Advanced Cooling for Power Electronics

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Abstract

Power electronics devices such as MOSFET's, GTO's, IGBT's, IGCT's etc. are now widely used to efficiently deliver electrical power in home electronics, industrial drives, telecommunication, transport, electric grid and numerous other applications. This paper discusses cooling technologies that have evolved in step to remove increasing levels of heat dissipation and manage junction temperatures to achieve goals for efficiency, cost, and reliability. Cooling technologies rely on heat spreading and convection. In applications that use natural or forced air cooling, water heat pipes provide efficient heat spreading for size and weight reduction. Previous concepts are reviewed and an improved heat sink concept with staggered fin density is described for more isothermal cooling. Where gravity can drive liquid flow, thermosiphons provide efficient heat transport to remote fin volumes that can be oriented for natural and/or forced air cooling. Liquid cold plates (LCP's) offer the means to cool high heat loads and heat fluxes including double sided cooling for the highest density packaging. LCP's can be used both in single phase cooling systems with aqueous or oil based coolants and in two-phase cooling systems with dielectric fluids and refrigerants. Previous concepts are reviewed and new concepts including an air cooled heat sink, a thermosiphon heat sink, a vortex flow LCP and a shear flow direct contact cooling concept are described.

1 Introduction

Power electronics devices and systems are vital in the efficient generation, transmission and distribution, conversion, and a huge variety of end uses of electric power. More and more applications are adopting power electronics technologies to improve energy efficiency, reliability, and control and it is anticipated that in the future all electrical power will flow through a power semiconductor device at least once [1]. The state of art and future trends in power semiconductors and devices is reviewed in [2, 3]. Silicon remains the workhorse material for power semiconductors and to avoid device failure due to thermal runaway, effective cooling is critical. Figure 1 shows the maximum safe junction temperatures for silicon devices [2]. Wide band gap semiconductors like SiC and GaN offer the advantage of high temperature operation. However, available packaging technologies, passives and peripheral components, solder materials, reliability considerations and cost presently limit the junction temperatures to ~175 °C even though the semiconductor device can, in principle, operate at much higher junction temperatures [4]. The maximum safe junction temperatures in SiC could exceed 300 °C [5] so that even in high ambient temperatures, sufficient cooling may be provided by smaller and lower cost heat sinks resulting in improved volumetric power density.

A number of thermal management solutions in use for cooling power electronic modules in automotive applications are reviewed in [6, 7]. The coolant in such applications is available at temperatures above 100 °C so cooling must be accomplished with a low temperature difference between the semiconductor and the coolant. Many highly integrated cooling solutions are presented in [6] focused on the thermal management challenges in this severe application. Similar cooling concepts can be applied also to other applications like industrial drives, wind power converters, HVDC power transmission etc. depending on the constraints of each application. Various packaging designs and cooling solutions for reducing the thermal resistance of high power modules are described and compared in [8]. With micro-channel liquid cold plates [9, 10] the cold plate thermal resistance can be reduced to such an extent that the internal thermal resistance of the electronics package becomes the dominant thermal resistance. Phase change cooling using forced convection boiling of refrigerant fluids in cold plates is described in [11] including the more isothermal operation that comes from using the latent heat rather than the sensible heat of the coolant to provide the cooling effect.

![Figure 1 Critical thermal runaway temperature and estimated maximum safe operating temperature of Silicon devices](image-url)
Cooling solutions in use today include:

- Natural and forced convection air cooled heat sinks
- Single and double sided cooling with liquid cold plates
- Micro-channel liquid coolers built into power module base plates or integrated with the DBC substrate
- Jet impingement and direct contact liquid cooling of module base plates or DBC substrates
- Two-phase liquid cold plates with boiling of dielectric refrigerant coolant

With the exception of underwater or marine applications, heat removed from electronics systems is ultimately dissipated to the air. Systems with air cooled heat sinks dissipate heat directly into the air that flows through the system either by natural or forced convection. Systems that use liquid cooling transfer heat from the electronics into the liquid that in turn transfers the heat to air in a liquid-to-air heat exchanger such as an automotive radiator. In two-phase systems, heat from the power electronics transforms liquid into vapor by boiling or evaporation and the vapor is then returned to liquid phase when it transfers heat to the air in an air cooled condenser.

This paper reviews some of the existing cooling solutions and presents new concepts for an air cooled heat sink, loop thermosiphon heat sink, a liquid cold plate, and a direct liquid cooling concept. Approximate performance levels are also presented.

## 2 Power Module Packages

Figure 2 shows two common packages used for power modules. The IGBT module on the left is commonly used in applications below 6.5 kV and exposes one flat surface for “single sided” cooling. The other module is well suited for stacking in series for high voltage applications and provides both top and bottom surfaces for “double sided” cooling. Both these modules are designed to be mechanically bolted to heat sinks or liquid cold plates.

![Figure 2 IGBT packages for single and double sided cooling](image)

Interest in hybrid car and railway applications in recent years have led to customized packaging solutions with double sided cooling from Alstom [12], Denso [13] and others. These modules are shown in Figure 3. A number of other packaging concepts and related cooling solutions have previously been discussed in [6].

![Figure 3 Custom power module designs for double sided cooling](image)

Figure 4 shows a schematic of a standard power semiconductor package mounted on a heat sink for cooling and the corresponding thermal resistance network. The key parameter for device cooling performance is the junction-to-air thermal resistance, $R_{ja}$. This is the sum of the internal thermal resistance within the device package, $R_{jc}$, the interface thermal resistance from the device to the heat sink, $R_{cs}$, and the heat sink to air thermal resistance, $R_{sa}$.

![Figure 4 Schematic of standard IGBT package on a heat sink and key thermal resistances](image)

The heat sink thermal resistance can be expressed through the following equation commonly used in heat exchanger design literature [14].

\[
R_{sd} = \frac{1}{\dot{m}c_p} \frac{R_{jk}}{1 - e^{-\frac{R_{jk}A}{\dot{m}c_p}}}
\]  

An examination of equation (1) shows that the heat sink thermal resistance can be improved by increasing the mass flow rate $\dot{m}$ of the fluid through the heat sink (e.g. forced convection rather than natural convection), the heat capacity $c_p$ of the fluid (e.g. a liquid versus a gas), the heat transfer coefficient $h$ on the heat sink surface (e.g. smaller channel dimensions like micro-channels, turbulent flow using turbulators and boundary layer interruption rather than laminar flow, two-phase mechanisms such as evaporation and condensation rather than single phase convection) and the effective heat transfer area $A$. The effective area is the actual heat transfer area in contact with the cooling fluid multiplied by the heat transfer efficiency of the surface (e.g. fin efficiency.) Area enhancement by using closely spaced fins and small scale flow channels
(e.g. micro-channels) is the focus of many new developments in heat sink manufacturing technologies.

The term in the parenthesis in equation (1) is the effectiveness of the heat sink and is defined as follows

$$\varepsilon = \frac{T_a - T_i}{T_s - T_i}$$  \hspace{1cm} (2)

From equation (1) we can see that improving the efficiency of the heat transfer surface (making it more isothermal) will improve the effective area $A$ and therefore the effectiveness $\varepsilon$ of the heat sink. This is also evident in equation (2) by noting that a more isothermal surface implies a smaller value of $T_s$ in the denominator. The surface efficiency includes both the heat spreading resistance in the base and the fin efficiency of the fins.

### 3 Forced Air Cooled Heat Sinks

Table 1 shows an example of thermal resistances in the path from the IGBT dies to the air for the aluminum forced air cooled heat sink illustrated in Figure 5. The heat sink base is 236 x 230 mm in size and cools one econo-pack IGBT module (162 x 122 mm) with a DBC substrate soldered to a 3 mm thick copper heat spreading base. Aluminum fins are pressed into grooves in the aluminum base. The IGBT module is mounted on the heat sink using thermal grease to reduce thermal interface resistance. Air flow rate through the heat sink is 0.25 m$^3$/s at 25°C and 1 Bar and the IGBT dissipates a total of 2kW of heat.

<table>
<thead>
<tr>
<th>Stack-up layer</th>
<th>Thickness (mm)</th>
<th>Resistance (°C/kW)</th>
<th>Resistance Percentage of total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Silicon</td>
<td>0.2</td>
<td>0.6</td>
<td>1.3%</td>
</tr>
<tr>
<td>Silver</td>
<td>0.1</td>
<td>0.2</td>
<td>0.4%</td>
</tr>
<tr>
<td>Copper</td>
<td>0.3</td>
<td>0.2</td>
<td>0.5%</td>
</tr>
<tr>
<td>Al-Oxide</td>
<td>0.38</td>
<td>4.0</td>
<td>9.7%</td>
</tr>
<tr>
<td>Copper</td>
<td>0.3</td>
<td>0.2</td>
<td>0.4%</td>
</tr>
<tr>
<td>Solder</td>
<td>0.1</td>
<td>0.6</td>
<td>1.3%</td>
</tr>
<tr>
<td>Copper base plate</td>
<td>3</td>
<td>0.7</td>
<td>1.6%</td>
</tr>
<tr>
<td>Thermal Paste</td>
<td>0.05</td>
<td>4.2</td>
<td>10.0%</td>
</tr>
<tr>
<td>HS-Base spreading</td>
<td>20</td>
<td>12</td>
<td>28.9%</td>
</tr>
<tr>
<td>HS-Fins convective</td>
<td>110</td>
<td>19</td>
<td>45.8%</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>41.5</td>
<td>100%</td>
</tr>
</tbody>
</table>

Data in Table 1 shows that almost 30% of the thermal resistance is due to inefficient heat spreading in the base. With guidance from equations (1) and (2), the base can be made more isothermal to reduce the heat sink thermal resistance. Figure 6 shows an example of a heat sink with a copper base to improve heat spreading. For the same fin configuration as Figure 5, a copper base would enable an ~40% reduction in the spreading resistance in Table 1 from 12 °C/kW to 7 °C/kW or a 10 °C reduction in junction temperature.

Table 1 shows that the solder layer, copper base plate, and paste comprise 50% of the thermal resistance not associated with the heat sink. This is even more significant in liquid cooling where the liquid cold plate resistance is ~0.1 to 0.2X the air cooled heat sink.

Figure 7 shows an example of a heat sink with heat pipes embedded in the aluminum base. The copper-water heat pipes use copper powder wick and are embedded in the base using Aavid’s hi-contact technology. The heat pipes are oriented along the fin direction to even out the front to back temperature gradient due to heat up of the air. An ~60% reduction in the spreading resistance to 5 °C/kW or a 14 °C reduction in junction temperature can be achieved. The better heat spreading with heat pipes is achieved at a smaller heat sink mass compared to using a copper base. Recent advances using carbon nanotubes
grown on copper mesh and powder wick enable heat fluxes in excess of 500 W/cm² [15] and reduction in thermal resistance. Such nanotechnology is expected to be available in commercial products based on market demands.

**Figure 7** Air cooled heat sink with heat pipes embedded in an aluminum base

A further advance in heat sink performance is achieved if the press-fin joint is replaced by a metallurgical joint made by soldering or brazing the fins to the base. This decreases the thermal resistance by ~4 °C/kW enabling an additional ~8 °C reduction in junction temperature. Both the heat sinks pictured in Figures 6 and 7 incorporate such an improved base to fin joint.

A recent innovation consists of a heat sink design with fin density increasing in the flow direction as shown in Figure 8. The fin surface area is lowest in the upstream region where the cold air enters the heat sink and increases in a stepped fashion towards the downstream region as the air heats up. In this way, the base of the heat sink can be kept more isothermal so that power modules mounted at different distances from the edge where the air enters can be cooled to similar temperatures. Furthermore, the total pressure drop through the heat sink and the total mass is reduced because of the lower fin density in the upstream region. This type of design is manufactured by bonding or brazing the fins to the base.

**Figure 8** Advanced heat sink with increasing fin density along air flow direction

Figure 9 shows an air cooled heat sink that uses a loop thermosiphon to transport heat away from the immediate space around the power electronics package. An evaporator is provided with a flat surface where the power module is mounted and an enhanced boiling surface on the inside to enhance heat flux capability and reduce thermal resistance. The air cooled condenser is similar to an automotive radiator with vapor flow inside extruded flat aluminum tubes and aluminum folded fins outside the tubes to provide a large cooling surface area on the air side. The liquid in the evaporator of the thermosiphon absorbs heat from the electronics module and changes into its vapor phase. Vapor travels through the vapor tube to the condenser where it rejects heat to the air and condenses back into liquid phase. The liquid condensate then returns to the evaporator through the liquid return tube to complete the cycle.

Thermal resistance of the thermosiphon heat sink can be determined by using the heat transfer coefficient of the boiling surface inside the evaporator per Figure 10 and a condensation heat transfer coefficient of ~5.5 W/cm²-K inside the condenser tubing. Assuming a 122 x 162 mm IGBT dissipating 2 kW of heat, the heat flux on the evaporator wall is ~10 W/cm² and the boiling heat transfer coefficient is ~5.5 W/cm²-K. Including a 3 mm thick copper wall thickness, the evaporator thermal resistance is 1.3 °C/kW. Assuming a 230 x 230 mm by 30 mm thick air cooled condenser with 0.25 m³/s air flow rate (same as in Table 1), the estimated condenser thermal resistance is 9.1 °C/kW so that the total thermal resistance $R_{th}$ is 10.4 °C/kW, about a third of the 31 °C/kW value for the press-fin heat sink studied in Table 1. This huge improvement in thermal performance comes with the added benefits of

4 **Loop Thermosiphon Air Cooled Heat Sinks**

The heat sink concepts discussed thus far position the heat sink to air heat transfer surface area in the physical space adjacent to the power module. This can limit the electrical design and packaging flexibility of the power electronics system. Heat pipes have been used to extend the heat sink fins some distance from the heat sink base but when gravity can be used advantageously, two-phase thermosiphons can provide a higher performance solution. The simplest thermosiphons are just gravity aided heat pipes with a groove type wick surface. In this type of design, the vapor flow in the tube is in the opposite direction to the liquid flow along the wall of the tube. This limits the maximum power that can be carried because of liquid entrainment in the high velocity vapor flow [16]. A loop thermosiphon avoids this limitation by transporting vapor in one tube and returning the condensed liquid through another tube.
lower mass, lower pressure drop, and an essentially isothermal heat sink base for good static current balance between parallel IGBT dies.

**Figure 9** Air cooled heat sink using loop thermosiphon

The main limitation of the two-phase loop thermosiphon is that it is not suitable for moving platforms such as automobiles where external body forces other than gravity could potentially move the liquid out of the evaporator and cause dryout.

Since the working fluid inside the thermosiphon is dielectric, in principle the power semiconductor can be immersed inside the fluid in the thermosiphon. Figure 11 shows a prototype thermosiphon heat sink based on this type of “immersion cooling” concept [17]. Periodic two-phase thermosiphons can transport heat against gravity using vapor pressure to drive the flow [18].

**Figure 10** Heat transfer coefficient on thermosiphon evaporator boiling surface

**Figure 11** Thermosiphon air cooled system with power electronics immersed in dielectric fluid. IGBT is clamped between enhanced boiling plates.

### 5 Liquid Cooled Cold Plates

In liquid cooled systems, liquid cold plates provide localized cooling of power electronics by transferring heat to liquid that then flows to a remote heat exchanger and dissipates the heat to air or to another liquid in a secondary cooling system. Liquid cold plates provide more efficient cooling and enable greater levels of integration and major reductions in the volume and weight of power electronics systems. Existing and emerging liquid cold plate solutions in 2008 were extensively reviewed in [6] including:

- Tube type and fin type liquid cold plates to cool packages with DBC substrates with and without copper base plates.
- Liquid flow through fins formed directly on copper and AlSiC base plates.
- Single and double sided cooling using liquid jet impingement.
- Direct cooling of the base plate or DBC substrate using concepts such as the Danfoss “shower power” design using jet impingement or liquid flow through meandering channels.
- Direct double sided cooling of back to back modules with and without fins integrated with DBC substrates.
- Micro-channel coolers built into base plates or integrated with DBC substrate or into specially customized package designs.
- Stacked power modules and liquid cold plates using extruded channel type, folded fin type and micro-channel type cold plates.

The paper [6] showed the great potential for achieving very high levels of cooling performance and system integration using liquid cold plates.
As noted earlier, equation (1) guides us towards higher heat transfer coefficients using turbulent flow or small channel dimensions such as micro-channels to reduce thermal resistance. Two recently developed concepts that apply these principles effectively are presented next.

In many applications where liquid filtration is not desirable, flow channels are required to be of a sufficiently large size (~2-3 mm) to avoid clogging by particles and precipitates in the cooling liquid. High liquid velocities and turbulator inserts are commonly used to enhance the heat transfer coefficient in such large size channels including in tube type liquid cold plates. This type of approach provides limited opportunity for heat transfer area enhancement. An alternative approach developed recently [19] uses helicoidal flow paths to create strong secondary flow with high vorticity to achieve high heat transfer coefficients at the flow channel wall and parallel flow paths to reduce liquid velocities and the corresponding pressure gradient. Area enhancement is achieved by orienting the helical path so that its axis is normal to the base. This design concept, named the “vortex liquid cold plate” or VLCP, is shown in Figure 12 in the double sided cooling version. This design is ideal for cooling a stack of press pack IGBT’s, thyristors or diodes in series as shown in Figure 13.

Figure 12 Vortex Liquid Cold Plate in a double sided cooling version.

In the example considered to illustrate performance, an IGBT dissipating 1600 W is attached to each side of the VLCP with thermal grease at the interface. The IGBT base plate size is 140 x190 mm and uniform heat flux is assumed on the VLCP over the IGBT contact area. Cooling is provided by a water-glycol coolant with 50% glycol by volume flowing at 11 liters per minute (LPM) at a temperature of 80 °C at the cold plate inlet. Under these conditions, the maximum temperature on the cold plate surface is 91.5 °C, yielding a thermal resistance of $R_{cl} = 3.6 \, ^\circ\text{C}/\text{kW}$. Using equation (2), the effectiveness is 47%. By comparison, the thermal resistances of suitable tube type and offset folded fin type liquid cold plates for the same application conditions were 19 °C/kW and 5.6 °C/kW respectively. The pressure drop of the VLCP was 52 kPa versus 10 kPa for the tube type and offset folded fin type designs.

Figure 13 Stack of press pack diodes cooled using Vortex Liquid Cold Plates

For liquid cooling systems where liquid filtration is possible, high heat transfer coefficients may be obtained using high shear flow in narrow channels. Figure 14 shows a high shear direct contact (HSDC) cooling approach using a cooling plate that enables direct contact between the liquid and the surface to be cooled. An elastomeric O-ring (not shown) is provided between the cooling plate and the electronic package to seal the liquid flow volume. Liquid is distributed through a highly parallel manifold system and made to flow at a low velocity through narrow flow channels in direct contact with the surface to be cooled. Liquid is distributed through a highly parallel manifold system and made to flow at a low velocity through narrow flow channels in direct contact with the surface to be cooled. The combination of low liquid velocity and small channel height insures laminar flow in the channels. By adjusting the spacing between the liquid supply and return channels in the manifold, the flow length of the cooling channels and liquid velocity can be set to achieve the desired pressure drop characteristics.

Figure 14 Liquid cooling of Power Electronics package using High Shear Direct Contact concept
It is obvious that the HSDC cooling plate may be designed to provide double sided cooling so that power electronics devices may contact the cooling plate on both the top and bottom surfaces. Furthermore, depending on the requirements of the application, the cooling plate may be made of non-conducting plastic or electrically conducting metal.

An attractive feature of the HSDC design is that the incoming cold liquid is supplied uniformly to cooling channels over the whole cooling plate so that temperature gradients over the power electronics package are considerably reduced. In its simplest implementation, there is no area enhancement on the base plate. However, the design can work well with enhanced surfaces as well. Figure 16 shows the heat transfer coefficient on a flat base plate depending on flow channel height assuming fully developed laminar flow in the channels. The actual heat transfer coefficient will be higher because of entrance length effects.

6 Conclusions

Significant advances have been made in cooling technology for power electronics. This paper discussed improvements in applications ranging from air cooling to liquid cooling. A simple equation was described to help guide design choices. Copper bases and heat pipes embedded in bases can significantly improve heat spreading in air cooled heat sinks. Even more dramatic gains can result from using loop thermosiphons to more efficiently pick up heat over the full base area in contact with the power module and passively transport the heat for dissipation in an air cooled condenser. New liquid cold plate concepts are discussed that are well suited for single side cooled packages as well as double sided cooling of stacked press-pack type modules. Performance estimates are provided through application examples.

Emerging nanotechnology is mentioned that has the potential to significantly improve thermal performance of evaporative and boiling surfaces in future products. Although reliability of some technologies need to be proven, it seems that cooling technology will keep up nicely with increasing power dissipation levels and compactness of power electronics.

7 Acknowledgments

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8 Nomenclature

\[
\begin{align*}
Q & \quad \text{Heat Load (W)} \\
R_{ja} & \quad \text{Junction to air inlet thermal resistance } \left[= \frac{(T_j-T_a)}{Q} \text{ (°C/W)} \right] \\
R_{sa} & \quad \text{Heat sink contact surface to air inlet thermal resistance } \left[= \frac{(T_s-T_a)}{Q} \text{ (°C/W)} \right] \\
R_{sf} & \quad \text{Heat sink contact surface to fluid inlet thermal resistance } \left[= \frac{(T_s-T_f)}{Q} \text{ (°C/W)} \right] \\
R_{cs} & \quad \text{Thermal interface resistance } \left[= \frac{(T_c-T_s)}{Q} \text{ (°C/W)} \right] \\
T & \quad \text{Temperature (°C)}
\end{align*}
\]

9 References


