Active fin structure for electronic device cooling using gap closing actuator

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The importance of cooling an electronics device increases these days with a vast amount of interest. According to this issue, a design of an active fin structure using the motion of gap closing actuator is suggested. A gap closing actuator is designed on a 5X5cm substrate. The thermal performance of this structure is calculated, based on a lumped capacitance method. 30 times increased performance is achieved through this structure compared to a plain heat sink with a mainstream air velocity of 5m/s. Because of the efficient heat pumping mechanism of active fin structure, both important factors, thermal performance and overall size of the heat sink is improved.

Introduction

Nowadays, heat generation from electronic devices is one of the critical problems that we have to solve. In a certain electrical circuit, a good performance in an aspect of thermal condition, requires a functional temperature limit. Out of this functional temperature limit, the probability of a cause of logic error, unexpected diffusion with phase transformation, thermal fatigue and plastic deformation in the system increases, which may result in a mis-operation of the component.

The maximum functional temperature to ensure operation and safe performance for an electronic device is typically under 100°C. For example, the Intel 850 chipset allowed the operating temperature to 50°C, with a recommended airflow over the package for 150 LFM(linear feet per minute). The thermal solution, which is applied to this Intel 850 chipset, is a passive simple extruded heat sink with thermal and mechanical interfaces. Additionally to these kinds of cooling methods, to improve the system cooling characteristics, a redesign of fans, vents and ducts can be considered.

But because of a vast development in the IC technology, electrical devices appears in smaller scale with a higher performance ability, which come along with a general aspect that, the more heat is generated as the size of electronic device is getting smaller.

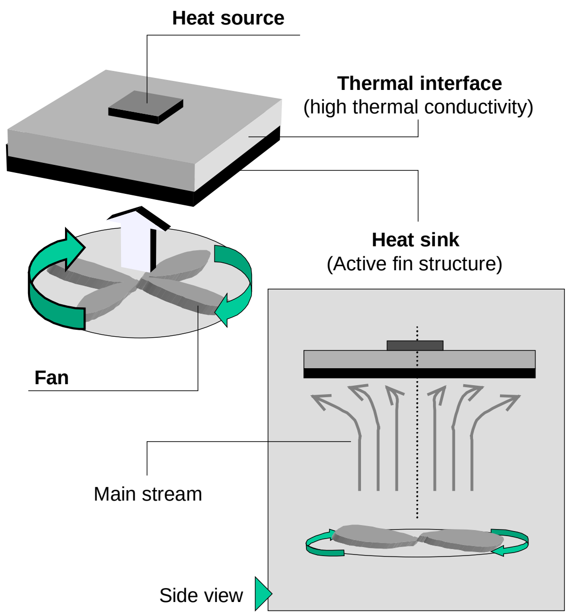

Fig.1 Generic configuration of active fin with heat source

Which provides a need of dissipation of heat with an enhanced cooling method. The recent research in this area suggests a few ways for a solution. One is using a fin structure with am additional fan, another is using a heat pipe, which is considered specially for laptop cooling. A cooling system by using micro-fluidic channels is also suggested, which has the advantage that heat transfer coefficient can be increased. Since heat transfer coefficient is inversely proportional to hydraulic diameter of the channel or it can be larger than single phase convection heat transfer case by order of magnitude if phase changed is used. But all of the above methods of cooling are "passive", which provided a limit in the effectiveness.

Based on these aspects, a new method is suggested by using an "active" micro-fin structure. A gap closing actuator is applied for this active fin structure on a 5X5cm substrate. The substrate is attached to the heat spreader plate, which is attached to heat source with a fan as described in Figure 1. Hot air near the surface can be pumped out and cooler air fills in the space by the motion of the gap closer.

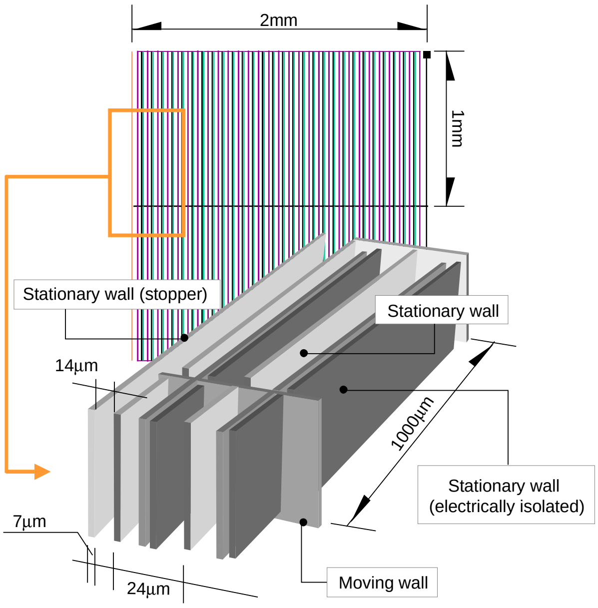

Fig. 3 Schematics of a basic unit active fin structure

Fabrication process and configuration

Figure 2 shows the schematic of fabrication process. For a high aspect ratio of approximately 30:1, a deep reactive ion etching (DRIE) is applied. Using a plasma source, DRIE can achieve a high aspect ratio with a side wall angle of 90°±2°. The etching rate is on the order of 2 to 3µm/min.

Additionally to a simple DRIE process a silicon fusion bonding (SFB) process is added. As shown in Fig2(d), a second wafer is fusion bonded onto the bottom wafer. After polishing down to approximately 200µm thickness. Followed by a patterning of a DRIE photoresist mask (e) and a DRIE etch through the second wafer (f).

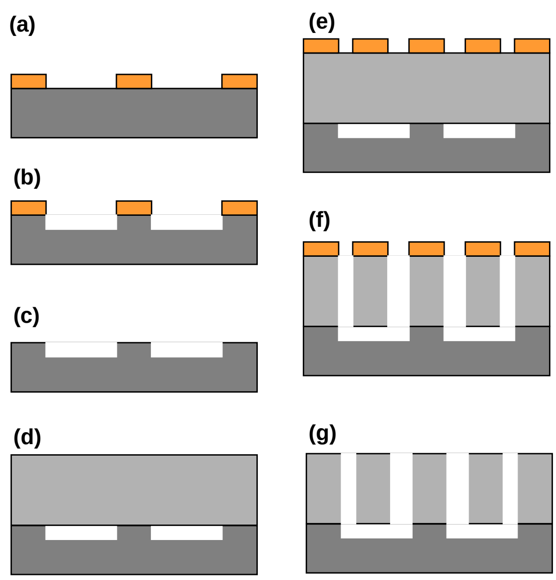

Fig.2 Schematics of the micro-fabrication process

Figure 3 shows a basic unit of the final layout. Each basic unit is in a size of 2x2mm. Based on this geometry 190 units can be placed on a 5X5cm substrate. An electrically isolated wall is also established to double the number of gaps. Based on this geometry, the total number of gaps in a 5X5 cm substrate is ~100,000, and the total effective volume of gaps (V_{tot}) is the multiplication of this number of gaps with the individual gap volume, which is 1.68×10^{-7} m^3.

Theoretical design and thermal performance calculation

Figure 4 shows the schematics of heat rejection during one cycle movement of gap closing actuator, the mechanism which is used in figure 3.
Based on this geometrical design, the calculation is done with the following approximated properties of air at 300K as follows:

Density (ρ) = 1.1614 kg/m3
Heat capacity (Cp) = 1007 J/kg.K
Thermal conductivity (k) = 0.0263 W/mK

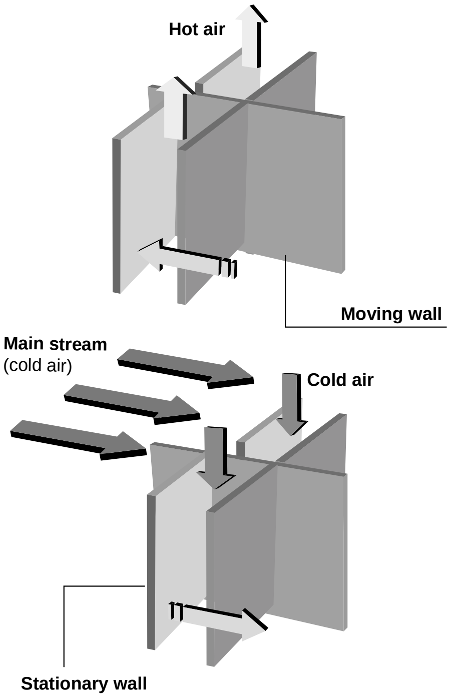

Fig. 4 Schematics of heat rejection during one cycle movement of gap closing actuator

To simplify the analysis, Biot number (Bi) is checked as follows:

$$Bi = \frac{hL}{k} < 0.1$$

Because of the smallness of characteristic length, L (on the order of 10^{-5}m) Bi is very small (Bi <0.1) for a wide range of h (heat transfer coefficient between the air and surface). It is, therefore, reasonable to assume a uniform temperature distribution across air in the gap at any time during a transient process (Lumped capacitance model). Then, the change in air temperature can be expressed as follows.

$$\frac{\theta}{\theta_i} = \frac{T - T_s}{T_i - T_s} = \exp[-(\frac{hA_s}{\rho Vc})t]$$

where, Ts is the surface temperature and Ti the main stream temperature

The above equation shows that the temperature difference between solid surface and air must decay exponentially to zero as t increases. The quantity ($hA_s/\rho Vc$) can be interpreted as a thermal time constant.

$$\tau_t = (\frac{1}{hA_s})(\rho Vc) = R_t C_t$$

where R_t is the resistance to convection heat transfer and C_t is the lumped capacitance of air.

Any decrease in R_t or C_t will cause air to respond more quickly to changes in its thermal environment and will decrease the time required to reach thermal equilibrium (θ=0).

To calculate the time constant, we need to know heat transfer coefficient, h which is very hard to know. But we can see that h will be much greater than usual case from the relation

$$Nu = \frac{hD}{k}$$

and D is very small (on the order of 10^{-5}). We can estimate the time ($t_{0.9}$) for air temperature to reach 90% of surface temperature as follows.

Thermal time constant, τ_t can be approximated as ~0.000016sec for h = 1000W/m^{2}K.

$$\frac{\theta}{\theta_i} = 0.1 = \exp[-\frac{t}{\tau_t}]$$

$$t_{0.9} = -\tau_t \ln 0.1 = -(0.000061\text{sec}) \ln 0.1$$
$$= 0.00003\,\text{sec}$$

Then, the time ($t_{0.9}$) for air temperature to reach 90% of surface temperature will be much smaller than 10^{-4} (the value corresponding to maximum repetition rate). When the two surfaces become closer, the air in the gap, which is already heated up almost the same temperature as that of surface, will be squeezed out mostly through the top. When the two surfaces moves apart, cooler air will be sucked into the gap between two surfaces and will be heated up almost instantly because of very small thermal time constant. This process will be repeated at a repetition rate of 10KHz.

Finally, total rate of energy transfer can be approximated as follows:

$$Q = (\rho V_{tot} C)(0.9\theta_i) / t_{rep}$$

Results and discussion

Figure5 is the results of thermal performance of active fin structure calculated from the above relation together with the calculated results on the plane surface with 5m/s mainstream air velocity. It shows that active fin structure has thermal resistance of 0.34^{o}C/W compared with 10.36^{o}C/W of plane surface, which means heat transfer rate can be increased as much as 30 times by using active fin structure.

In the analysis, the effect of main stream temperature increase is not considered. If a cooling

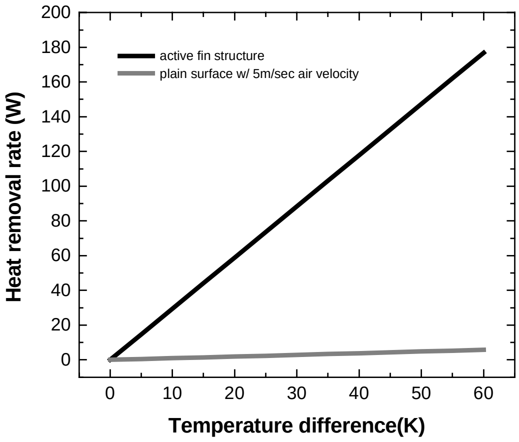

Fig. 5 Calculated thermal performance of fin structures

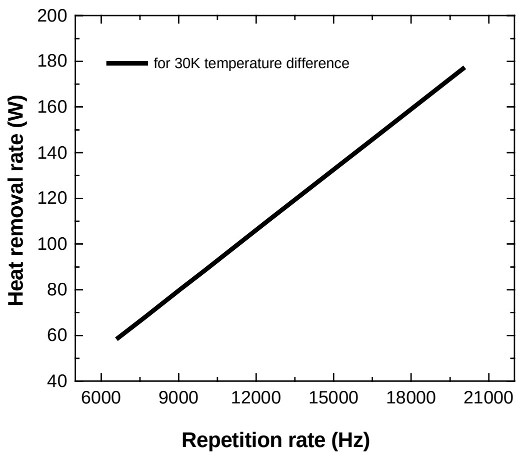

Fig. 6 The relation between thermal performance and repetition rate

fan is located right in front of the active fin structure and air speed is 5m/s, then the time for the air traveling from the center to the edge will be approximately 0.005sec. During traveling, air will be heated up and in case of including this heating effect, the calculated thermal performance will be decreased. However, this will be compensated if the convection heat transfer through the rest of fin area is considered since the heat rejection only through the gap area is considered in this calculation.

Figure 6 shows the results of relation between thermal performances and repetition rate of gap closing actuator. Which shows the greater the repetition rate, the better the thermal performance. But there is a limit in the maximum repetition rate with which a gap-closing actuator can be run (~10kHz). Even though we can go beyond this, there will be another limit that is imposed by the thermal time constant (It will take some time for the air in the gap to be heated up).

Conclusion

An active fin cooling structure is suggested. A gap closing actuator is designed on a 5X5cm substrate. The thermal performance of this structure is calculated, based on a lumped capacitance method. 30 times increased performance is achieved through this structure compared to a plain heat sink with a mainstream air velocity of 5m/s.

The achieved results can be summarized in two aspects. The first is a dramatic reduction in the volume of fin structures for an electronic device cooling, and the second is a higher effective cooling ability through “active” fin structure design compared to usual “passive” fin structures.

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