Numerical study of heat transfer of embossed plate fined heat sink

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Abstract. In this article, a study of the thermal performance of embossed fined mini-channel heat sink numerically has been carried out using CFD. Commercial software STAR-CCM+ was used for the investigation. Embossed impressions were vertical to the heat sink base. The numerical results were presented in terms of flow visualization,a temperature distribution on the heat sink, velocity contour, and streamlines, Nusselt number for heat supply ranging from 18 to 70.4 W under natural convection. The existence of artificial geometry on the fin surface leads to greater Nusselt number due to the high energy fluid flow mixing near the heat sink wall by the separation and reattachment of the boundary layer. In addition, an embossed heat sink has shown better temperature distribution. The result showed a good agreement with available experimental data.

1. Introduction

The electronics industry is gradually developing. Light, small, and high-power electronic components are becoming more ubiquitous. Without efficient heat transfer, excessive temperatures may make the working performance of these components unstable, reducing their lifespan or even causing damage to them. Therefore, a major challenge in the design of electronic components is to enhance the efficiency of their heat transfer (Hung-Yi Li, 2013). Effective method to remove high rate heat is essential to control substrate temperatures under critical temperatures which causes devices to fail. Even though liquid is used widely in cooling system as the most important applicants for high flux applications, air refrigerating is still by far the most used cooling method to cool electronics devices for its reliability, simplicity, and low cost. Air has low heat transfer rate, due to its low thermal conductivity. As a result, the general method for heat transfer enhancement is through manipulation of smaller fin space to accommodate extra fin surface. The cannelure fin with the hollow/cavity configuration of heat sink could attain 25% increase of heat transfer with increasing in friction decrease of around 20% (Chi-Chuan Wang, 2011). Furthermore, opposite to interrupted fin or triangular vortex generator, it is approved that their suggested fin configuration leads to important increase in fully developed flow. Vortex generators shape, location, dimension have major contribution on the thermal and hydraulic ability of heat sink (Timothy Dake, 19-23 July 2009). They have shown that the angle of attack of generators play significant part in the heat transfer rise of heat sinks. Moreover, when the vortex generator height, the heat transfer improves and the greater improvement was registered when the generator and channel heights are equal. Finally, they denoted that the vortex generators could enhance the thermal ability of heat sink. High frontal velocity and large fin pitch lead to more useful vortex generators (K.S. Yang, 2010).

Starner at el (Starner & McManus, 1963) investigated four different fin groups and three different base plates to measure natural convective heat transfer rate considering fin pitch and the fin height. Welling and Wooldridge (Welling, 1965)) approved that the vertically fin array is the best among all

kinds of fin arrays comparing fin heights. The change of temperature with the variation in the fin height to fin pitch ratio was argued and an equation for best magnitude of the fin height to fin pitch ratio was suggested. Chaddock they changed the fin spacing and the fin height in the maximum vertically fin groups and explained that the radiation mode shares around 20% of the total heat transfer (Chaddock, 1970). Alhara studied the influence of fin geometry and temperature on average heat transfer rate obtained empirical correlation (Alhara, 1970). Singh and Patil studied experimentally the effect of semi-circle embossed impressions (Singh & Patil, 2015).

2. Geometry

A three-dimensional geometry has been drawn using SolidWorks and the simulation has been performed using starccm 8.1. The heat supply is assumed at the bottom of the mini-channel heat sink. The test heat sinks of 150 mm 150 mm 50 mm dimensions having aluminum fins of 150 mm length, 50 mm height and 1 mm thickness were used in the experiments. Three power supply t values, 18, 39.6, 70.4 W were used with each heat sink configuration. Impressions have dimensions of 40 mm length and 2 mm height were placed in the fins at different angles (45°, 90°) and at pitches 12 mm.
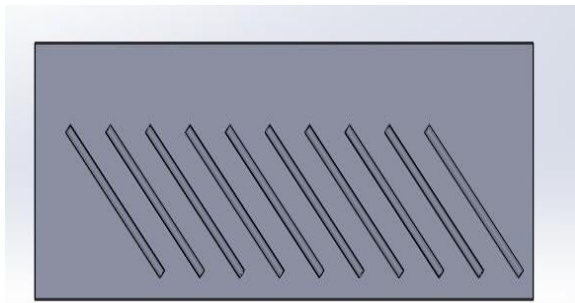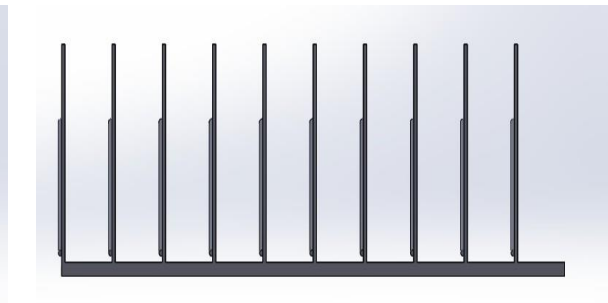

Figure 1. front view of heat sink geometry

Figure 2. side view

3. Meshing Independence Study

To understand the sensitivity of the heat sink grid, four different grids, 34347 coarse grids cells, 399351 fine grids cells and 498976 and 589388 very fine grids cells were used to check the stability of the grids.

Best grids have been used within the heat sink in order to measure the fluid flow and heat transfer characteristics. Difference between surface temperature and ambient temperature has been calculated.
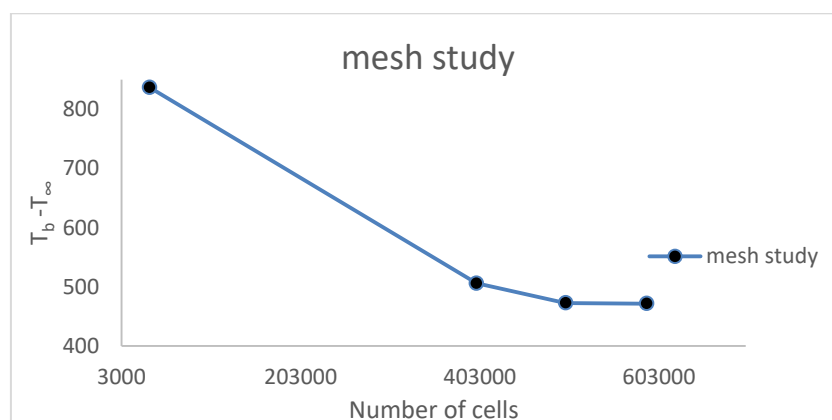

Figure 3. meshing independence study

4. Results and Discussions

Figure 4 presents the variant of the Nusselt number with different values of heat input for changed mini channel heat sink. The graphs visibly represent that the Nusselt number increases with the increase in heat input regardless of the shape of heat sink. It may be due to the fact that as the heat input to the sink higher the influence of convective rate of heat transfer increases more. The maximum value of the nusselt number is 47.647 correspond to embossed heat sink at angle 45° , while the minimum value of nusselt number is 24.5 correspond to smooth heat sink.

A significant improvement in the Nusselt number values can be seen by the use of inclined impressions on the fin surfaces for all the values of heat supply.
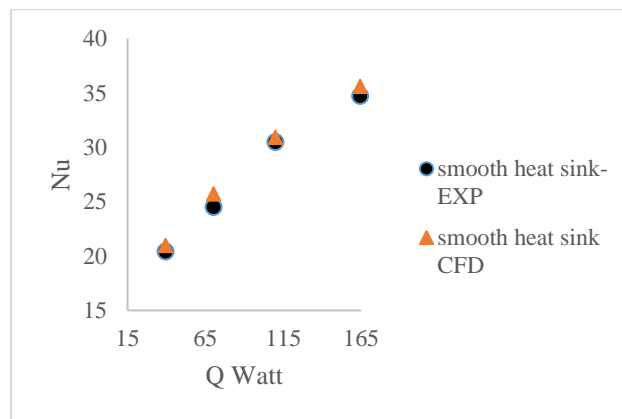

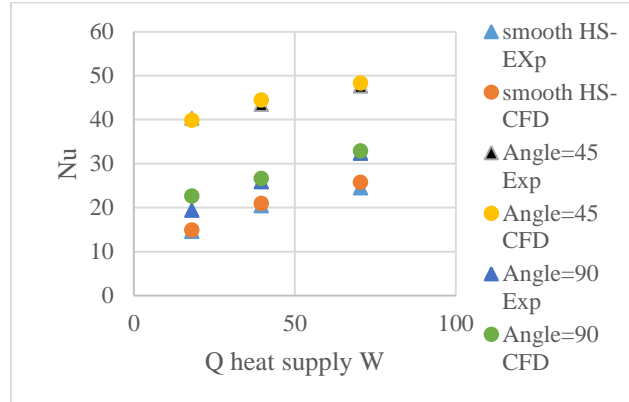**Figure 4.** nusselt number and heat input of smooth heat sink.

Figure 5. Nusselt number and heat input for different impression angles at Pi = 12 mm.

Figures 5 explains the consequence of impression angle on Nusselt number matching to the impression angle of 45 and 90 degree and smooth, for the different values of heat source. Same kind of trend is noticed in these graphs excluding the fact that the Nusselt number of embossed heat sink is more changeable in heat supply for the impression angle of 45 degree comparing to other values of angle. The existence of artificial geometry on the fin surface leads to greater Nusselt number due to the high energy fluid flow mixing near the heat sink wall by the separation and reattachment of the boundary layer. Also the restart of fluid film between the two successive impressions, the secondary fluid flow along the declined impressions interacts with the stream flow and so increases turbulence and heat transfer rate at the interface of fluid wall.

Figures 6, 7, and 8 represent the temperature distribution on the heat sink while figures 9 and 10 show the velocity contour at two different heat supplies. It can be seen from figures that embossed heat sink has better temperature than smooth heat sink. At heat supply 18 W, smooth heat sink, maximum temperature is 554.3 K, while it is 412.2 K at embossed heat sink. The minimum

temperature is 551.76 K and 399.87 K at smooth and embossed heat sink, respectively. Temperature at heat sink wall increases with heat supply, it is seen clearly at Q=70.4 W, the maximum and minimum temperatures are 859.79, 850.35 and 838.72, 629.46 K at smooth and embossed heat sink respectively.
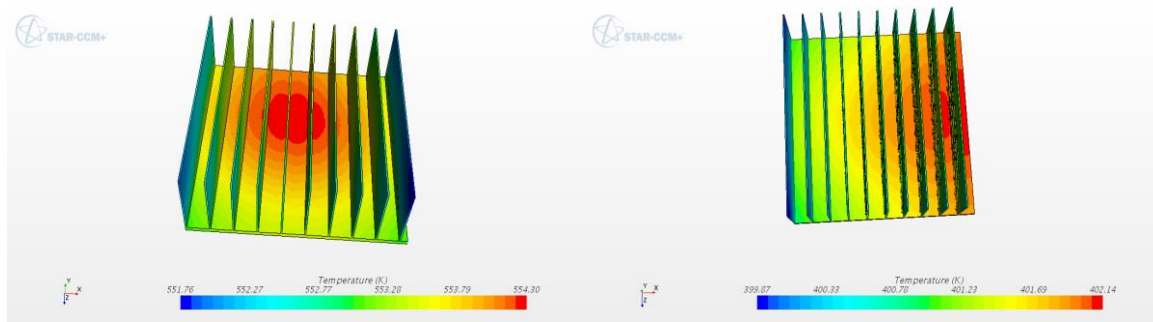

Figure 6. Temperature distribution on heat sink at Q=18 w, Smooth, Angle =45 respectively.
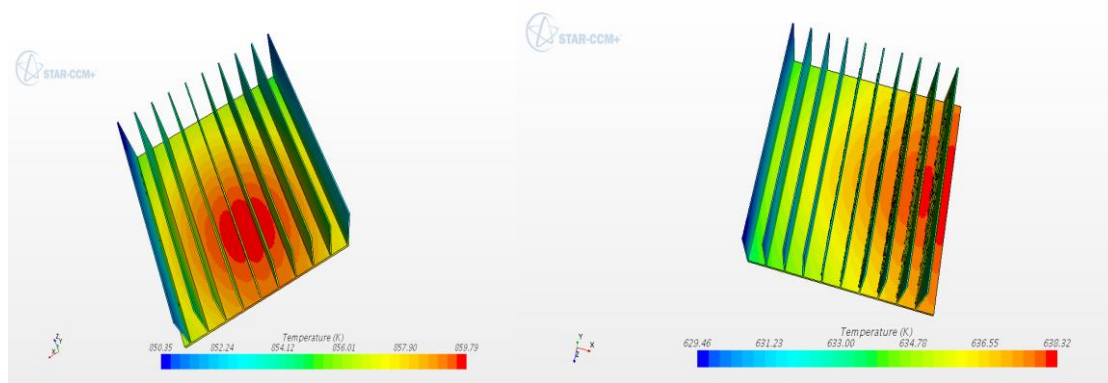

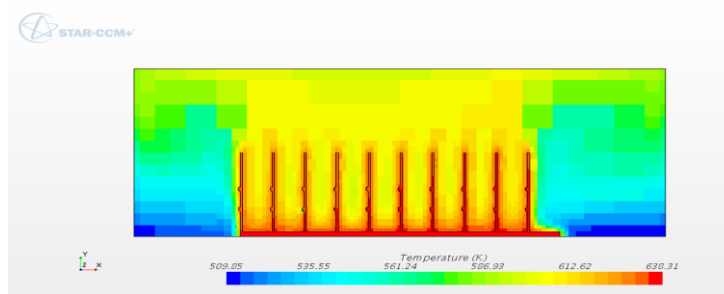**Figure 7.** Temperature distribution on smooth heat sink, Angle of impression is 45 at Q=70.4 w.

Figure 8. Temperature distribution at the middle of heat sink at Q=70.4 w, Angle of impression is 45.
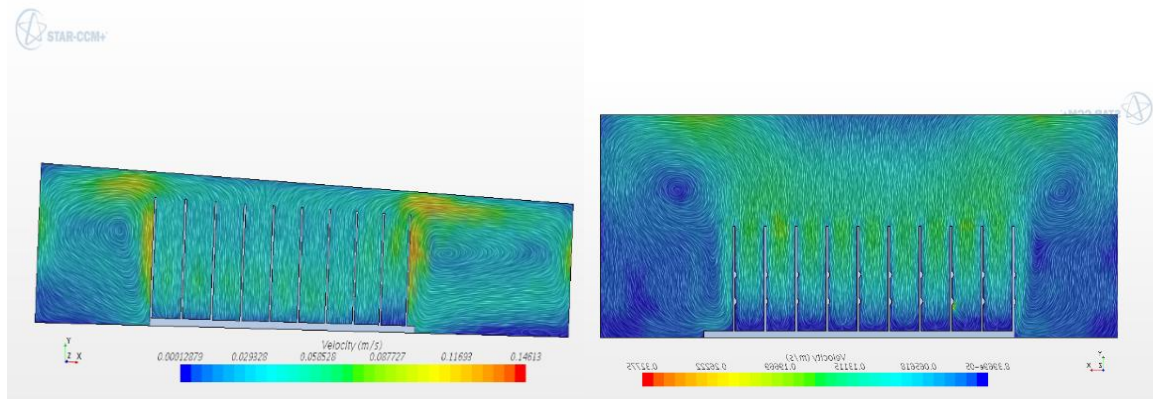
Figure 9. Velocity contour at Q=70.4 w, Smooth heat sink, Angle=45 at middle z=0.
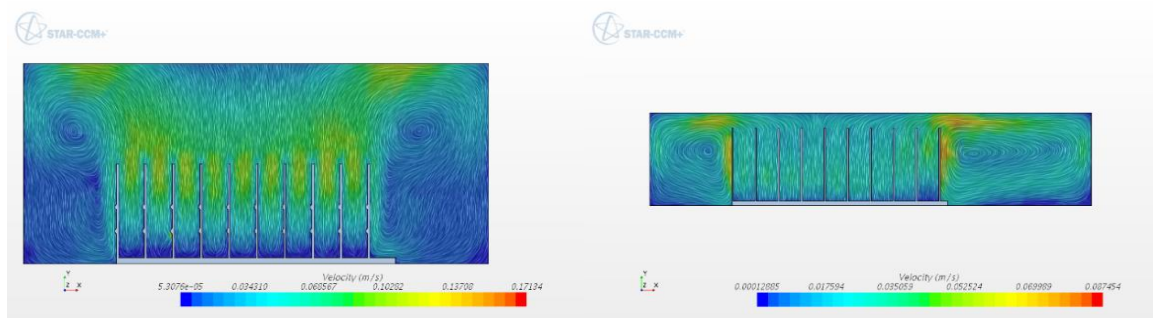
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Figure 10. Velocity contour at Q=18 w, Angle=45, Smooth heat sink and at middle z=0.

5. Conclusion

The hotness dissipation capability of the heat sink under natural cooling has been found to increase by the adding surface geometry on the fin surface. Computational evidences validated with experimental results explain that the variation in the impression inclinations angle has important result on the heat sink thermal performance. Based on the numerical study, it can be conclude:The value of the Nusselt number raised directly with the increase of heat supply to the base for all heat sink shapes.Remarkable improvement in the Nusselt number values has been seen for the embossed fins heat sink, which confirms that the nusselt number is a strongly affected by impression angle. the Nusselt number increased and reaches the maxima at 45 angle with the lessening at impression angle 90.The improvements in Nusselt number between 1.12   and 1.76 matching to the impression pitch and angle of 12 mm and 45, correspondingly.

Embossed heat sink better temperature distribution than smooth het sink, 102 K differnce between embossed and smooth heat sink.

References

[1]   Alhara T and et al 1970 Natural convection heat transfer from vertical rectangular-fin arrays: Part 2 heat transfer from fin-edges *Bull. JSME* 1108–1120.

[2]   Chaddock B 1970 Free convection heat transfer from vertical rectangular fin arrays *ASHRAE J* 12 53–60.

[3]   Chi-Chuan Wang K and et al 2011 Effect of cannelure fin configuration on compact aircooling heat sink. *Applied Thermal Engineering, 31* pp 1640-1647.

[4]   Hung-Yi Li and et al 2013 Enhancing heat transfer in a plate-fin heat sink using delta winglet

vortex generators *International Journal of Heat and Mass Transfer* 67 pp 666–677.

[5]   Yang J J and et al 2010 On the heat transfercharacteristics of heat sinks: with and without vortex generators *IEEE Trans.Compon. Packag. Technol 33 pp*. 391–397.

[6]   Singh P and Patil A. K 2015 Experimental investigation of heat transfer enhancement through embossed fin heat sink under natural convection. *Experimental Thermal and Fluid Science* 61 24–33.

[7]   Starner K and McManus H 1963 Experimental investigation of free convection heat transfer from rectangular fin arrays *Heat Transfer* 85 273–278.

[8]   Timothy Dake J M and et al 2009 Improving Flow Circulation in Heat Sinks Using Quadrupole Vortices *ASME Summer Heat Transfer Conference.*

[9]   Welling C W and et al 1965 Free convection heat transfer coefficient from rectangular vertical fins *Heat Transfer* 87 439–445.