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Effect of Notched Pin Fin Heat Sink on the Heat Transfer Performance: Numerical Study

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ABSTRACT

The increasing complexity and miniaturization of electronic applications necessitate the development of efficient and compact heat sink designs for effective heat dissipation. This study presents a numerical investigation of heat transfer performance in circular pin-fin heat sinks featuring notches of varying sizes. Five distinct heat sink models are analyzed, with the first model comprising a solid fin, while the remaining four incorporate notched fins of different dimensions. Results indicate a significant influence of notch size on heat transfer performance, particularly for larger notches. The Nusselt number for a heat sink with a 4 mm notch size exhibits an increase of approximately 9% compared to that of a solid-fin heat sink. Additionally, the average temperature of heat sinks decreases with the introduction of notched fins, resulting in a temperature difference of 2.15℃ between solid-fin and 4 mm-sized notch heat sinks. An assessment of overall efficiency and overall effectiveness reveals that all notched heat sinks are viable options, with the 4 mm-sized notch heat sink demonstrating optimal performance in this study. This investigation provides valuable insights for the design of high-performance heat sinks in compact electronic applications.

1. INTRODUCTION

Air cooling strategies have gained significant popularity due to their extensive applications across various fields. As electronic devices are increasingly employed in complex processes, a higher amount of energy must be dissipated to prevent excessive temperature increases [1, 2]. The miniaturization of these devices has driven designers to explore more efficient cooling techniques, with notched pin-fin heat sinks presenting a potential solution for numerous electronics, such as computers. Prior research has primarily focused on alternative heat sink designs, including longitudinal fins and perforated fins [3, 4].

Various studies have been conducted to investigate different heat sink designs. For instance, Brignoni and Garimella [5] examined a pin-finned heat sink equipped with four single nozzles of varying diameters. Utilizing a Reynolds number range of 5,000–20,000, their findings revealed that heat source output power had minimal impact on heat transfer. El-Sheikh and Gurimella [6] performed an experimental analysis of the enhanced heat transfer in pin-fin heat sinks with a single air jet, within a typical range of Reynolds numbers (8,000 - 45,000). Relative to a heat sink without fins, the total fin effectiveness for pin-fin heat sinks was determined to be between 2.4 and 9.2. Experimental results from Thrasher et al. [7] demonstrated an increase in free convection using a pin-fin heat sink with a chimney. According to their study, an ideal porosity of 0.91 yielded the highest performance. Computational models and experimental setups were employed to showcase improved heat transfer in a pin-fin heat sink enclosed within a casing.

Yu and Joshi [8] investigated free convection, conduction, and radiation from various heat sink sources. The dimensions of the enclosure were 127×127 mm, with a height of 41.3 mm, while the heat source measured 25.4×25.4 mm. The results indicated that the orientation of the heat sink significantly influenced the rate of heat transfer. In an experimental study, Kim and Kim [9] evaluated the thermal performance of cross-cut heat sinks, examining the effects of cross-cut positions, lengths, and numbers. Their findings demonstrated that cross-cut length had a substantially greater impact compared to other parameters.

Jang et al. [10] presented a numerical optimization study on LED lighting application heat sinks to identify a lighter-weight design with comparable cooling performance. The optimal design exhibited a 30% weight reduction relative to the plate type, while maintaining a similar cooling performance. Yeom et al. [11] conducted a numerical investigation of the ideal perforation diameter to determine the most effective heat sink design for enhanced temperature dissipation. According to their results, the base plate temperature could be reduced by 1.5% to 2.1% compared to a solid case. Cai et al. [12] reported their findings from a numerical comparison of notched, slotted, and circular perforations. Data suggested that notched and slotted perforations reduced the heat sink base temperature more effectively than circular perforations.

Maji et al. [13] performed a computational evaluation of various perforation geometries in in-line and staggered pin-fin heat sinks to demonstrate enhanced heat transfer performance. The results indicated a notable improvement in heat transfer compared to solid-fin heat sinks with different hole geometries. Yasin and Oudah [14] presented an experimental and numerical analysis to investigate the effect of perforated fins on finned tube heat exchangers. Heat transfer and pressure drop calculations were made and compared with solid finned tubes, resulting in a 26% increase in the Nusselt number relative to a solid finned tube. Dhumne and Farkade [15] reported a numerical analysis of the importance of perforation with circular fins in a rectangular channel. They observed a significant increase in heat transfer rate compared to solid fins.

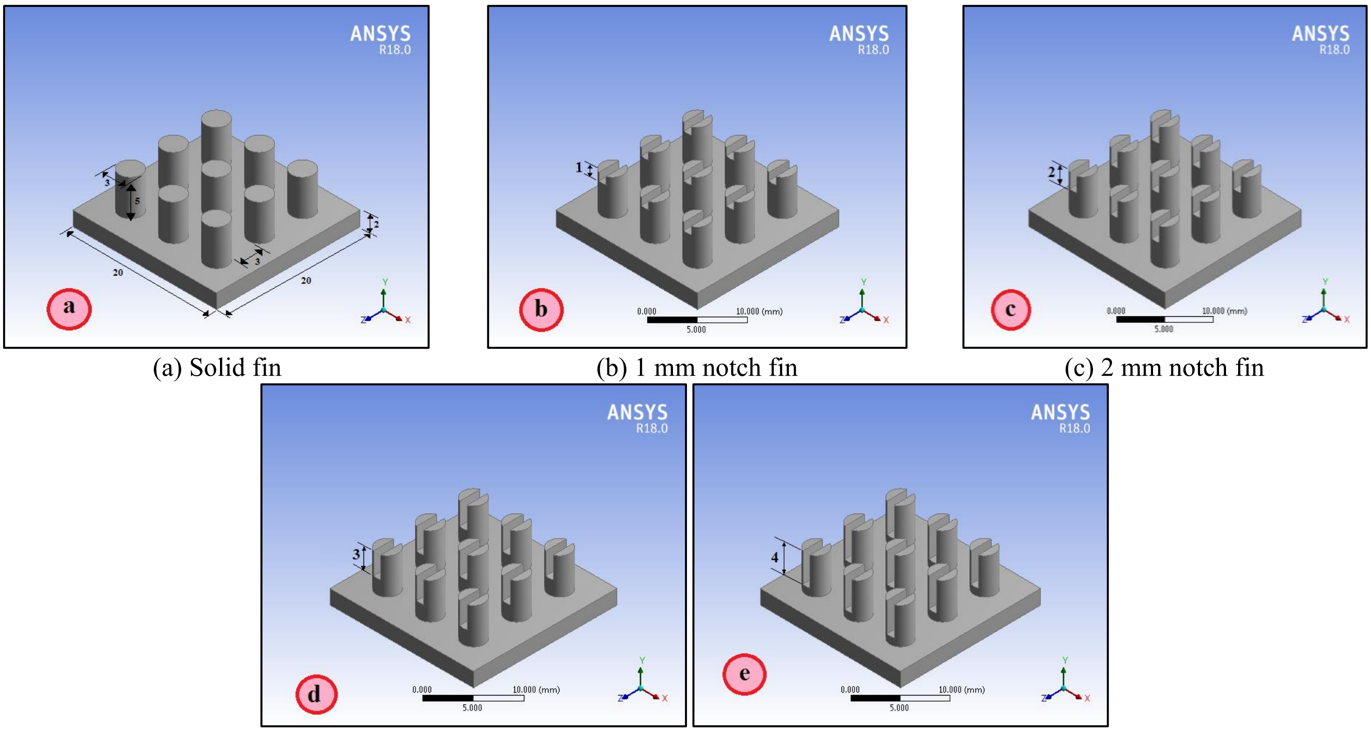
Venkatesham et al. [16] examined 3D forced convection heat transfer from pin-fin micro heat sinks with solid and perforated fins in electronic cooling applications, utilizing a Reynolds number range of 20,000-39,000. Results indicated that fins with circular perforations significantly improved heat transfer and reduced pressure loss. Shaeri and Yaghoubi [17] conducted a numerical investigation of laminar forced convection heat transfer between rectangular solid and perforated fins, with a Reynolds number range of 100–350. Thermal performance and effectiveness of perforated fins were found to be higher than those of solid fins, and perforation significantly reduced fin weight. Shaeri and Jen [18] explored numerically optimal natural and radiation heat transfer in a radial pin-fin heat sink to achieve a lighter heat sink with comparable performance. According to their findings, the optimal design for a radial heat sink weighed 30% less than a plate-finned heat sink. Hobbi and Siddiqui [19] carried out an experimental and numerical study to demonstrate the effect of perforation shape on a heat sink. Their results showed that finer perforations had a significant impact on convective heat transfer. Additionally, several studies focused on lateral perforations of varying geometries and dimensions on rectangular fins, revealing a substantial effect on heat dissipation compared to solid fins [20, 21].

The primary focus of this paper is to examine the influence of notch depth on pin fins and predict the optimal depth for maximum heat dissipation. While previous studies have concentrated on perforated fins and fin arrangements to enhance heat dissipation, this work aims to investigate the effect of heat sink notches on heat transfer enhancement and explore the feasible range of notch depths for pin-fin heat sink designs.

2. NUMERICAL APPROACH

2.1 Problem description

The pin-fin heat sink designs considered are shown in Figure 1 and include five heat sink designs, both solid and with variable-depth notched pin fins. The heat sinks in this paper are subjected to forced convection with 5 different inlet velocities of 4, 5, 6, 7, and 8 m/s. The dimensions of the heat sink are 3 mm fin diameter, 5 mm fin height, and 3 mm spacing between fins. The notch dimensions are 1 mm in width and various notch depths of 1, 2, 3, and 4 mm [22]. The models and simulations were created using ANSYS Fluent 18.0. The numerical approach used to solve this case is the SST method. The study focused on the effect of notch depth on the heat transfer performance of a pin-finned heat sink to predict the optimum notch depth. Each model consists of nine fins in an in-line arrangement as shown in Figure 1.



(a) Solid fin (b) 1 mm notch fin (c) 2 mm notch fin

(d) 3 mm notch fin (e) 4 mm notch fin

Figure 1. Schematic diagram of models pin fin heat sink under consideration

2.2 Governing equations and assumptions

The governing equations used to solve this problem are the continuity equation, Navier-Stokes equations, and energy equation. A steady-state, three-dimensional flow is assumed for this case. There are some assumptions that must be made to make the case solvable under numerical solutions. The assumptions of fluid flow are as follows:

1. Three-dimensional heat transfer.
2. Incompressible flow (Mach number <0.3).
3. A steady-state and steady flow (no change of fluid properties with time).

4. Turbulent flow (Reynolds number between 33000-51000).

5. Constant air properties over the temperature range under study.

6. Negligible heat transfer by radiation.

According to the assumptions above the basic equations are reduced to the following equations with the supplementary equations [23]:

Continuity equation:

$$\frac{\partial}{\partial x}(\rho u)+\frac{\partial}{\partial y}(\rho v)+\frac{\partial}{\partial z}(\rho \omega)=0 \tag{1}$$

Navier-Stokes equations:

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+\omega\frac{\partial u}{\partial z}\right) = -\frac{\partial P}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \tag{2}$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+\omega\frac{\partial v}{\partial z}\right) = -\frac{\partial P}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \tag{3}$$

$$\rho\left(u\frac{\partial \omega}{\partial x}+v\frac{\partial \omega}{\partial y}+\omega\frac{\partial \omega}{\partial z}\right) = -\frac{\partial P}{\partial z}+\mu\left(\frac{\partial^2 \omega}{\partial x^2}+\frac{\partial^2 \omega}{\partial y^2}+\frac{\partial^2 \omega}{\partial z^2}\right) \tag{4}$$

Energy equation:

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+\omega\frac{\partial T}{\partial z}\right) = k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \tag{5}$$

The governing equation of the solid domain (aluminum) is an only conduction heat transfer of the heat sink [24]:

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}=0 \tag{6}$$

The Reynolds number of the air could be calculated from:

$$Re=\frac{\rho_a.u_a.D_h}{\mu_a} \tag{7}$$

where, D_h is the hydraulic diameter of the heat sink [25]:

$$D_h=\frac{4V_o}{A} \tag{8}$$

Then, Nusselt number is calculated:

$$Nu=\frac{h_a.D_h}{k_a} \tag{9}$$

The heat transfer to cold air can be calculated by:

$$Q=\dot{m}C_p(T_e-T_i) \tag{10}$$

The rate of heat transfer to or from heat sink can be determined from:

$$Q=h_a.A.(T_w-T_b) \tag{11}$$

Then by equaling Eqns. (10) and (11), h_a can be calculated as flow:

$$h_a=\frac{\dot{m}.c_p(T_e-T_i)}{A.(T_w-T_b)} \tag{12}$$

The overall efficiency of heat sink could be found as following:

$$Q_{no\,fin}=h_a.A_{no\,fin}.(T_s-T_\infty) \tag{13}$$

$$Q_T=h_a.A.(T_s-T_\infty) \tag{14}$$

$$\eta=\frac{Nu/Nu_s}{\Delta P/\Delta P_s} \tag{15}$$

The overall effectiveness of heat sink could be found as following:

$$\varepsilon=\frac{Q_T}{Q_{no\,fin}} \tag{16}$$

The total thermal resistance $R_{th.}$ is calculated as follows:

$$R_{th.}=\frac{T_{base}-T_\infty}{\dot{Q}_T} \tag{17}$$

2.3 Boundary conditions

The mesh creation procedure of the ANSYS-FLUENT program specifies the boundary conditions, as indicated in Table 1 and Figure 2.

The temperature and velocity are set for the inlet. The inlet temperature was 27°C, while there are five values of velocity, which are 4, 5, 6, 7, and 8 m/s. The boundary conditions are gauge pressure, adiabatic, and 75°C for the outlet, outlet walls, and surface temperature, respectively.

Table 1. Boundary conditions of the problem under study

Boundary - inlet	**Type**	**Velocity inlet**
	Static temperature	27°C
	Inlet velocities	4,5,6,7 and 8 m/s
Boundary - outlet	**Type**	**Pressure outlet**
	Pressure outlet	0 Pa gauge pressure
Boundary – outer wall	Heat transfer	Adiabatic
	Mass and momentum	No slip wall
Boundary - constant temperature	**Type**	**Wall**
	Location	Bottom surface of the heat sink
	Heat transfer	Constant surface temperature
	Surface temperature	75°C

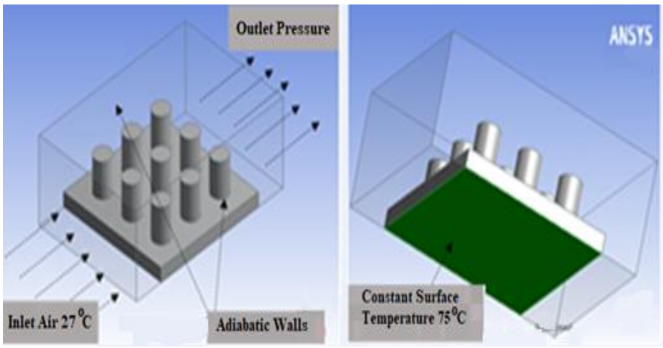


Figure 2. Boundary conditions of the current study

2.4 Mesh independence

The mesh resolution greatly affects the results' accuracy, not only for boundary conditions. To ensure that the results are due to the boundary conditions used, mesh independence must be calculated. Figures 3 and 4 show the change in temperature and pressure, respectively, due to element number. When there are fewer elements, the results are very good, but this effect gets less strong as the number of elements goes up.

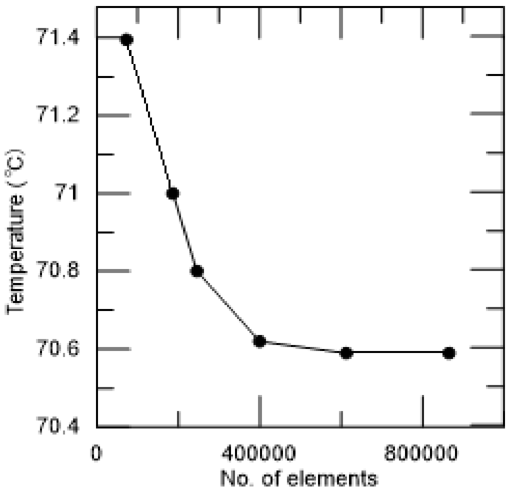


Figure 3. Variation of fin tip temperature with a total element number

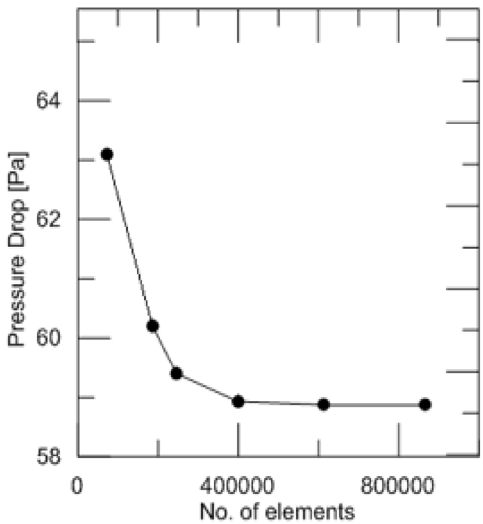


Figure 4. Variation of pressure drop with a total element number

3. RESULTS AND DISCUSSION

With prior efforts, the numerical technique in this case is validated. The contrasts between the current work and earlier studies in terms of Nusselt number (Nu) are shown in Figure 5. A rectangular fin heat sink's performance in turbulent heat transfer was presented in an experimental study by Ehteshum et al. [26]. Good agreement was found during validation, with an average Nu deviation of only 6%. Additionally, Ismail [27] published a numerical analysis of a heat sink with longitudinal fins. Again, there was good agreement when compared to this study, with an average Nu divergence of only 7%.

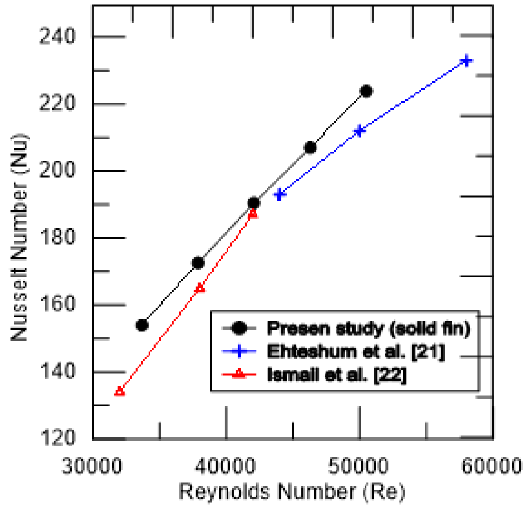


Figure 5. Comparison between present study and previous works

In this section, the effect of notch size on different heat sinks is discussed. As mentioned earlier, five heat sinks are used in this study. The first case involves a solid-fin heat sink, while the others involve notched-fin heat sinks. As shown in Figure 6, the Nusselt number of different pin-fin heat sink notch sizes show the enhancement in heat transfer. The smaller notch size of 1 and 2 mm gives a little enhancement, which reaches a 4% increase in Nusselt number, but the enhancement becomes significant with notch sizes 3 and 4 mm, with a maximum increase in Nusselt number of 9% as compared with a solid fin. That means a heat sink with a deep notch will dissipate more heat.

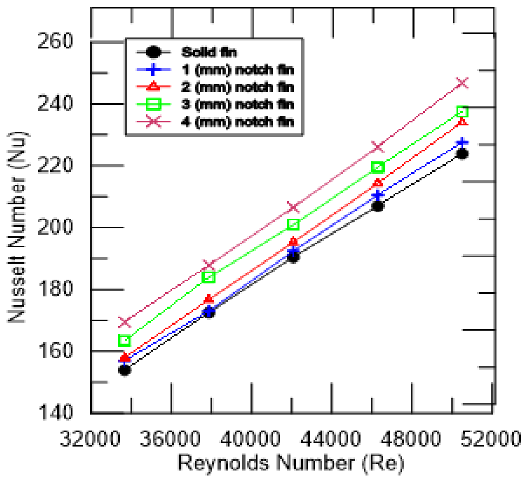


Figure 6. The effect of notch size on Nusselt number

The pressure drop gives an indication of the pumping power required for heat sink under work. As shown in Figure 7, it is clearly evident that pressure drop changed with different notch sizes, and its decrease with a 4 mm notch reached 10% less than a solid fin heat sink. That is, the model with a 4 mm notch fin required less pumping power.

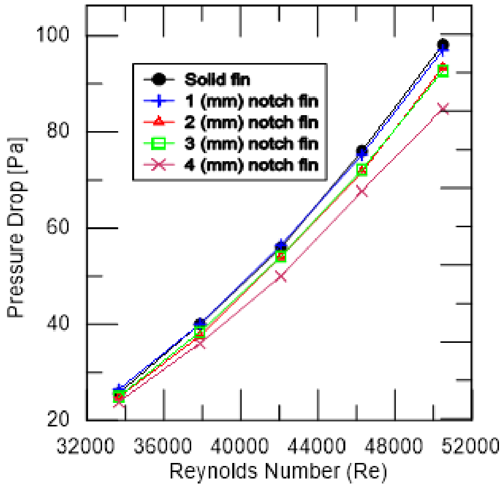


Figure 7. The effect of pressure drop due to notch size

From Figures 7 and 8, the formula fitting for case 4 mm notch size can be expressed linearly as following equations:

$$Nu = 0.0046Re + 14.668 \tag{18}$$

$$\Delta P = 0.0036Re - 101.06 \tag{19}$$

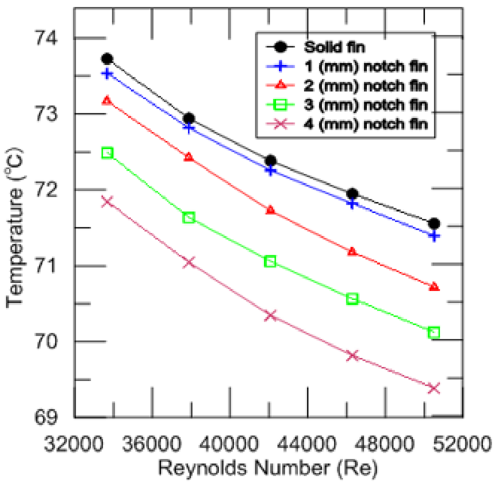


Figure 8. The effect of notch size on heat sink average temperature

The effect of notch size on the average heat sink temperature is shown in Figure 8. The notch size gives a good decrease in heat sink temperature with the same boundary conditions due to an increase in heat transfer area. The temperature decrease due to the 4 mm notch fin heat sink reaches 2.15℃ as compared with a solid fin heat sink. This decrease gives the heat sink the ability to be used in more applications without increasing the risk of temperature increases.

The overall efficiency of different notched pin-fin heat sinks gives a good indication of which is the optimum design. Eq. (15) shows the relation between Nusselt number and pressure drop to predict overall efficiency. Figure 9 shows that all notch type heat sinks in this study are acceptable designs, but a 1 mm notch size gives a very small enhancement. As shown in Figure 9, the 4 mm notch-size heat sink is the best design among the other heat sinks in this study, with an increase in overall efficiency of about 17% when compared to the 1 mm notch-size heat sink. Figure 9 also shows fluctuating especially with 1 mm, 2 mm and 3 mm notch size. This fluctuating occurs because the vortexes on the other hand the different between two points is very little which can be neglected.

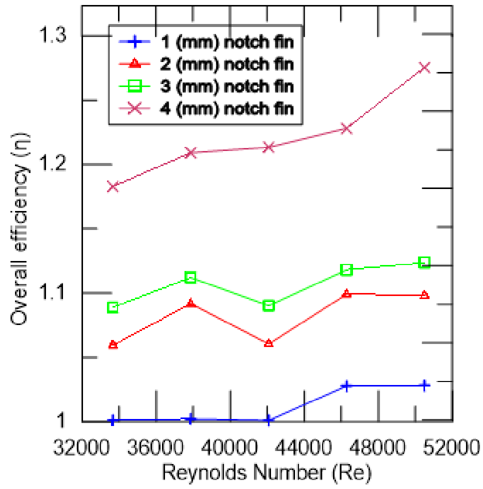


Figure 9. The overall efficiency of different notch heat sink

The notch depth effect on the overall efficiency is shown in Figure 10. The notch size 1 mm shows less overall efficiency, but when the velocity increase from 4 to 8 m/s the overall efficiency increases by about 4%. The overall efficiency gives good enhancement with higher notch size. The effect of air velocity on overall efficiency is less than notch size effect.

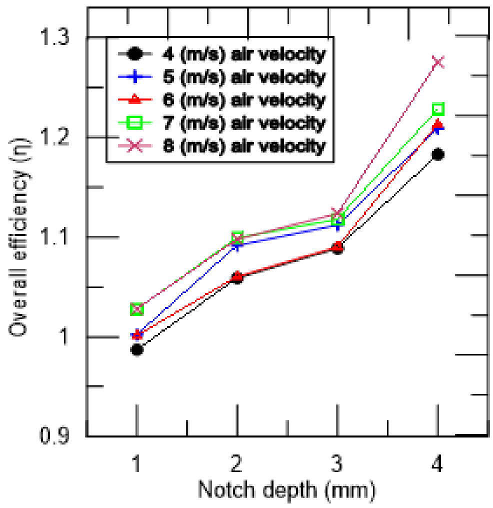


Figure 10. The effect of notch depth on overall efficiency of different air velocities

The ratio of the total heat transfer rate of a heat sink to the total heat transfer rate of a base without fins, known as the overall effectiveness, provides a good measure of the heat sink's thermal performance. The effectiveness of the heat sink is clearly influenced by the size of the notch, as shown in Figure 11, where a maximum improvement of 4 mm in notch size results in a 2% improvement over a solid fin heat sink.

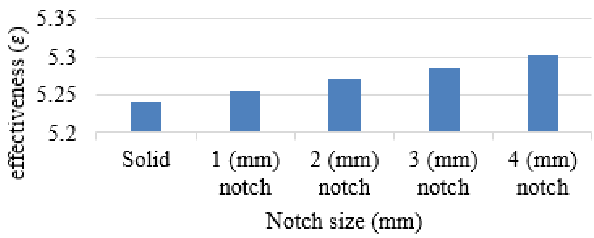


Figure 11. The effect of notch size on the overall effectiveness

Thermal resistance, defined as the ratio of the total heat transfer to the difference between the ambient temperature and the fin base temperature (Eq. (17)), the heat resistance reduced as the Reynolds number increased, as illustrated in Figure 12. Due to decreased heat transfer, the solid-fin heat sink has a stronger thermal resistance. The thermal resistance of the 4 mm heat sink is about 5% less than that of the solid fin heat sink.

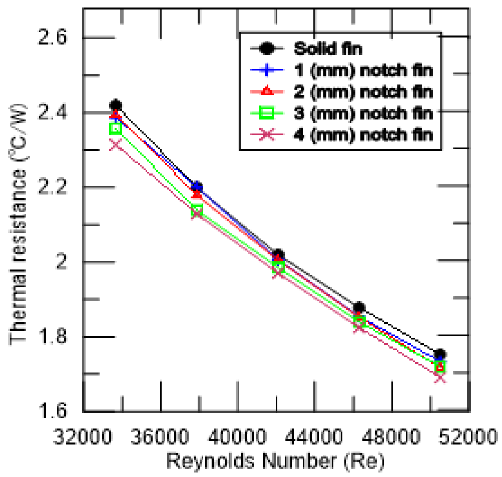


Figure 12. The effect of notch fin on thermal resistance

Figures 13 and 14 display the temperature, the shape of a solid fin, and a heat sink with a 4 mm notch. The improvement in heat transfer rate is highlighted by the temperature distribution in Figures 13 and 14. When using a 4 mm notch heat sink Figure 14 as opposed to a solid fin heat sink Figure 13, the temperature contour clearly demonstrates the increase in heat transfer rate.

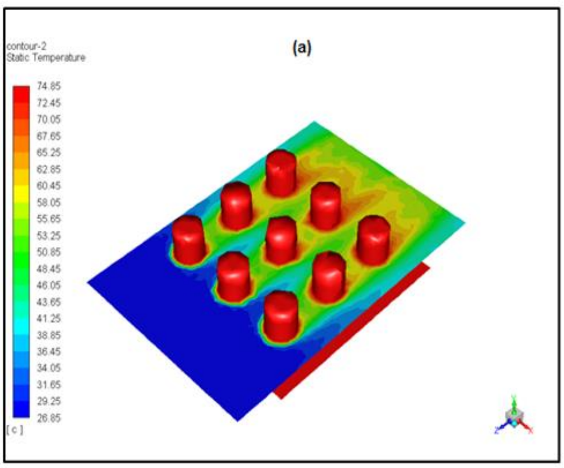


Figure 13. Temperature contour of solid fin heat sink

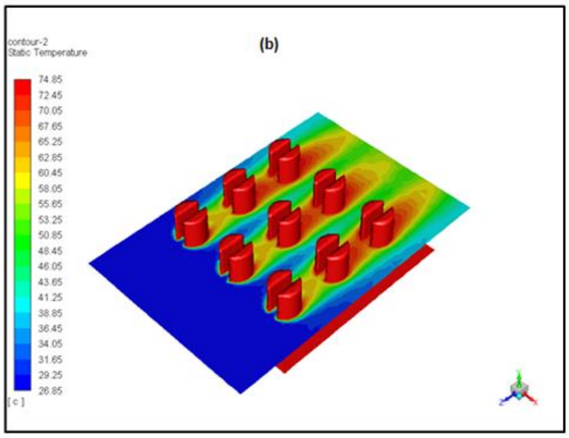


Figure 14. Temperature contour of 4 (mm) notch fin heat sink

4. CONCLUSIONS

This work mainly focuses on the enhancement of a circular pin-finned heat sink by making a notch at the fin tip. A solid fin and four different-sized notch heat sinks are used to show the effect of a notch and the optimum notch size. From all the above results, it can conclude the following:

•The notch effect on heat transfer performance becomes significantly stronger with higher notch sizes, where the Nusselt number increases by about 9%, accompanied by a 10% decrease in pressure as compared with a solid-fin heat sink.

•The average heat sink temperature will decrease when using a 4 mm notch fin heat sink by about 2.15°C as compared with a solid fin heat sink with the same boundary conditions.

•The best notch size noted in this study is 4 mm.

•The smaller notch size of 1 mm and 2 mm is given a little enhancement, so it is useless.

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NOMENCLATURE

A	Surface area, m^2
cp	Specific heat at constant pressure, $kJ.kg^{-1}.K^{-1}$
D_h	Hydraulic diameter, m
h	Heat transfer coefficient, $W.m^{-2}.K^{-1}$
k	Thermal conductivity, $W.m^{-1}.K^{-1}$
$\dot{m}$	Mass flow rate, $kg.s^{-1}$
Nu	Nusselt number
Q	Heat transfer, W
Re	Reynolds number
t	Time, s
T	Temperature, K
V	Volume, m^3
ΔP	Pressure drop, Pa

Greek symbols

ε	Overall effectiveness
η	Overall efficiency
μ	Viscosity, $N.m^{-2}.s$
ρ	Density, $kg.m^{-3}$

Subscripts

∞	Free stream

a	Air	s	Solid fin
b	Bulk	T	Total heat transfer
e	Exit	w	Wall
i	Inlet		