Recent developments in the design of Heat Sinks for heat transfer enhancement – A review

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Abstract: Continuous and ever growing requirement of electronic industry for the production of compact and miniature electronic devices has led to the development of Heat Sink (HS) with enhanced heat-transfer capability. Challenges posted by electronic industry ask for the better heat transfer rate per unit area for the safe and efficient operation. In this regard, there is a necessity for the development of HS to remove additional heat generated per unit area in order to limit the temperature of electronic component below the maximum operating temperature. This paper intent to provide the recent developments in the design of HS. Size, shape, material, design etc, were used as the geometrical parameters. Nusselt number and pressure drop was used as the thermal performance parameter. Since pressure drop is one of the main parameter to be considered and cannot be neglected, the choice of particular HS design depends on the pumping power available and the cost considerations involved.

Keywords: Heat Transfer rate, Heat sink, Vortex generators, Nusselt Number.

1. INTRODUCTION

In the recent times, the most challenging task faced by the electronic industry is the dissipation of heat from the electronic components to keep the temperature below the maximum operating temperature. Miniaturization and increased performance has further increased the heat load on the electronic components. This heat has to be dissipated for safe and efficient functioning of those devices.

For a particular application and ambient conditions, the maximum operating temperature and the heat-transfer coefficient value remains constant. Thus in order to increase the heat-transfer, the only possible way is to increase the surface area.

\[ Q = hA(T_s - T_o) \]

Where;

\[ Q \] = Heat-transfer in W.

\[ h \] = Heat transfer co-efficient in W/m²K.

\[ A \] = Surface area in m².

\[ T_s \] = Operating temperature.

\[ T_o \] = Ambient temperature.

On the other hand, increase in surface area increases the overall size of the electronic component. This phenomenon is in contrast with the need of electronic industry for the production of miniature components. Hence, the use of additional attachments which can transfer the heat in a better way becomes necessary.

Even though the heat can be carried through free convection and radiation without any external energy input and attachments, forced convective Q with elongated surfaces are employed to cool those electronic devices due to the need for high heat transfer rate. Since air is available in abundant and the ease of operation makes it a preferable choice over other fluids to carry the heat generated.

Heat sink (HS) are that kind of heat exchangers with extended surfaces that are used to dissipate heat at a higher rate. These HS are mounted on to the electronic component that needs cooling. For effective transfer of heat from the component to the HS, thermal interface material such as thermal paste having higher thermal conductivity is added between them.

One important parameter that has to be considered in choosing a particular HS design is the pressure drop across the component. Due to the interruptions caused to the flow of air by the extended surfaces of the HS, there is a considerable chance of reduction in pressure across the flow area. This may result in need for extra pumping power to compensate for the pressure loss.
Hence, the process of designing a HS is a complex endeavor, since several parameters are involved. In this paper, the recent innovations and the developments happening in the design area of HS's are discussed.

2. HEAT SINK DESIGN DEVELOPMENTS

Design of HS is never a tedious job, as the designer has to look into various design aspects and criteria such as the surface area (A) and thermal resistance ($R_{th}$) of the HS, pressure drop ($\Delta P$) across the flow, cost of production etc.

T Saravanakumar et. al. [1] in their investigation on Q characteristics of HS have designed the HS with baffle attachment. The analysis were carried out for plate fin HS (PFHS) with and without baffles experimentally and validated numerically using commercial CFD software at different heat inputs (80W, 100W and 120W) and air velocity ($R_e = 1000$ to 4000) to study $R_{th}$ and temperature distribution. Copper strips were used in the preparation of baffles. These baffles which act as vortex generators were placed in between the PFHS channels to promote turbulence and mixing in the flow. The results showed considerable decrease in $R_{th}$ with increase in $R_e$. It is found that the value of $R_{th}$ for the PFHS with baffles is less as compared to the PFHS without baffles. The results show that there was about 12.9% increase in $Q$ with the use of HS attached with baffles as compared to the plain one. Even though there is considerable boost in $Q$ value, $\Delta P$ value as stated by the author is on the higher side due to increase in the obstruction for flow. This higher $\Delta P$ (over 14%) resulted in no improvement in the profit factor, thus the profit factor decides the use of PFHS with or without baffles. The difference in the thermal resistance found between CFD and experimental results was 4.75%.

[Figure 1: PFHS with baffles]

Y. Y. Ho et. al. [2] in their work analyzed the novel airfoil HS for convective Q performance, which were produced by Selective-Laser-Melting [SLM] process. HS having NACA-4424 and NACA-0024 airfoil with staggered configuration were analyzed and the results were compared with rounded rectangular fins and circular fins. Addition to the above cases, consequence of angle of attack ($\alpha$) on thermal performance of HS was analyzed. For this purpose NACA-0024 airfoil with angle of attack ranging from $0^\circ$ to $20^\circ$ ($0^\circ$, $5^\circ$, $10^\circ$, $15^\circ$ and $20^\circ$) were fabricated. Experiments were carried out in a rectangular air flow channel with $R_e$ ranging from 3400 to 24000. At first computational analysis was carried out to validate the experimental results of the circular fins and good agreement between the two methods was observed. As compared to circular fin HS, the rounded rectangular, NACA-0024 and NACA-4424 HS produced better thermal performance with $\text{Nu}_{th}$ (Nusselt number based on HS base area) found to be 34.7% , 29% and 28.5% higher respectively. In addition, it was observed that the heat transfer enhancement for NACA-4424 and NACA-0024 decreases with increase in $Re$ value, while for the rounded rectangular HS remained relatively constant. $\text{Nu}_{th}$ (Nusselt number based on total heat transfer area) is found to be 20.8% , 34.8% and 34% more for rounded rectangular, NACA-0024 and NACA-4424 respectively. Thermal performance of HS with NACA-0024 airfoil is found to increase with increase in $\alpha$ due to vortices generation and fluid mixing. Significant changes in thermal performance are noticed for $\alpha$ greater than $10^\circ$. The results showed that streamlined airfoil geometry produced less air flow resistance. Finally, the authors concluded that correlations and experimental outcomes obtained from the work provide predictive tool and data for using airfoil HS for thermal management.

[Figure 2: Airfoil HS]

Gaofeng Lu [3] analyzed Q and $\Delta P$ of a micro channel HS (MCHS) with the combination of dimples and vortex generators (VGs). The use of dimples and VGs in combination in a HS has improved the Q value. The use of VGs has resulted in the decrease in $R_{th}$ with moderate $\Delta P$ value. The analysis was carried out numerically. Following cases were studied; MCHS having only dimples, MCHS having only VGs in a common flow up arrangement, MCHS with dimples and VGs in a common flow down arrangement and finally MCHS with dimples and VGs in a common flow down arrangement. These dimples and VGs were used on the side walls of the MCHS. The VGs were kept at a distance of...
2.5 mm from the entry, since placing the VGs at the entry had very small effect on the improvement of non uniformity in the temperature distribution. The working fluid used was deionized water with Re ranging from 167 to 834. The results showed that the use of dimples and VGs has increased the Q value by 23.4% to 59.8% with the increase in friction factor of 22.1% to 54.4%. It was found that the magnitude of vortices generated with the use of VGs was much higher as compared to the one generated by dimples. Furthermore, MCHS with dimples and VGs in common flow down arrangement showed better thermal performance as compared to other cases.

**Figure 3: HS with dimples and VGs**

Hamed Mousavi et al [4] investigated the Q value for free convection and radiation numerically from vertical finned HS. The continuous finned HS 3D simulation was validated with the experimental outcomes available from the previous study. The study-involving analysis of HS configurations of 10 types as shown in figure 4, consisting of capped, interrupted and staggered fins with different fin spacing (s) value. For each case the Q behavior and thermal performance by natural convection and radiation were estimated. Analysis on HS was carried out at five different heat fluxes (827, 1066, 1515, 1796 and 2066 W/m²). It was found that in all the cases temperature at bottom was less than the top portions of the HSs. Use of staggered fins acted as dam which interrupt the boundary layer thereby enhancing the Q value. Further in the next case the use of capped fins imposed two new phenomena, first one was additional no slip condition and the other was the increase in hot surface area of the HS, which accelerated more air than before. It was also found that the air velocity reduced with reduction in fin spacing. With the use of L – shaped capped fins there was increase in velocity effectively. It has to be noted that out of 10 configurations the velocity intensity was found to be lowest and highest for CCF2 and SSIF respectively. The temperature counters obtained showed that by reducing the fin spacing less than 3 mm would increase the temperature, which was not a favorable result. With the use of capped fins increased the heat transfer area without any change in volume of HS. Further it was observed that the configuration SSIF and then the configuration S2IF had maximum temperatures at the tip of the HS, while LCCF configuration had the lowest temperature. Further it was found that S8IF configuration had the highest Nusselt Number (Nu) since it had the highest magnitude of velocity and the use of staggered fins decreased the Nu value considerably. Use of L shaped capped fins increased the Nu value to a certain extent but was less as compared to CF configuration. The authors mentioned that the Nu value was not a actual parameter for assessing the Q happening from the HS. This was due to the fact that, Nu value was based only on the free convection value of Q and not the radiation value of Q. Final analysis showed that LCCF fins had the lowest temperature without any raise in weight, which showed that this configuration was the best design among all the cases.

**Figure 4: HS with different configurations of fins**

Mustafa Ozisipahi et al [5] in their numerical analysis on hydraulic and thermal performance of HS, investigated honeycomb HS using FVM based CFD solver. Honeycombs with 0.05 mm thick having hexagonal topology were used in the design of HS. The analysis was carried out for different values of width (W) and Length (L). For obtaining better contact between honeycombs fins, these honeycomb mesh were united between the aluminum strips. The fin thickness (t) was maintained at 6 mm. Spacing between the fins (Sy) (20 mm, 40 mm and 60 mm), Reynolds number (Re) (8000, 16000 and 25000) and fin height (H) (20 mm, 40 mm and 60 mm) were selected as design parameters. ΔP and Re of HS were selected as the thermal performance parameters. Total of 21 simulations were carried out to investigate those parameters. Computational outcomes obtained showed topping agreement with the experimental outcomes of earlier investigations, implying that the HS analysis could be carried out using CFD simulations. It was found that Re of honeycomb HS decreased by 19% with increase in H value. However, it is important to note that with increase in H value greater than 40 mm increased (19 %) to a greater extent. HS having lowest Sy value produced the minimum Re value.
This was due to increase in number of fins, thereby increasing the surface area. But, on the other hand due to increase in the obstruction for the flow of fluid, this configuration showed the maximum $\Delta p$ value. Further, increment in $Re$ value produced better $Re_0$ value, but it also resulted in increase in $\Delta p$ value.

Su Min Hoi et al. [6] conducted heat transfer numerical investigations for plate fin HS (PFHS) with fractal insert (figure 6). The analysis was carried at a $Re$ value of $7.3 \times 10^4$ with 3 fractal grids having $N$ number of fractal iterations. These included rectangular fractal grid (RFG), square fractal grid with 3 (SFG3) and 4 (SFG4) number of iterations respectively. This numerical model was first validated to check its agreement with the existing experimental data. The forced convection CFD heat transfer analysis was carried out on a plate fin HS having 9 fins at a heat flux value = $3.587 \times 10^5$ W/m² at the base. The numerical analysis result showed that the presence of fractal grid had significantly improved heat transfer in all the three cases with SFG3, RFG2, SFG4 at 57%, 51% and 43% improvements respectively. The fractal bar thickness ($t_0$) was considered to be an important geometrical parameter which influenced the strength of turbulence generated. It was found that large value of $t_0$ gave rise to greater value of turbulent intensity but, resulted in higher pressure drop value. Authors have suggested that fine tuning of heat transfer performance based on $t_0$ value develops energy sustainable configurations of the fractal, instead of varying inter fin distance ($d$) and blockage ratio. RFG2 and SFG4 configurations required about 29% less pumping power than other configuration. The preferred combinations of the fractal plate and number of fractal iteration were, RFG2 having $\delta = 50$ mm and $t_0 = 46$ mm , which gave $Nu$ value = $6.82 \times 10^4$. With the use of SFG3 configuration having $t_0=100$ mm and $\delta = 10$ mm or 25 mm resulted in a $Nu$ value of $7.07 \times 10^4$. Finally SFG4 of $\delta = 10$ mm and $t_0 = 100$ mm delivered $Nu$ value of $6.71 \times 10^4$. In all the instances $Nu$ value increased with increase in $t_0$ value. The authors concluded that the heat transfer phenomenon greatly depend on the interaction between the flow structures within fins and the insert configuration.

*Figure 5: Honeycomb HS*

Ayush Gupta et. al. [7] in their work to enhance the value of $Q$ and flow performance, investigated plate fin HS with protrusions and dimples. Nusselt number ($Nu$) and friction factor ($f$) were considered as the performance parameters, while depth ($D$), spacing between the dimples ($s$) and their type of arrangement (inline and staggered) were considered as the design parameters. Experiments were carried out at different $s/d$ and $D/d$ ratios for the Reynolds number ($Re$) value ranging from 6800 – 15200. Where ‘$d$’ was the diameter of the dimple. It was observed that in all the cases the $Nu$ value increased with increase in $Re$ value. Further the value of $Nu$ in case of staggered configuration was found to be more when compared to inline configuration, irrespective of the $Re$ value. The results showed that the HS with dimples produced more heat transfer compared to smooth one. This was due to flow disturbances caused by the dimples. $Nu$ had maximum value when $D/d$ was kept at 0.5. It was seen that with increase in $s/d$ ratio, the value of $Nu$ decreases. The value of $f$ increased with increased in $D$ value, this was due to higher-pressure drop occurring because of the formation of deeper vortices.

*Figure 6: PFHS with fractal insert*
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Susmitha Sundar et. al. [8] investigated the performance of circular base HS with fins having perforations and arranged in staggered manner under natural convection and radiation. Analysis was carried out numerically and validated experimentally at different base radius and heat flux values for HS with perforated and non-perforated staggered fins. Finning factor, orientation angle and porosity factor was considered as the performance parameters. Number of fins per layer (Nf), number of layers of fins (Nf_l) and the width of the fins were used as the geometry optimization parameters. The results showed that perforated staggered configuration had a thermal resistance of about 7% to 12% less and the mass had reduced by around 9% when compared with non-perforated staggered HS configuration at different orientation angle. The maximum deviations in the results between the numerical and experimental approaches was found to be less than 10% and acceptable.

Figure 7: PFHS with dimples

Sakkarin Chingulpiatk et. al. [9] studied the fluid flow and heat transfer characteristics of HS with laterally perforated plate fins (LAP-PFH’s) with different number of circular perforations (Np) (14, 27 and 75) and diameters (Dp) (10 mm, 7 mm and 4 mm) as the geometrical optimization parameters. The results of Solid fin HS (SFHS) and LAP-PFH, which were analyzed computationally, was validated with the experimental results and measured data from the existing literatures. The difference in percentage of results obtained from these procedures were around 3.6% and 5.3% with respect to ΔP and Rth respectively. The numerical analysis of the SFHS and LAP-PFH showed that LAP-PFH showed favorable results compared to SFHS. The value of Q in case of LAP-PFH with Np = 75, Dp = 3 mm, was 11.6% more when compared to SFHS. The authors have also claimed a reduction in volume of LAP-PFH to be around 28% when compared SFHS.

Figure 8: Radial HS with circular base

Ammar A. Hussain et. al. [10] in their numerical investigation of heat transfer enhancement in plate fin HS (PFHS) studied the effect of fillet profile and flow direction i.e; optimum position of the fan. The numerical results were validated against the experimental data available from the previous studies. The maximum discrepancy between the numerical results and experimental results were found to be 12.4% and 8.8% for pressure drop and thermal losses respectively. The PFHS with and without fillet were compared in the study. Analysis was carried out at four different mass flow rates (0.00092 kg/s, 0.00218 kg/s, 0.0033 kg/s and 0.00433 kg/s). The PFHS with fillet profile proved to produce more heat transfer rate compared to PFHS without fillet. This improvement in heat transfer rate was due to increase in heat transfer area. This improvement in heat transfer rate resulted in decrease in base temperature of the PFHS by 3.78% and thermal resistance by 5.18%. Further the analysis was carried out to investigate effect of flow direction on the heat transfer performance. It is found that parallel flow of fluid showed more uniform and effective temperature distribution as compared to impinging flow. The base temperature and thermal resistance in case of parallel flow was found to be 3.85% and 5.29% lower as compared to impinging flow. The investigation resulted in the overall improvement of 18% in thermal performance for the proposed design (PFHS with fillet and with parallel flow) as compared to conventional design.

Figure 9: PFHS with lateral perforations
3. HEAT SINK ANALYSIS PROCEDURE

Similar to HS design, HS analysis procedure also play a prominent role in finding out the effectiveness of the HS. Most of the works carried out in the recent times involved numerical or computational analysis due to the complications and limitations in the fabrication of experimental model.

In most of the numerical analysis 2nd order upwind scheme was used to solve the energy and momentum equations. The pressure equation was discretized with PRESTO scheme and k-ε turbulence model was used to take care of turbulence term involved in the equation. Convergence criteria for energy equation were restricted to $10^{-6}$. Residues of other parameters were set to $10^{-6}$. Under relaxation factors for pressure, thermal energy and velocity components were set to 0.3, 1 and 0.7 respectively. Grid independence study was carried out to check the independency of grid size on the numerical result. Constant heat flux, velocity inlet, pressure outlet, no slip (adiabatic wall) were used as the boundary condition.

Even though numerical analysis are at most use, experimental analysis have also proved their importance with modern day testing and analysis equipments. The experimental works as stated in the literature used forced convective and free convective method to carry the heat from the sink. Forced convective environment was created with the help of blower. $K$ type or $T$ type thermocouples were used to measure the temperature of the HS, heat source, inlet fluid temperature and outlet fluid temperature. Micro manometer was used to find out the $\Delta P$ across the HS. Cartridge heaters were used to provide necessary heat input. Suitable data acquisition system was used to collect and store all the measured data. Complete analysis was carried out in a duct with proper insulation to take care of heat loss.

CONCLUSION

Present paper is intent to provide recent developments in the design of HS for heat transfer enhancement and minimize the pressure loss. It was found that the geometry of HS plays very important role in the improvement of thermal performance. Size, shape, material, porosity and design, were used as the geometrical optimization parameters. Nusselt number and pressure drop was used as the thermal performance parameter. Study showed that most of the work has been carried out in the forced convective environment and there is huge scope for further development in the thermal performance. Numerical analysis is proved to be reliable and there is huge potential for improvement to match up with experimental results. Most of the design proved to provide better thermal efficiency but with penalty of pressure drop. Thus, it is recommended to choose a particular HS design based on the available pumping power and the cost considerations involved.

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REFERENCES


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