{conv.(LT=0\rightarrow L1)}} h\theta^2 dA_{conv.} + \frac{1}{2}\iint_{A_{conv.(LT=L1\rightarrow LT)}} h\theta^2 dA_{conv.} \quad (4)$$

Where: Ih= combined functional of hybrid mode.

The geometric parameters and thermal properties are shown in table 1. These specifications used to solve eq.(4) numerically using solver program (MATLAB) to find the temperature distribution. The mean value of the convection heat transfer coefficient calculated was calculated according to the empirical correlation of the vertical plate [21 ,22] and that related to temperature distribution, eq(5).

$$h_{average} = \frac{Q_{act}}{eff.[[2.(w+t).L1]+NO.[4.t.L2]].\theta_b} \quad (5)$$

$$eff = \frac{T_{average} - T_{air}}{T_b - T_{air}} \quad (6)$$

According to the temperature distribution, heat transferred rate can be calculated as follows:

$$Q_{act} = -k.A(\frac{d\theta}{dx})_{x=0} \quad (7)$$

Area ratio is adopted to indicate the approach of thermal properties when used the hybrid models. It represents the ratio between the surface area of hybrid model to the original surface area as described in eq.(8).

$$A^* = \frac{\text{Surface area of hybrid model}}{\text{Surface area of original model}} \quad (8)$$

Thermal resistance for hybrid geometry was calculated based on the finite element techniques that solved by using the Variational approach [20]. Accordingly, equation (9) used to calculate the thermal resistance [22].

$$R_{th} = \frac{LT}{\sum_{j=1}^{n} \sqrt{h.P.k.A}\,\Delta x \ \tanh(\sum_{j=1}^{n} \sqrt{\frac{h.P}{k.A}}\Delta x)} \quad (9)$$

After that, the stability of the lowest temperature at fin tip adopted as a measure of the successful choices in the numerical iteration process. Then, all necessary parameters to discuss the thermal profile can be found.

Table 1. Specifications of the geometric models

Geometric parameters				
w(mm)	t(mm)	NO.	L(mm)	L1(mm)
69	3	12	30	5-25

Thermal properties			
k	T_P ºC	T_{air} ºC	Ra

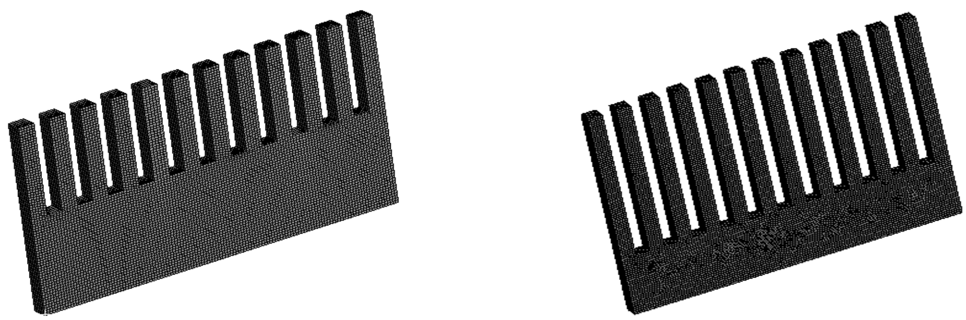
200	70	25	10^7

RESULTS AND DISCUSSION

Validation of the general solution

Fig.(2) show the grid configuration of some presented models using ANSYS-16 steady-state thermal. The advanced size function changed in proximity and curvatures with increasing of the face size towards the successfully obtained of the independent grids due to curvatures and edges.

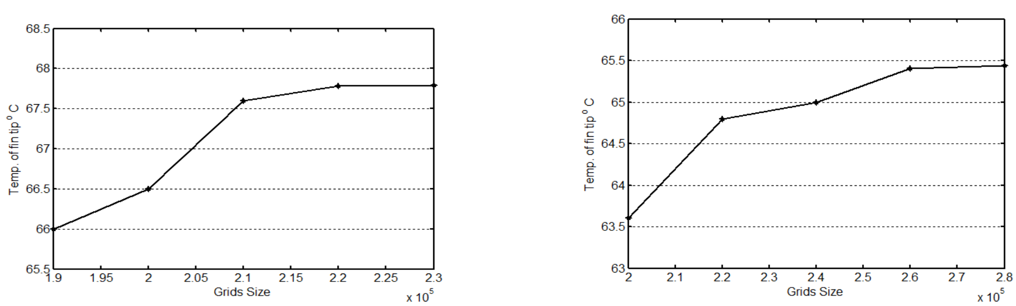
To obtain the appropriate grid, The lowest temperature (at fin tip) adopted as a measure of the successful grid. Fig. (3) Shown the test of several grids to ensure the study is steady-state according to tip fin temperature . Then, the temperature distribution found by adopting the number of grid points in the independent case.



(a) (b)

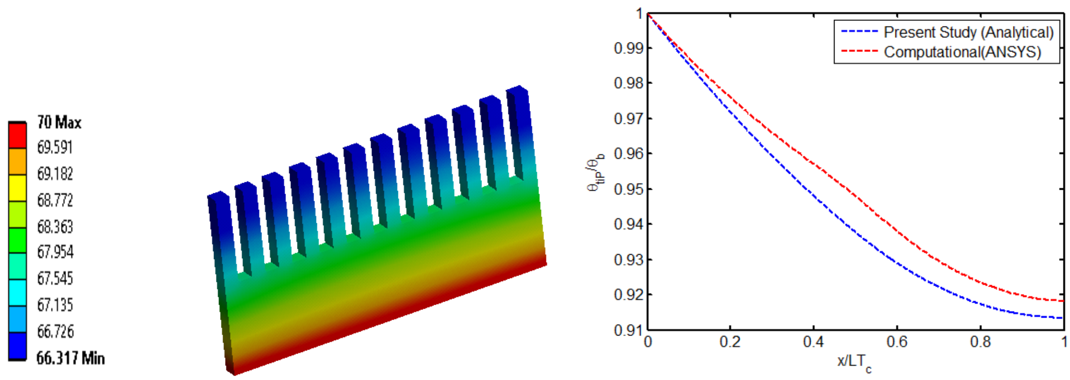
Figure 2. Grids configuration a) L1=0.02m, b) L1=0.01m

Validation procedure deals with the same specifications to show the accuracy between the eq.(4) and Ansys simulation. The temperature distribution of hybrid design (θ/θb) along the fin was adopted to demonstrate the convergence of the results at the optimum grid size as is shown figs. (4 and 5). The agree of results shows in well level between the comparison cases and maximum difference do not exceed (0.52%).



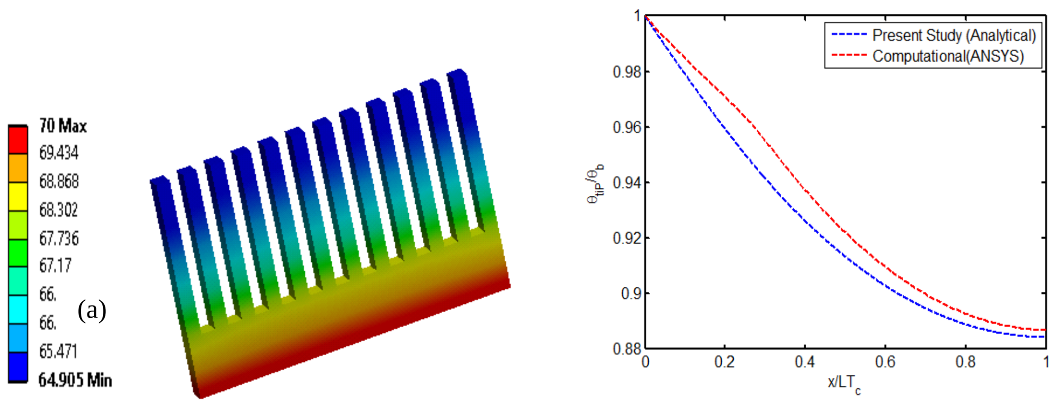
(a) (b)

Figure 3. Grids independent test a)L1=0.02m, b) L1=0.01m.



(a) (b)

Figure 4. (a) Geometric configurations of the computational simulation. (b) Comparison between the present study and ANSYS results for L1=0.02m, A^*=1.3307.



(a) (b)

Figure 5. (a) Geometric configurations of the computational simulation. (b) Comparison between the present study and ANSYS results for L1=0.01m, A^*=1.636.

Behavior of temperature distribution

Figures 4 and 5 show the temperature profile of area ratio 1.3307 and 1.636, respectively. It's clear, that greater temperature (at base) drop towards minimum value (at tip) at varying levels depended on the amount of heat rejected. In fact, the heat transfer relates to geometric parameters and thermal properties of suggested models. When used the combined fins, additional improvement can achieved as shown in Fig.(6). The grade of improvement lies between (2.7% to 8.8%) as a result of distributed the original model (longitudinal fin) to two different shapes (longitudinal-pin fins). This reconfiguration and stabilizing the weight of fin allows to increase the heat transfer area. That explains the reason of down the temperature of the hybrid heat sink.

The temperature drop across the fin (TDA) was used to explain the behavior of dropping temperatures with fin height based on the eq.(10). The results show the dropping of temperature difference was improved with increases of the area ratio as illustrated in fig(7). Meanwhile, the thermal approach suffers from vulnerability points for any fin due to convergence of temperature at near the fin tip. Fig.(7) shows the decrease the length of the fin which has the temperature drop equal or less than 1. This led to increase the amount of heat transfer rate with area ratio.

$$\mathrm{TDA} = \mathrm{T(x)} - \mathrm{T(LT)} \tag{10}$$

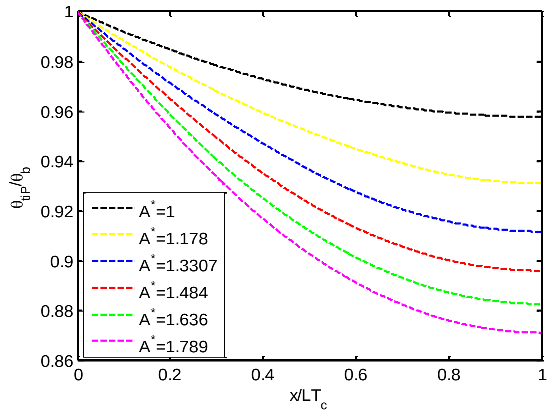


Figure 6. Variation of temperature distribution with the fin length (analytical results).

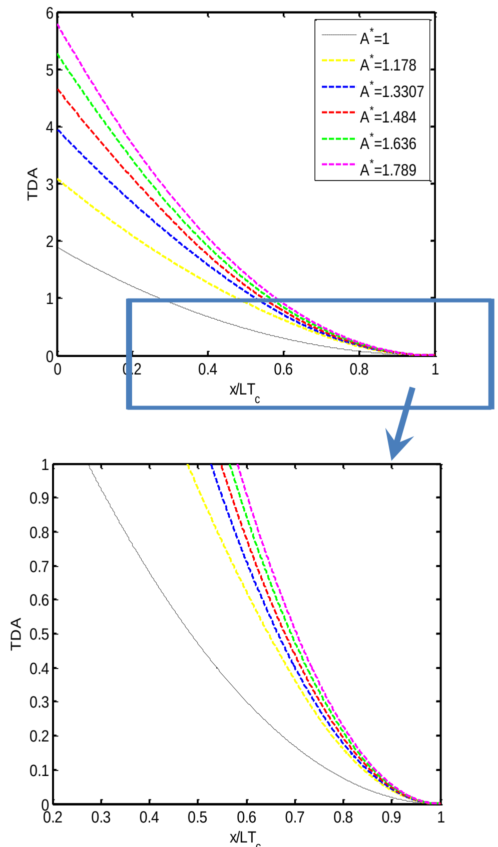


Figure 7. Variation of the temperature drop across the fin (TDA) with area .

Changes in the thermal resistance value

Fig.(8) shows that as area ratio increase, thermal resistance decreases by (24% to 46%). This phenomenon occurs, because of the great modernization in the geometric distribution of the hybrid model which allow for significant contact area between heat sink and cooling fluid. This improvement associated with an increase of fin height, which in turn leads to convergence of values and approximate stability of improvement at A* >1.5.

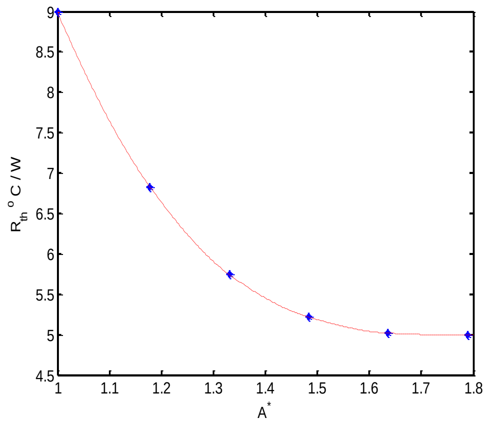


Figure 8. Variation of thermal resistance with area ratio.

Performance of heat transferred

In this study, temperature distribution and thermal resistance were improved. Which leads to increase the amount of heat passing through the combine heat sink as shown in figure(9). The results show, a greater extent of improving in heat transferred followed by little increases started from (A*=1.5) as a result of the overlap between the effects both of convection coefficient and surface area. Indeed, heat transferred is augmented by (31% -80%), compared with original heat sink (A*=1).

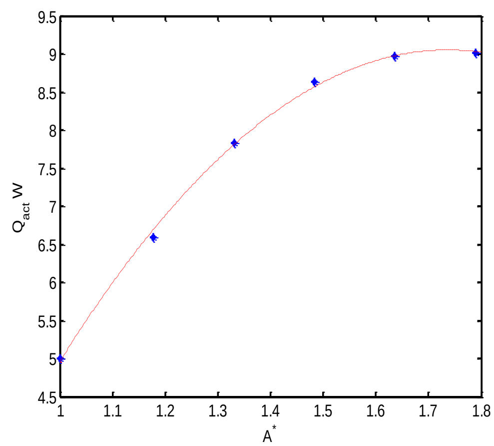


Figure 9. Change of heat transferred behavior with area ratio.

Convection coefficient approach

The value of convection heat transfer coefficient depended on the many parameters which effect on the overall performance. Like, heat (in the numerator) and surface area (in the denominator). The results in figure (10) showed the discrepancy in behavior of convection coefficients towards A* =1.5 due to an overlapping between the effects of miscellaneous parameters . In other words, the improve of heat transferred (at A* < 1.5) overcomes on the growing in surface area. While, this behavior is reflected at A* >1.5.

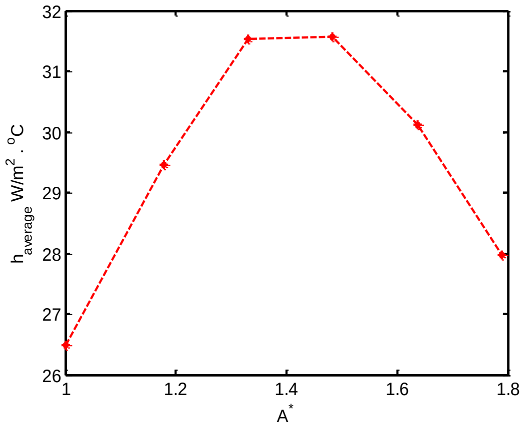


Figure 10. Variation of convection heat transfer coefficient.

Entropy generation minimization ($S_{gen.}$)

An important goal of this study was to improve the performance of heat sink using the combination between two types of fins which led to found many models .Therefore, should be predict the optimal model that would minimize the thermal resistance that necessary to increase heat transfer. Entropy generation minimization used as an optimization way to optimize the overall performance based on the mean value of heat transferred [23].

According to the concept of enthalpy change in the second law of thermodynamics, entropy generation can be calculated by the equation (11). For low velocity conditions, the second term (pressure drop) can be neglected [24].

$$S_{gen} = \iint_A \frac{Q_{act_{mean}}(T(X)-T_{air})dA}{T(X)\,.T_{air}} - \frac{m^{\cdot}.\Delta P}{\rho\,.T_{air}} \tag{11}$$

Finally, Fig. 11 shows the variation of the entropy generation with the area ratio using three values of Ra. It is clear that hybrid models have lowest S_{gen} since the thermal resistance is lower with increasing area ratio. While, S_{gen} increases with Ra as a result to increase the dissipated heat from the heat sink . However, the optimum point can be achieved at an area ratio equal or greater than 1.5 according to the stability of entropy generation values.

CONCLUSION

The main objective of this study is improving the thermal performance of heat sink according to the shape changing. This change consists the vertical combination between longitudinal and pin fins. Relying on a hypothesis of constant both of material and size, many models were obtained with various fin heights. The finite element technique was used to solve the mathematical modeling. While, Ansys simulation was applied to validate the analytical model and the results show the excellent reliability based on a high level of agreement by maximum difference does not exceed (0.52%).

Generally, thermal parameters have substantial enhancement with change of area ratio towards 1.5 due to the increase of heat transfer area and convection coefficient value. Then, the discrepancy of behavior happened as a result of overlapping between the effects of miscellaneous parameters. In other words, increase in area ratio (at A* > 1.5) become inconsistent with the improve of heat transfer which leads to decline of convection coefficient. Moreover, many advantages can achieve with hybrid models; a significant drop in temperature profile about (2.7% to 8.8%), decrease in thermal resistance by 46% and increases in heat transferred by 80%.

Entropy generation was minimized with an increase in area ratio for constant heat flux as a result of the decrease in thermal resistance. Further improvements in entropy generation can occur if the Rayleigh number was increased. But for the area ratio of greater than 1.5, approximate steady-state can happen. For that, the optimum point may be selected at the 1.5 area ratio.

Finally, Combed fins- heat sink may be possible to use as a new term to rename the hybrid model based on the shape of the presented design.

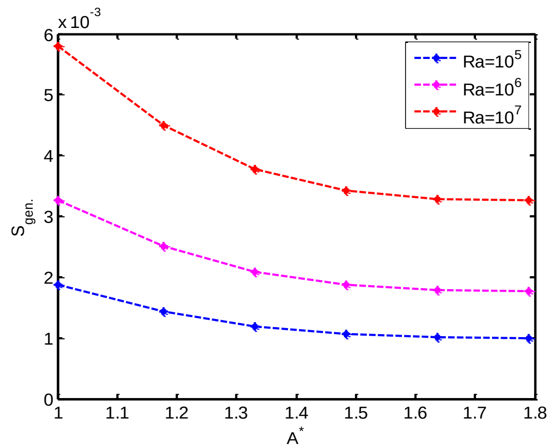


Figure 11. Variation of S_{gen} with area ratio.

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NOMENCLATURE

A	Cross section area [m^2].
A^*	Area ratio.
d	Differential dimension.
eff	Efficiency.
h	Convection coefficient [$W/m^2.\ ^oC$].
k	Thermal Conductivity [$W/m.\ ^oC$].
L	Length of original heat sink [m].
L1	Length of longitudinal fin in hybrid model [m].
L2	Length of spin fin in hybrid model [m].
LT	Total length [m].
m	Fin performance factor [1/m].
$\dot{m}$	Mass flow rate.
Nu	Nusselt number.
P	Perimeter [m].
Pr	Prandtl number.
Q	Heat transfer [W].
Q^*	Heat transfer ratio.
R_{th}	Thermal resistance [$^oC/W$].
Ra	Rayleigh number
S_{gen}	Entropy Generation (W/ K).
t	Thickness [m]
T	Temperature [^oC]
W	Width of Fin [m]
X	Length axis

Subscript

air	Cooling fluid
act	Actual
average	Average
b	Base
c	Correct

tip	Tip of fin
sur	Surface

Greek Letters

Θ	Temperature difference
Δ	Step increment
Δp	Pressure drop
ρ	density