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Understanding Capacitance and Time Scales

Jim Wilson, Editor-in-Chief, March 2014

Watching Thermocouples Slowly change temperature was one of those important tasks I was given as a new thermal engineer. We were required to test some avionics hardware at the extremes of a large ambient temperature range and one of my responsibilities was to determine when we were close enough to steady-state conditions.

The test plan indicated we would verify that thermocouples mounted to the system were changing at a rate of less than a certain temperature change/hour (one degree C if I remember correctly) but what I do remember is the test going into the late hours and wanting to calculate the temperature/time derivative over shorter and shorter time scales so we could execute the test and go home. This particular system was somewhat insulated from the external environment to mitigate aerodynamic heating effects and also had significant thermal capacitance. Establishing equilibrium between the thermal mass of the system and the ambient required a long time scale so we learned patience and an appreciation for thermal capacitance. This lesson has found application over time when interpreting thermal test results, especially when insulation is used to mitigate heat loss from part of the system.

An understanding of capacitance, or inertia to be more precise, and appropriate time scales is also important for effectively participating and functioning in a work environment. As thermal engineers, we should well understand an analogy between the inherent inertia of a well-functioning design team and the appropriate time scale of measuring team performance. A common conflict is that management desires a measure of performance over a time scale that may not be consistent with long term needs. For example, training and overhead support costs can be reduced and the team inertia will most likely assure that for a short term, the team is achieving good performance at a lower cost.

Unfortunately, a frequent scenario is the eventual decrease in team performance does not show up until the responsible manager has been promoted for his ability to reduce expenses. Efforts by a future manager to improve the team’s performance are complicated by the team inertia, which is initially trending towards deterioration, and results from corrective training are not apparent in the short term.

A reasonable question is what is the right time scale for assessing team performance? While the answer is complicated and beyond my engineering skill set, it is helpful for us as thermal engineers to recognize at least two factors influencing a desire for short term measurements. One is the business cycle and the need to show money is being made on a scale consistent with investor expectations. The other is the desire for quick feedback. It is not easy for us to learn when the outcome is not apparent or the outcome is measured on a scale that is longer than desired. For example, telling a child to eat vegetables so they will be healthy 50 years later is rather hard for them to learn. A time scale of that duration is just not relevant to their experience. We naturally desire quicker feedback so it is not unreasonable that the manager funding increased training wants to know how soon it will show benefits.

While it is easy to find faults with management decisions, I would like to make this editorial a bit more personal. Since this is the first issue of 2014, it is a good time to assess your own career development and make sure you are continually finding opportunities to learn and improve your skills. If you were fortunate to have a good year in 2013, remember that we all have a risk of relying on past accomplishments and assuming that our credibility, or inertia, will carry us forward. Finding the right time scale for self-assessment can be difficult but it does need to be long enough to assure you are improving your skills.
Lower Power Consumption under Extreme Conditions

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GALIUM NITRIDE CONTRIBUTES TO LASER COOLING BREAKTHROUGH

Researchers have discovered a new method of engineering gallium nitride that could help eliminate the need for current heat dissipation methods. Considered one of the most important semiconducting materials after silicon, GaN’s hardness, crystalline structure and wide bandgap make it ideal for a variety of applications in the optoelectronics industry. Now, researchers at Lehigh University believe GaN can be engineered so that light passing through the compound cools it instead of heating it, a phenomenon known as laser cooling or laser refrigeration that would eliminate the need for expensive cooling methods currently in use.

The technique relies on a phenomenon known as anti-Stokes photoluminescence, which refers to the small fraction of photons whose frequency increases after hitting a material. Previous studies have shown no such effect. When we first saw that, we were intrigued.”

Source: Lehigh University

ELECTRICAL FIELD FACILITATES BETTER HEAT TRANSFER

The application of an electric field to a condenser could double the efficiency of surface heat transfer in power plant and high performance computing cooling systems, according to new research released by scientists at MIT.

The discovery builds upon previous research completed last year, in which the team found that a superhydrophobic surface produced via a specific nanopatterning of condenser surfaces caused water droplets to combine and leap from those surfaces as the result of a release of excess surface energy.

According to the researchers, the phenomenon improved the efficiency of heat transfer from condenser surfaces by 30 percent. Upon further examination, the team also noticed that the water droplets acquired a positive electric charge as they jumped away. “We found that when these droplets jump, through analysis of high-speed video, we saw that they repel one another midflight,” one researcher said. “Previous studies have shown no such effect. When we first saw that, we were intrigued.”

Source: MIT

RESEARCH COULD LEAD TO NEW THERMAL FLOW CONTROL DEVICES

Researchers have proposed a new method of controlling heat flow similar to the way electronic components handle electrical current that could deliver more efficient thermal management in a variety of applications.

Developed by scientists at Purdue University, the technology utilizes tiny triangular or T-shaped structures made of graphene nanoribbons to control phonons, quantum mechanical phenomena that illustrate how vibrations travel between atoms and can be used to determine the rate of heat flow throughout a system.

Advanced simulations conducted by the Purdue team have shown the triangular structures offer a way to control this heat flow through a process known as “thermal rectification,” which allows heat to flow more efficiently in one direction.

It could also provide a breakthrough in the development of thermal transistors. Engineers have begun to design and test thermal transistors with some success in recent years, but overall the work remains in its infancy. Unlike conventional transistors, thermal transistors would not require the use of silicon, are based on phonons rather than electrons and may make use of the waste heat already generated by most electronic devices.

Source: Purdue University

Datebook

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### CARBON NANOTUBES OFFER THERMAL STRESS RELIEF AT CRITICAL JUNCTIONS

Scientists at Stanford University have released new findings that demonstrate the benefits of using carbon nanotube arrays at critical junctions between two materials to help better relieve thermal stress.

Thermal stress, often caused by the joining of two materials that expand and contract at different rates as temperatures change, can lead to unnecessary strain and component damage if not mitigated properly.

Research indicates it has grown less densely together, seemed to have the best combination of flexibility, heat conductivity and strength for use in electronics and other industrial applications where thermal stress is expected.

**Source:** Stanford

### MAGNETIC NANOPARTICLES PREVENT HOTSPOTS IN COOLING SYSTEMS

An international team of researchers has discovered a way to enhance heat transfer using magnetic fields, a method they say could prevent hotspots that can lead to system failures.

The new system, which relies on tiny particles of magnetite, a form of iron oxide, is the result of several years of research on nanofluids and could be used to cool everything from small electronic devices to advance fusion reactors. The recent work involved experiments where magnets were placed on the outside of tubes containing magnetite nanofluid, increasing the heat transfer coefficient of the nanofluid up to 300 percent that of plain water.

**Source:** MIT

### NEW ULTRATHIN MATERIAL LEADS TO BETTER THERMALY CONDUCTIVE COATINGS

Scientists at Kansas State University have discovered a new ultrathin electrically conductive material they say may lead to advances in the efficiency of electronic and thermal devices.

They found that manipulating molybdenum disulfide (MoS2)—a three-atom-thick inorganic compound material—with gold atoms improved its electrical characteristics significantly.

The discovery is the latest step in the team’s research on the synthesis and properties of next-generation atomically-thick nanomaterials, such as graphene and boron-nitride.

MoS2 was recently shown to have better transistor-rectification than that of graphene, which is a single-atom-thick sheet of carbon atoms.

**Source:** Kansas State

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**MAY 8-9**

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Use of JEDEC Thermal Metrics in Calculating Chip Temperatures in Packages with Attached Heat Sinks

Bruce Guenin
Assoc. Technical Editor

INTRODUCTION

The previous column (from Dec. 2013) dealt with the application of JEDEC thermal metrics to IC packages that do not have heat sinks attached to them [1]. The methods are applicable to prediction of the junction temperature before the system is built and also to determining it from temperature measurements in the completed system.

When no heat sink is attached to a package, in a typical system, the dissipated heat is removed from the package and its associate printed circuit board (PCB) by a convection process (either natural or forced) supplemented by radiation cooling. Since, typically the PCB has a much larger surface area than the top of the package, most of the heat flows to the air via the PCB, as illustrated in Figure 1a. The figure identifies the relevant temperatures and their location to be used for calculating the appropriate thermal metrics, as described in the previous column. These temperatures are measured at the following locations: incoming air (TA), package top center (TT), board (TB), and the junction (TJ).

However, when a heat sink is attached to the top of the package, it enhances the flow of heat from the top of the package to the air, as shown in Figure 1b. In addition to the temperature locations measured when no heat sink is present, the temperature on the bottom of the heat sink, TS, is also measured. Note that, when a heat sink is present, it is customary to refer to the temperature of the top center of the package as the case temperature (TC).

Figure 2 depicts a thermal resistance network representing the two primary heat flow paths. The upward path has two thermal resistances in series: ΘJC, the junction-to-case thermal resistance and a second one representing the heat flow path from the top of the package to the air. When no heat sink is present, this resistance is ΘCA, the case-to-air thermal resistance. When a heat sink is attached to the top of the package, then this resistance is represented by ΘSA, the sink-to-air thermal resistance.

The downward heat flow path also has two resistances in series: the junction-to-board thermal resistance, ΘJB, and the board-to-air thermal resistance, ΘBA.
Electronics cooling heat sink

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**METHOD OF CALCULATING Θ_{JA} WITH HEAT SINK ATTACHED**

This column explores a method for predicting Θ_{JA}, with a heat sink attached, using the metrics measured on the package without the heat sink attached (Θ_{JA}, Ψ_{VB}, and Ψ_{JT}), the junction-to-case thermal resistance (Θ_{JC}), and the sink-to-air thermal resistance (Θ_{SA}), normally measured by the manufacturer.

Two different package configurations are evaluated: The first with two metal layers on the top and bottom surfaces of the package substrate. These are both signal layers. There are no metal planes. This nomenclature to describe this substrate stackup is: 2S0P. The second has two metal planes in addition to the two signal layers of the previous package. These planes provide much more efficient lateral spreading of heat than do the discontinuous signal layers. The stackup of this substrate is typically referred to as 2S2P. The thermal performance of the 2S2P package was studied in the previous column. Other details of the packages, test board, and heat sinks are provided in Table 1.

The packages are each attached to a four-layer JEDEC-standard board, containing two metal planes, for testing. In performing the Θ_{JC} tests, the top of the package is pressed into contact with a water-cooled cold plate. A compliantly-mounted thermocouple measures the case temperature.

Figures 3a and 3b illustrate the construction of the package. Figure 3a shows a top view closeup of the package attached to a PCB. The substrate is shown with the overmolded cap covering the IC. When the heat sink contacts the package, it is pressed against the top surface of this cap. The cap is made of low-thermal-conductivity molding compound, which makes a significant contribution to the measured value of Θ_{JC}. Figure 3b depicts the peripheral rows of solder balls that connect to the surface traces of the test board and the 6 x 6 array of thermal balls that are routed by vias to one of the two internal planes in the board to further enhance the thermal performance of the package. Figures 3c and 3d show the results of simulations of the Θ_{JC} test environment with each of the two package configurations. The temperature contours show clearly the superior heat spreading ability of the 2S2P substrate compared to that of the 2S0P one, leading to a lower value of Θ_{JC} for the 2S2P package. [Note that the cooling effect of the cold plate was represented in the model by impos-
the ambient air. It is assumed that most of the heat will flow to the top surface of the package directly from the die and a much smaller amount from the surrounding substrate. The value of $h$ can be calculated from the following expression, which is a rearrangement of terms of Eqn. 3 in the previous column [1]:

$$h = \frac{\Psi_{JT} \cdot \kappa_{EMC}}{\Theta_{JA} \cdot t_{EMC}} \quad (2)$$

where $\kappa_{EMC}$ and $t_{EMC}$ are the thermal conductivity and thickness above the die of the epoxy molding compound comprising the overmolded cap. Specific values of these parameters are provided in Table 1.

$\Delta T_{TA}$ can be calculated as follows:

$$\Delta T_{TA} = P \cdot (\Theta_{JA} - \Psi_{JT}) \quad (3)$$

where $P$ is the total power dissipated in the chip.” Next, $P_{Board}$ is calculated from

$$P_{Board} = P - P_{Top} \quad (4)$$

Finally, any thermal characterization parameter (psi value) can be converted into the equivalent thermal resistance.

| Table 2: Measured Values of Thermal Metrics with Calculated Values of $h$ and Power Partitioning. |
|---|---|---|---|---|---|---|---|---|---|
| Package | Laminate | $V_{air}$ (m/s) | $P$ (W) | $\Theta_{JA}$ (°C/W) | $\Psi_{JB}$ (°C/W) | $\Psi_{JT}$ (°C/W) | $\Psi_{BA}$ (°C/W) | $\Theta_{JC}$ (°C/W) | $h$, Top (W/m²K) | $P_{Top}$ (W) | $P_{Board}$ (W) |
| 2SOP | 0.5 | 2.0 | 14.5 | 0.4 | 5.8 | 18.0 | 0.11 | 18.0 | 2.89 |
| | 1 | 18.9 | 14.2 | 0.5 | 4.7 | 26.4 | 0.15 | 2.85 |
| | 2.5 | 17.1 | 13.8 | 1.0 | 3.3 | 53.0 | 0.26 | 2.74 |
| 2S2P | N/A | 6 | 6.1 |

| Table 3: Final Thermal Resistance Values for Input into Thermal Resistance Network. Final Up-Path and Down-Path Resistances. % Heat Flowing Out Top. |
|---|---|---|---|---|---|---|---|---|---|---|
| Package | Laminate | $V_{Air}$ (m/s) | Power (W) | HS, Thk (mm) | $\Theta_{CA}$ (°C/W) | $\Psi_{SA}$ (°C/W) | $\Psi_{JC}$ (°C/W) | $\Psi_{JB}$ (°C/W) | $\Theta_{BA}$ (°C/W) | $\Theta_{Up}$ Path (°C/W) | $\Theta_{Down}$ Path (°C/W) | % $P_{Top}$ |
| 2SOP | 0.5 | 3 | 0 | 555 | N/A | 6.1 | 15.0 | 6.0 | 561 | 21.0 | 4% |
| | | | 6 | N/A | 12.4 | 6.1 | 15.0 | 6.0 | 18.5 | 21.0 | 53% |
| | | | 15 | N/A | 5.5 | 6.1 | 15.0 | 6.0 | 11.6 | 21.0 | 64% |
| 2S2P | 0.5 | 3 | 0 | 446 | N/A | 5.5 | 11.1 | 6.1 | 451 | 17.2 | 4% |
| | | | 6 | N/A | 12.4 | 5.5 | 11.1 | 6.1 | 17.9 | 17.2 | 49% |
| | | | 15 | N/A | 5.5 | 5.5 | 11.1 | 6.1 | 11.0 | 17.2 | 61% |
(theta value) using the following relationship:

\[
\Theta_{XY} = \frac{P \cdot \Psi_{XY}}{P_{XY\text{ Path}}} = \frac{\Delta T_{XY}}{P_{XY\text{ Path}}} \quad (5)
\]

Thus, \(\Theta_{JB} = P \cdot \Psi_{JB} / P_{\text{Board}}\).

RESULTS AND DISCUSSION

Table 2 presents all of the theta and psi value test results for the package without the heat sink attached. It also provides the calculated values of \(h\), \(P_{\text{Top}}\), and \(P_{\text{Board}}\). One notes that the measured values of \(\Theta_{JC}\) are close to the simulated ones. The calculated values of \(h\) are consistent with handbook values [3]. Also, it is notable that the heat out of the top of the package increases with air velocity and is less than 10% of the total in all cases. This is consistent with the image in Figure 1a, which depicts the board functioning as an extended fin to transfer most of the dissipated power to the air.

Table 3 lists all the thermal resistance values needed for input into the thermal resistance diagram of Figure 2. It includes the calculated values of \(\Theta_{CA}\) for the package without a heat sink. Note that the values of these resistances are quite high, owing to the very small area of heat exchange at the top of the package (~ 100 mm\(^2\)). It is a very interesting contrast in the % of heat flowing out of the top of the package: no heat sink – 4%; with heat sink – 49-64%. The latter result would be expected since the thermal resistance for the up and down heat paths are nearly equal for the 6mm heat sink. For the 15 mm heat sink, the resistance for the up path is about 7/8 of that for the down path.

Table 4 lists the final calculated values of \(\Theta_{JA}\) for the situations in Table 3 and compares them to the actual test values. The calculated and test values for the packages without a heat sink are nearly equal. This is an simply an indicator that the arithmetic for these cases was correct. The results for the cases with the heat sinks are much more interesting. The model consistently predicts a value of \(\Theta_{JA}\) that is about 30% lower than the test value.

That there is such a large difference should not be much of a surprise. The model is explicitly based on the assumption that \(\Theta_{JB}\) is invariant, independent of the partitioning of the heat between the up and down paths. This is clearly not the case. As more heat is drawn out of the top of the package, more heat flux lines that had been flowing from the die to the board are diverted toward the top of the package. This would, essentially, leave a smaller number of flux lines flowing to the board. This modified path to the board would have a higher thermal resistance since the cross-sectional area of the metal participating in transmitting a smaller number of flux lines would be less, leading to a higher thermal resistance of this path.

It should be noted that the error is exacerbated by the fact that the up and down path thermal resistances are roughly comparable. If one of the resistances were dominant, then the error would be reduced.

A second effect is that the presence of the heat sink would divert the air flow from the top surface of the board downstream from the heat sink. This would reduce the effective heat transfer coefficient, leading to a higher \(\Theta_{RA}\) than that assumed in this model.

CONCLUSIONS

A method was demonstrated to integrate package-level and heat sink thermal measurement results in the calculation of the thermal performance of a package with the heat sink attached. The method has the benefit of being straightforward and intuitively satisfying. However, in the case explored here, the error was about 30% and overpredicted the final level of performance.

The fact that the competing heat flow paths were similar in magnitude exacerbated the error. Nonetheless, the user should be cognizant of the possibility of errors of this magnitude resulting from the use of thermal metrics that are not boundary-condition independent and exercise appropriate caution in interpreting the results.

The use of boundary-condition-independent thermal compact models involves more complexity. However, it is a much more robust method and is recommended for use in critical applications [4].

REFERENCES

When powerful technology gets smaller, things can really heat up. How do today’s leading companies develop designs that look great and solve the thermal management problem? Thanks to a higher level of thermal conductivity, aluminum extrusions can be as much as 53% more efficient than aluminum castings. In addition, Sapa has developed a new method of manufacturing high ratio air cooled heat sinks. Our new technique uses Friction Stir Welding (FSW) technology in a modular concept allowing for maximum flexibility and fin ratios in excess of 40:1. FSW also allows for the production of large scale heat sinks up to 20” wide.

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The purpose of the study was to characterize the thermal performance of the constituent components of an RF device package, as shown in Figure 1(a). The device, comprising of the resistive heater, was bonded to a metallic heat spreader with gold/tin (AuSn) solder. The spreader, in turn, was bonded to a Copper or a Copper-Molybdenum mounting bar with indium solder [1]. The mounting bar tab was bolted on to an Al baseplate. The objective of the test configuration was to characterize various spreader and thermal interface material (TIM) combinations to verify the optimal thermal packaging configuration. The resistive heater was used both as a heat source and a thermometer, the schematic of which is shown in Figure 1(b).

Three tests were conducted, with the details for different configurations given in Table 1. Test configuration 1 involved a 121µm x 350µm resistor, while tests 2 and 3 had larger resistors, 355µm x 5000µm each.

The temperature-resistance relationship of the heater was calibrated from 20°C to 120°C by adjusting the baseplate temperature while passing a small current through the resistor so not to generate appreciable self-heating. The resistance of the heater was calculated by the measured voltage drop and current for each test temperature, thereby establishing the temperature coefficient or resistance. As anticipated, the measured electrical resistance-temperature curve for these Pt resistors was essentially linear. Each R vs. T curve was represented as:
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Material properties were applied and heat load was modeled as a uniform heat flux over the resistor area. The Al baseplate fixed-temperature boundary condition was adjusted until the thermocouple location in the mounting bar matched the test temperature of 30°C.

RESULTS

Experimentally measured and numerically predicted thermal resistance results are compared in Figure 3 for three test configurations under consideration. Applied power has been normalized by the nominal electrical resistance at 20°C to enable comparison of device testing with different power dissipations. Observed experimental and analytical trends are as expected, with thermal resistance increasing due to the decreasing thermal conductivity of SiC with increasing temperature. However, analytical predictions consistently under predicted the experimental data, requiring further investigation into potential discrepancies.

The experimental and analytical set-ups were carefully evaluated to establish repeatability. The following potential causes were investigated as a cause of the difference between the measurement and analysis.

\[ R = R_0 [1 + \alpha (T - T_0)] \quad (1) \]

with \( R_0 \) being the nominal resistance at room temperature, \( T_0 \) of approximately 20°C; and \( \alpha \), the derived constant from calibration. Table 2 shows the measured quantities used to establish the \( R \) vs. \( T \) curve. The values of \( \alpha \) are in reasonable agreement with the reported numbers for thin Pt films [2].

With this relation established, testing was performed for the test configurations presented in Table 1. Each test involved gradually increasing the heat dissipation in the resistor in several increments such that the lowest applied heat load resulted in minimal ohmic heating, while the highest applied load maintained the resistor operating temperature at 130°C. For all tests, a temperature of 30°C was maintained at the thermocouple in the mounting bar for all power levels. The measured thermal resistance (mm²°C/W) was calculated using equation (2), normalizing for device footprint and enabling comparison of data collected with devices of varying footprint.

\[ (\text{Device footprint area}) \cdot \frac{T_{\text{thermocouple}} - T_{\text{resistor}}}{\text{Input power}} \quad (2) \]

ANALYSIS

Detailed finite element (FE) models were created to represent the set-up described in the “Experiments” section with the goal of replicating the experimental data and verifying understanding of constituent thermal properties. Figure 2 shows the details of the FE modeling performed with a commercial software [3].
ANALYSIS INDUCED POTENTIAL ERRORS:

1) **Modeling errors, software issues, inadequate mesh density** – Analysis was conducted in two different software packages [3, 4]. Model predictions were mesh independent and results were replicated between the two software packages.

2) **Application of power as uniform heat flux on the resistor area** – Multi-physics electro-thermal modeling [5] confirmed a non-uniform distribution of heat flux over the resistor area due to the temperature-dependent electrical resistance of the heater. Figure 4 shows the results for Test 1 configuration at approximately 8 volts or, 11 W of heat. Non-uniformity in current density is observed to be within a ±1.5% band around a mean of 1.275e+11 A/m², and ±6% for heat flux around a mean of approximately 3e+8 W/m². Larger current flow resides around the edges, producing a large heat flux along the periphery.

Despite a small non-uniformity in heat flux, the temperature profile predicted at the resistor was very similar between the thermal-only and electro-thermal models. Hence, treating the heat flux as uniform in thermal simulations was deemed adequate.
3) Uncertainty in the knowledge of material properties – Indium solder was estimated to be the primary contributor, and was a source of investigation as discussed below.

TEST INDUCED POTENTIAL ERRORS:
1) Heat loss from the test set-up to the environment – Any uncharacterized heat loss in equation (2) would result in the measured thermal resistance being lower rather than being higher than the model. Natural convection losses were assessed based upon a conservative heat transfer coefficient of 10W/m²K and found to be negligible. There were no parasitic conduction paths present in the experiment.

2) Unintended introduction of electrical resistance or leakage current - This was illustrated by electro-thermal modeling conducted two ways, 1) keeping the electrical voltage the same as tests and computing the electrical current, and 2) keeping the electrical current the same as tests and computing the voltage drop across the resistor. The results are shown in Figure 5 for test 1 only, but the behavior remained similar for all tests.

As evident in Figure 5, models predicted a larger current for the same applied voltage as compared to tests. One potential explanation for this result is current leakage, perhaps to another electrical resistance in parallel. This is also consistent with the prediction of a lower voltage drop for the same current, as if an unintended electrical resistance was introduced in series with the Pt resistor. The reader may note that although the V-I behavior of tests and model looks similar in Figure 5, even a small difference can be quite substantial in terms of temperature due to the very small temperature coefficient of resistance of 0.003 Ω/°C (Table 2).

If it is assumed that the same-voltage model (same as tests) best represents reality, an extraneous electrical resistance can be synthesized as a function of the missing current. This would, for example, be an electrical resistance due to poor electrical contact between the resistor and the underlying GaN layer to which it interfaces, among other possibilities. Figure 6 shows the predicted behavior of the potential extraneous electrical resistance, which exhibits good qualitative and quantitative correlation for all tests. On the other hand, if it is assumed that the same-current model represents reality, an additional resistance in series with the Pt resistor can be modeled, again shown in Figure 6. The behavior of the extraneous resistance appears to be consistent across different tests. The resistance in each case shows a strong non-ohmic behavior in which the slope of V-I curve is not constant.

Alternatively, if a similar curve for thermal resistance is presented, as in Figure 7, a fairly constant missing thermal resistance of 6 mm²°C/W is observed for tests 1 and 3, and about 10 mm²°C/W for test 2. The electrical resistance in Figure 6 (a) and (b) and the thermal resistance in Figure 7 are all equivalent from the standpoint of explaining the discrepancies between the experimental data and analyses. Any of those could be the source of observed discrepancy in Figure 3.

Figures 6 and 7 illustrate that electrical resistance can appear as thermal resistance in tests and vice-versa. There
is no easy way of distinguishing between the two, which leaves the experimentalist with the question as to whether thermal or electrical phenomena are responsible for the measured result, and therefore an inability to draw the desired conclusions on thermal performance.

To resolve this, SEM images of the device cross-section and X-Ray inspection was performed to evaluate the quality of the indium solder bond. As Figure 8 shows, indium solder was observed to be about 1.5-2X the expected thickness, was porous and exhibited voiding over approximately 50% of the area. These results and subsequent testing strongly pointed to a degraded thermal interface to be the most likely cause; however, it was not possible to definitively rule out the possibility of electrical phenomena influencing the measured thermal performance in experiments conducted using these custom resistor-thermometer devices.

**FIGURE 6:** Extraneous resistance calculations based on electro-thermal modeling for each test case, (a) same-voltage model prediction, and (b) same-current model predictions

**FIGURE 7:** Thermal resistance delta between the test-measured and model-computed values

**FIGURE 8:** SEM picture of the cross-section through TIM2 solder and X-ray scan of the plan view.

**CONCLUSIONS**

The use of electrical resistance is prevalent as means to simultaneously dissipate heat and measure temperatures of simulated micro-electronic devices in their packaged environment. It’s a time-tested technique that is robust and provides meaningful answers. However, just like any metrology method, it is not fool proof and there are pitfalls that an engineer must consider. This article summarizes some of the potential pitfalls, such as the intertwining of electrical and thermal resistance in certain situations, possibly leading to erroneous conclusions about thermal performance.

**REFERENCES**

Towards Reproducible ASTM D5470 Measurements at Lower Cost

Baratunde A. Cole
Georgia Institute of Technology

INTRODUCTION

ASTM D5470 (updated to ASTM D5470-12 in 2012) [1] remains an industry standard for characterizing thermal interface materials (TIMs) despite some drawbacks. Ultimately, a TIM must be tested in its application to conclude its merits. Standardized testing is useful, however, for comparing TIMs in research and development. It is also useful for customers to validate data provided by vendors. Several modifications to ASTM D5470 have emerged to improve the accuracy of TIM measurements, and to increase measurement precision [2, 3]. Such improvements, however, usually come with increased equipment cost, and they do not eliminate the need for testing in-applications, so the total cost for testing can be high. Further, reproducible measurements based on ASTM D5470 remain challenging to achieve, and recent studies to improve the accuracy of this standard have not focused on quantifying measurement reproducibility with large sample sets. The need for improved measurement reproducibility (e.g., a type of sample is measured, removed from the apparatus, and measured over many cycles with very similar results) is known to the community, and continues to limit cost-effective TIM development.

The approach to calculate the specific thermal resistance of a TIM based on ASTM D5470 measurements is

\[ R = \frac{\Delta T}{Q A_c} \]  (1)

where \( \Delta T \) is the difference between the upper interface temperature and the lower interface temperature of the sample, \( Q \) is the average heat rate through the TIM, and \( A_c \) is the effective contact area between reference bars and the TIM. The heat rate and interface temperatures are calculated from temperature measurements in the reference bars in contact with the TIM. The typical practice with ASTM D5470 is to use reference bars of equal cross sectional area [1]. The temperature profile in these bars is linear when they are insulated well, so a simple one-dimensional heat conduction analysis is used. But the simplified heat transfer analysis comes at the cost of making it difficult to align the reference bars in contact with the TIM. The typical practice with ASTM D5470 is to use reference bars of equal cross sectional area [1]. The temperature profile in these bars is linear when they are insulated well, so a simple one-dimensional heat conduction analysis is used. But the simplified heat transfer analysis comes at the cost of making it difficult to align the reference bars with each other and the sample – four surfaces must be aligned. Misalignments reduce the effective contact area between the sample and the reference bars as shown in Figure 1a. Misalignments as low as 1% often go undetected and are difficult to control, yet they can propagate significant errors through the system, and make reproducible measurements.

Baratunde "Bara" Cola is an assistant professor in the George W. Woodruff School of Mechanical Engineering and the School of Materials Science and Engineering at the Georgia Institute of Technology. He received his B.E (2002) and M.S. (2004) from Vanderbilt University, while a member of the Vanderbilt Football Team, and his Ph.D. (2008) from Purdue University, all in mechanical engineering. Bara has received prestigious early career research awards from DARPA (2009), NSF (2011), and the Army (2013), and received the Presidential Early Career Award for Scientist and Engineers (PECASE) in 2012 from President Obama for his work in nanotechnology, energy, and outreach to high school art and science teachers and students. Recently, Bara was awarded the 2013 AAAS Early Career Award for Public Engagement with Science. In addition to building knowledge and training students, his many published journal articles, book chapters, and conference proceedings have helped to produce 2 issued patents and several pending patents, which lead to Bara founding Carbice Nanotechnologies, Inc. in 2012 to commercialize carbon nanotube thermal interface materials. Cola’s work is currently focused on characterization and design of thermal transport and energy conversion in nanostructures and devices. He is also interested in the scalable fabrication of organic and organic-inorganic hybrid nanostructures for novel use in technologies such as thermal interface materials, thermoelcctrics and thermo-electrochemical cells, infrared and optical rectenna, and materials that can be tuned to regulate the flow of heat.
measurements difficult to achieve.

Because possible misalignments are usually neglected in uncertainty analysis, we could not find published experimental data that quantifies the role misalignments have in measurement uncertainty (this is done numerically in [4]). But our colleagues in industry and academia tell us that their data scatter for a large set of the same type of sample is often much larger than what their measurement uncertainty analysis predicts, even when considering reasonable variations in the properties of the samples. This is especially true for delicate nanomaterials (e.g., carbon nanotube TIMs [5]) and other samples that can be damaged while trying to optimize alignment of the reference bars.

This article highlights the “stepped-bar” modification of ASTM D5470 shown in Figure 1b. The modification can improve measurement reproducibility, while also reducing system cost. A comprehensive uncertainty analysis derived from the Kline and McClintock method for the stepped-bar ASTM D5470 is presented in [4]. This analysis shows that, in addition to high reproducibility, the stepped-bar approach can produce lower measurement uncertainty than the standard ASTM D5470 when very precise thermocouples (± 0.001 K) are used, and the mismatch in reference bar area alignment is greater than 1% in the standard approach. The reader is referred to [4] for a comprehensive description of the stepped-bar ASTM D5470 technique.

**STEPPED-BAR ASTM D5470**

The stepped-bar design oversizes the upper reference bar (URB) as shown in Figures 1b and 2. This modification allows machining errors and bar misalignments without a reduction in the contact area between the URB and the TIM. The operator only has to align the TIM to the lower reference bar (LRB) – a two surface alignment – rather than aligning the sample to both reference bars – a four surface alignment. This obviates the need for expensive bar alignment and loading mechanisms, which can often be the largest component costs in the system.

A mechanical transducer, comprised of a threaded rod and steel hand wheel, applies pressure to the TIM under test in the stepped-bar approach. The use of a mechanical transducer can greatly lower the cost (~50%) of the entire apparatus in comparison to a stepper motor or pneumatic-based loading system. Because instabilities of the threaded rod introduce imprecision in aligning the URB with the LRB, the oversized URB ensures that these misalignments do not reduce the contact area of the sample and introduce errors into the thermal resistance measurements. In the design shown in Figure 2, the URB has a square cross-sectional area measuring 1.4 cm to a side. And the LRB has a square cross-sectional area measuring 1.0 cm to a side. The URB and LRB are 1.5 cm and 3.0 cm long in the axial direction, respectively. Additional details of this design are reported elsewhere [4].

The major drawback of the stepped-bar approach is that heat flow is constricted from the URB to the TIM, resulting in a nonlinear temperature profile in the URB, so a simple one-dimensional heat conduction model cannot be used to determine the upper interface temperature. But this is addressed by fitting a 2nd order regression line through thermocouple readings near the TIM interface. Placement of several thermocouples near the TIM interface as shown in Figure 1b minimizes additional measurement uncertainty that could be introduced by the nonlinear regression near the interface.
Based on a numerical heat transfer analysis of the stepped-bar [4], the URB can be divided into two parts for curve fitting: 1) a linear fit to the top part of the URB, and 2) a 2nd order fit through the bottom of the URB. The linear fit to the top part of the URB is used to determine the rate of heat flow and the 2nd order fit extrapolates the upper interface temperature of the TIM. An example of this curve-fitting procedure is shown in Figure 3. The error bars arise from the precision of the thermocouples and their placement.

The oversized geometry of the URB, and dividing the URB into 2 regions for curve fitting are the major differences between the stepped-bar ASTM D5470 and the standard ASTM D5470. The temperature profile remains linear in the LRB for both approaches.

**STEEPED-BAR MEASUREMENTS OF TIMS**

The stepped-bar ASTM D5470 (aluminium 2024 reference bars with Ra ~ 0.5 µm, flatness = 0.1-0.3 µm) was used to characterize Arctic Silver 5 and Ceramique thermal paste [4]. A small amount of spacer-grade, soda lime glass microspheres (22-25 µm) from Cospheric was added to these pastes to maintain a constant bond-line thickness. Thermal resistances of 32 ± 4.5 mm²K/W and 41 ± 4.5 mm²K/W were measured at an

![Figure 3: Curve-fitting procedure in upper reference bar of the stepped-bar ASTM D5470 for a representative measurement. The slope of the blue dashed line is used to determine the input heat rate. The red curve is used to extrapolate the interface temperature. This figure is reproduced from [4] with permission from ASME.](image-url)
applied pressure of 300 kPa for Arctic Silver 5 and Ceramique, respectively. The United States National Renewable Energy Laboratory (NREL) reported a resistance of 30.9 mm²K/W for Arctic Silver 5 with a bond line thickness of 23.5 um [6], which is similar to the value measured with the stepped-bar. The NREL study used an experimental apparatus based on the standard ASTM D5470 with reference bars polished to 0.5 μm. This result indicates that the stepped-bar approach can be implemented with no significant loss in measurement accuracy.

The thermal resistance of aluminium foil 10 microns thick and three graphite samples from GrafTech (HT-1220, HT-1210, and HT-1205) were also measured with the stepped bar. The thermal resistance versus applied pressure for these samples is shown in Figure 4. The measurements on thermal paste, graphite, and aluminium foil span an order of magnitude in thermal resistance, and show a range of practical resistance that can be characterized with the stepped-bar approach.

Stockwell Elastomeric TC100 thermal gap (1.575 mm thick, manufacturer’s reported resistance of 12.12 cm²K/W [7]) pad was also tested. The gap pad was tested at the manufacturer’s specified pressure of 690 kPa. For each measurement, the sample was removed from the apparatus and reinserted to account for operator error in aligning the sample to the LRB. Figure 5 shows the frequency histogram for all measurements of the TC100 gap pad using the stepped-bar ASTM D5470 with stainless-steel reference bars. The average specific thermal resistance measurement was approximately 12.09 cm²K/W. The standard deviation of the eighteen measurements was 0.38 cm²K/W. A confidence interval based on the student’s T distribution suggests that 95% of all measurements of the TC100 gap pad fall within 6.4% of the mean value. These results imply excellent repeatability of the stepped-bar ASTMD5470 under these testing conditions. Likewise, the manufacturer-specified thermal resistance value of 12.12 cm²K/W falls within this 95% confidence interval, suggesting excellent measurement accuracy.

**CONCLUSION**

This article highlights the importance of accounting for reference bar alignment for reproducible TIM measurements. A stepped-bar modification to ASTM D5470 [4] is discussed and shown to produce very reproducible and accurate measurements of a commercial thermal gap pad. The thermal resistance of Arctic Silver 5 was also measured to be in good agreement.
with other reports, which used the conventional ASTM D5470. In order for the stepped-bar apparatus to achieve the same uncertainty as that of a conventional ASTM D5470 apparatus, more temperature probes are required. Each additional temperature probe increases the cost of the apparatus. However, the stepped-bar apparatus offsets this marginal expense because it utilizes an inexpensive mechanical transducer instead of more costly pneumatic-based alignment mechanisms that are required to precisely align reference bars of equal cross-section area. The stepped-bar ASTM D5470 is simple enough for a skilled technician to set up in one day when all parts are available, and the total cost of the parts is about $9,700 (for a parts list see [4]). This cost is about 3-5 times less than the cost of several commercial systems based on ASTM D5470.

A round robin of tests in different labs, and with several different TIMs, is ultimately required to conclude the superior reproducibility of stepped-bar ASTM D5470 approach. But there is a clear cost advantage, which could save resources for increased testing of TIMs in their application.

ACKNOWLEDGEMENTS

Assistance with data collection and analysis from Dakotah R. Thompson is appreciated.

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The engineer’s choice
Bitcoin 2-Phase Immersion Cooling and the Implications for High Performance Computing

Alex Kampl
Allied Control Company

ABSTRACT

RECENTLY, BITCOIN and Bitcoin mining have aroused international interest. This article describes the cooling challenges faced by Bitcoin miners and discusses the implications on the future of cooling for supercomputers, HPC clusters and other high density electronics. We explain the inner workings of the first commercial open bath immersion cooling systems and how Bitcoiners seemingly refine the technology used to create high powered ASICs, and keep them cool.

BITCOIN

Bitcoin is a peer-to-peer payment system and cryptographic currency introduced as open-source software in 2009 [1]. Users install a free client (the wallet) onto their computer, which allows them to send and receive payments using Bitcoin addresses. At the time of this writing, more than 50,000 payments are sent on a daily basis. Participants known as “miners” verify transactions into blocks and broadcast them to the public block chain by employing a proof-of-work concept using a secure hashing algorithm (SHA-256). Miners get rewarded with 25 new Bitcoins in return for each block. To stabilize the block creation rate at one block about every 10 minutes, the network self-adjusts the difficulty of the hashing calculations. The cost of verifying transactions is the capital cost plus the power consumption of the hardware in use. The more computational power is employed to do the “hashing”, the bigger the share of the reward that goes to the miner.

Miners initially employed conventional CPUs to perform the SHA-256 hashes. When GPUs (graphics processing units) were found to be more efficient, the cost to run CPUs quickly exceeded the profits that could be made with them. As Bitcoin grew in popularity, gamers realized that they were in possession of “money making machines”, and the arms race for the most efficient mining hardware was on. What is now known as the "Bitcoin mining industry", moved through several hardware generations within two short years. This included FPGA educational kits, custom FPGA hardware with “beefier” power supplies, low-end ASIC chips working in parallel, to arrive at high-end 28nm and 20nm ASIC chips we see today. Power consumption per chip now exceeds that of high-grade gaming GPUs and supercomputer CPUs and accelerator cards (Table 1).

Today, mining is often conducted by industrial mining farms, with densely packaged hardware that becomes outdated within months of coming online as electricity costs exceed profits. The rapidly evolving power density of mining clusters coupled with their short life cycle makes deployment in conventional air cooled data centers challenging.

2-PHASE IMMERSION

2-phase immersion thermal management, Figure 1 is viewed by many as a viable technology for meeting the power density and energy efficiency needs of the high performance computing market. Power densities up to 100 times higher than a typical air cooled server have been cooled this way with efficiency superior to direct water cooling [2]. Though new technical approaches [3] simplify its adoption, 2-phase immersion cooling has been slow to catch on in the risk-averse IT industry.
A primary hurdle to its adoption is the lack of energy-dense hardware. The cost and weight of fluid needed to fill a server chassis that is 95% air or more is prohibitive. Yet commodity hardware does not lend itself to density increases. Air cooling constrains the area of a motherboard and DIMMs, capacitors and connectors restrict the pitch between boards. For this reason, technology demonstrations to date have only been able to showcase the energy efficiency merits of the technology. Hardware sufficiently dense to showcase the density and resulting cost advantages has not been available except in specialized applications such as Bitcoin mining.

HEAT TRANSFER CHALLENGES

Keeping hardware cool, Figure 2, has proven to be a challenge right from the early days of custom Bitcoin mining hardware. FPGA chips used at or over the manufacturer specified limits down-clock when overheating, resulting in lower hash rates. A number of commercial mining hardware manufacturers were forced to make last minute changes, such as adding unplanned additional fans and heat sinks, before delivering hardware to customers [1]. While the market has somewhat matured in the meantime, this remains a continuing trend.

As they are approaching the limits of traditional cooling methods, the latest high powered ASIC devices face a new problem altogether. Power trains and mining chips are exceeding heat dissipated by commercially available computer hardware, creating a significant amount of engineering overhead, and hardware often ships with throttled performance due to thermal constraints. Removing these thermal bottlenecks, and reducing the associated engineering overhead, is central to the value proposition of 2-phase immersion, Figure 3.
or higher, the system can be run year-round with simple free cooling equipment. There are ports for two more condensers in each bath, allowing up to 75kW per tank or 225kW per rack. Baths of this density have already been demonstrated \[5\], as have boards packaged at up to 4kW per liter \[3\].

This scalability is central to the value proposition of 2-phase immersion. With the addition of more commodity dry tower hardware outside of the datacenter, the mechanical infrastructure can accommodate newer mining hardware with higher power densities. With boiling enhancement, emerging 1000W ASICS can be accommodated with the same fluid boiling temperature $T_f=60^\circ\text{C}$, allowing the facility to continue operating at $\text{PUE}<1.02$ in Hong Kong’s hot summers. This efficiency is central to the value proposition, for its direct effect on income and hardware lifespan.

The custom tank condensers use enhanced tubes borrowed from the refrigeration industry. Each is fed facility water directly from a four module dry tower sized for Hong Kong’s hot and humid weather (Table 3). The data center requires only a small air-conditioning unit for comfort cooling when people are present.

### IMMERSION-2 500KW BITCOIN MINING CLUSTER

Hong Kong, with its hot and humid climate, is a particularly challenging environment for data centers. With average Power Usage Effectiveness (PUE) of 2.52 \[4\] being the norm and pollution on the rise, companies need to be creative when it comes to IT cooling. The city is home to Immersion-2, and it comes as no surprise that its first large-scale implementation was a 500kW Bitcoin mining cluster.

Immersion-2 has a footprint smaller than a shipping container and is comprised of 20 conventional 19-inch racks, each housing 3 separate tanks or “baths”, as shown in Figure 4. The computational hardware is comprised of 80W boards originally cooled with a large aluminum heat sink to which two fans would be attached. When deployed in the insulated stainless steel immersion tanks, 92 boards are stacked vertically on a backplane with less than 10mm board-to-board pitch (Figure 5). To reduce space and fluid usage, boards are arranged in pairs with component sides facing each other and through-hole components (i.e. fan connectors) are left unpopulated during production. The weight of the original 92 heat sinks is roughly the same as the weight of an empty tank that contains about 2.75 liters of fluid per kW during operation.

### CONDENSER FOR HOT WATER COOLING

When the system was put into operation, the 92 boards and 4 power supplies in each tank dissipated about 8kW, but the condenser in each bath is designed to accommodate 25kW at Hong Kong’s hottest temperatures (Table 2), making it simple for the client to move through hardware generations. With possible water temperatures of $50^\circ\text{C}$ or higher, the system can be run year-round with simple free cooling equipment. There are ports for two more condensers in each bath, allowing up to 75kW per tank or 225kW per rack. Baths of this density have already been demonstrated [5], as have boards packaged at up to 4kW per liter [3].

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### TABLE 2: IMMERSION-2 CONDENSER SPECS AT 30 LITER / MIN WATER FLOW

<table>
<thead>
<tr>
<th>$Q$ (kW)</th>
<th>$T_{in}$ ($^\circ\text{C}$)</th>
<th>$T_{out}$ ($^\circ\text{C}$)</th>
<th>DP (PSID)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30.22</td>
<td>32</td>
<td>46.58</td>
<td>&lt;0.5</td>
</tr>
<tr>
<td>26.09</td>
<td>36</td>
<td>48.6</td>
<td>&lt;0.5</td>
</tr>
<tr>
<td>21.96</td>
<td>40</td>
<td>50.62</td>
<td>&lt;0.5</td>
</tr>
</tbody>
</table>

### TABLE 3: 1.5kW COOL HUNDREDS OF KILOWATT ON 30 JANUARY 2014

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Outdoor Temperature:</strong></td>
<td>24°C</td>
</tr>
<tr>
<td><strong>Relative Humidity:</strong></td>
<td>64%</td>
</tr>
<tr>
<td><strong>Water-In Temperature:</strong></td>
<td>39.9°C</td>
</tr>
<tr>
<td><strong>Water-Out Temperature:</strong></td>
<td>45.4°C</td>
</tr>
<tr>
<td><strong>Room Temperature:</strong></td>
<td>27.6°C</td>
</tr>
<tr>
<td><strong>Radiator Power:</strong></td>
<td>0.4kW</td>
</tr>
<tr>
<td><strong>Pump Power:</strong></td>
<td>1.1kW</td>
</tr>
</tbody>
</table>
FACILITY MANAGEMENT

Since the cooling system is operating isolated from the indoor environment, and the hardware is submerged in a constant temperature fluid, facility management and maintenance are reduced to a minimum. Without chillers or air handlers, there is no raised floor or dust that needs to be taken care of, and no static electricity that needs counter measures such as humidification. Environmental monitoring is carried out merely for statistical reasons. Instead vital water-in and out temperatures are monitored.

Though the hydrofluoroether working fluid is non-flammable, the buildings original water sprinkler system was kept in place for regulatory reasons. Should the sprinklers go off, the hardware is shielded from water, whereas the “outside” electrical wiring is well protected. Data centers using water sprinkler systems traditionally have to replace hardware after a catastrophic event. Ironically, after a fire in one of Google's data centers in 2010, it was reported that the sprinkler system did more damage than the fire itself [8].

POWER DELIVERY AND CABLING CHALLENGES

Some of the challenging aspects of the first installations were unrelated to cooling. In Bitcoin mining, power hungry ASICs are often restrained by thermal limitations in the power delivery portions of the system. Concerned about overheating, the primary manufacturer was unwilling to provide a backplane with sufficient power density. As with many other components for such an unusual application, the backplane was eventually designed and assembled in-house. Tasks like LAN wiring can become an art form when 92 Ethernet cables need to go in the smallest space possible without obstructing the way in and out for hardware. Standard RJ45 connectors suddenly seem oversized, leading to the idea of stacking boards face to face and removing the outer cable jacketing. It is these small take-home lessons that will prove invaluable when designing dense immersion hardware for HPC or cloud computing. Large-size connectors and cables don't seem to be the way forward, creating an opportunity for new connector and power solutions.

MATERIAL COMPATIBILITY

When Immersion-2’s predecessor went into operation in 2012, it consisted of 24 tanks totaling 70kW of FPGA hardware, and relied on power and USB cables for its operation. Each tank contained over 2 kg of off-the-shelf, polyvinylchloride (PVC)-clad USB cables. Upon dismantling the system the following year, it was observed that the dioctyl phthalate (DOP) oil used to plasticize the insulation had been extracted. This caused the cables to become thinner and more rigid in the vapor region above the fluid, although no failures were ever observed. 20 grams of DOP were absorbed by passive carbon filters inside each tank, and a similar amount of DOP was later
extracted from the fluid, with no degradation of boiling performance observed during the system's lifespan.

Full immersion has a positive impact on hardware reliability, due to hardware not being exposed to temperature fluctuations, uniform temperature across components, and elimination of the risk of electrostatic discharge in dry air. Without doubt, hardware failure will never be completely eliminated. Designing power supplies and backplanes without heat constraints can be less of a headache, but practical considerations need to be taken into account: the rules of hot swapping, redundancy, and systems protection are changed, due to the increased number of nodes.

HARD DRIVES AND IMMERSION

Another challenge has been building systems with the requirement for external storage. Traditional HDDs and immersion systems don’t get along well due to the mechanical structure of HDDs. In prototype setups the easy fix has been the use of solid state drives (SSDs) or leaving storage devices in adjacent air cooled enclosures. Experiments have been performed with keeping spinning hard drives in the head space of immersion tanks. The latest advances in HDD development may solve this challenge altogether. The first generation of hermetically sealed Helium-filled hard drives, aiming for increased capacity and speed, became available at the end of 2013.

THE PATH FORWARD (SUPERCOMPUTERS)

Though the hardware replacement cycle tends to be faster in the mining world, there are strong parallels between this market and the high performance computing market. Tianhe-2, the world’s fastest supercomputer, is comprised of 16,000 nodes, each containing two Intel® Xeon® Ivy-Bridge processors and three 250W Xeon® Phi co-processors, Figure 6. It consumes 17.8MW with an additional 6.4MW reported for cooling, resulting in a calculated PUE of 1.36. If this could be reduced to 1.05, Tianhe-2 would save 48 million kWhr per year with the potential to boost its 40th place standing in the Green 500 [6]. The resources that went into the development, validation and deployment of the closed loop water-air hybrid cooling system could have been avoided [7]. If conventions were established, Tianhe-3 might be deployed more rapidly with the same mechanical infrastructure and increased reliability. Immersion systems have already been built that show the functionality of high speed copper and optical interconnects in the immersion environment.
CHANGING THE RULES

Bitcoin is perhaps an ideal incubator for 2-phase immersion. With a basic system, developers have the luxury of quietly (both literally and figuratively) refining the technology on their own terms while exploring the reliability, serviceability and safety of the technology. It might seem unlikely that Bitcoin miners would be the ones to redefine thermal limits, but there’s an interesting precedent in the Bitcoin hardware industry. Professor Michael Bedford Taylor, who heads the Center for Dark Silicon at the University of California at San Diego (UCSD), is the author of a paper describing how Bitcoiners have disregarded all traditional rules of chip design, "leading to the development of machines that had orders of magnitude better performance than what Dell, Intel, NVidia, AMD or Xilinx could provide" [1]. Taylor, who was among the developers of the TILERA many-core computing architecture, says this path was “unheard of in modern times, where last-generation chip efforts are said to cost $100 million or more and the number of ASIC starts drop yearly.”

FULL DISCLOSURE

Alex and his company have no horse in the Bitcoin race, other than building systems and consulting for a client in the Bitcoin industry.

REFERENCES


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Relationship between Supply Flow Rate of Small Cooling Fans and Narrow Flow Passages in High-Density Packaging Electronic Equipment

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Katsuhiro Koizumi, Cosel Co., Ltd.

INTRODUCTION

Forced convection driven by cooling fans is widely used for removing heat from electronic equipment. A net cooling performance of the cooling fans is significantly dependent on a supply flow rate. Therefore, accurate prediction of cooling fan performance is extremely important in the thermal design of fan-cooled electronic equipment.

Generally, the supply flow rate of the fans is decided by the intersection of its pressure (\(\Delta P\)) – flow rate (\(Q\)) characteristic curve, which shows the relationship between the pressure rise at the fan (\(\Delta P\)) and the supply flow rate (\(Q\)), and a flow resistance curve of electronic equipment. In recent years, electronic equipment has become smaller and thinner while their performance increases and their functions become more complex. As a result, a lot of electrical devices are mounted in electronic equipment and the airflow passage in high-density packaging electronic equipment such as SMPS (Switched-Mode Power Supply), laser printers, lap top computers and servers becomes narrow and complex because gaps between mounted components generally become the flow passage. The pressure drop characteristic in such equipment may not be predicted easily by using well-known basic pressure drop characteristics [1]. In addition, several components such as heat sinks, capacitors and an enclosure are installed near the fan as shown in Fig. 1. This may affect the fan performance itself [2]. Several researchers have evaluated the fan performance when the components are mounted near the fan [3-5]. Therefore, the accurate prediction of the supply flow rate of the fan mounted in equipment is generally difficult. A further investigation of how the accurate supply flow rate of the fan mounted in electronic equipment is predicted rapidly and easily is required in order to shorten a design period while maintaining reliability of products.

With this as a background, this present study focuses on an investigation of the supply flow rate in electronic equipment [2, 6]. In this article, the relationship between the supply flow rate of the small axial fan which is mounted at the outlet of the enclosure and the flow passage area in electronic equipment is reported as an example of our investigation. To investigate a change of the fan’s flow rate in high-density packaging electronic equipment, a test enclosure which includes a test fan and an obstruction was prepared. The obstruction simulated electrical components in front of the fan. A change of

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the supply flow rate of the fan was measured while changing the obstruction and an opening area of an air inlet of the enclosure. We then explored the relationship between the supply flow rate and the flow pass area in order to obtain information for estimating the supply flow rate easily. Here, in this article, we investigated the supply flow rate when electronic equipment is mounted at the upstream side of test fans. Moreover we assumed that one axial fan is mounted at the air outlet of electronic equipment and fan-mounted electronic equipment operates in ambient atmosphere.

MEASUREMENT SYSTEM OF SUPPLY FLOW RATE

Figure 2 shows the schematic of our system for measuring the supply flow rate of the fan mounted in the test enclosure. It comprises an enclosure part that models the external enclosure of electronic equipment and the fan performance measurement system designed by Nakamura and Igarashi [7] based on JBMS-72-2003 [8]. This system is made up of an enclosure part with a test fan, a chamber part, an orifice, a valve, and an assist fan. Removal of the enclosure converts the system to the JBMS-72-2003 measurement system. The flow rate, \( Q \), is measured at the orifice. An analog valve and the assist fan can be used for controlling the static pressure in the chamber. As mentioned above, we investigated the supply flow rate of the fan when the fan-mounted electronic equipment operates in ambient atmosphere. Therefore in this study, the supply flow rate of the test fan when \( \Delta P_{ch} = 0 \) Pa was measured while changing the opening area of the narrow inlet and the obstruction. The static pressure difference \( \Delta P_{ch} \) was measured with a highly accurate differential pressure gauge (Shibata Kagaku, ISP-350, range; 0 – 500 Pa).

TEST FANS

Table 1 shows the details of the test fans. In this study, two types of 40 mm-scale fans, with a fan case of about 40 mm × 40 mm, were measured. Here, \( d_{tip} \) is a fan tip diameter and \( d_{hub} \) is a fan hub diameter. When we conduct the measurement, the voltage was adjusted to run the fan at a rated rotation speed, which was checked by a stroboscope.

TEST ENCLOSURE

Figure 3 (a) shows a schematic diagram of the test enclosure used in this study. Air flows into the enclosure from the surroundings through an inlet and exhausted to the atmosphere.
mounted in the enclosure. To simulate the actual electrical components in electronic equipment, a rectangular obstruction was mounted in front of the fan while changing the dimensions of the obstruction and the mounting position. Here, the ratio of flow passage area $A_{cl}$ divided by the fan flow area $A_{tip}$ is defined as the flow area ratio $\beta_{ob}$. Here, $A_{cl}$ is defined as:

$$A_{cl} = h_{en} \times w_{en} - h_{ob} \times w_{ob} \text{[mm}^2\text{]}$$ \hspace{1cm} (3)

$\beta_{ob}$ is expressed using the following formula:

$$\beta_{ob} = \frac{A_{cl}}{A_{tip}} \text{[-]}$$ \hspace{1cm} (4)

The value of $\beta_{ob}$ was set at between 0.9 and 2.6. The distance between the fan and the obstruction was changed from 5 mm to 40 mm.

**EXPERIMENTAL RESULTS**

Figures 5 and 6 show the change of the operating flow rate of each fan when the enclosure is mounted in front of the fan. Here, we evaluated the supply flow rate by using the non-dimensional flow rate ($Q / Q_{max}$). $Q_{max}$ is the flow rate when there is no enclosure and obstruction.

---

**TABLE 2: OPENING AREA OF ENCLOSURE INLET.**

<table>
<thead>
<tr>
<th>$\beta_{in}$</th>
<th>Hole Number</th>
<th>$A_{in}$ [mm$^2$]</th>
</tr>
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<tr>
<td>3.4</td>
<td>-</td>
<td>1681</td>
</tr>
<tr>
<td>1.6</td>
<td>64</td>
<td>804</td>
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<td>0.8</td>
<td>32</td>
<td>402</td>
</tr>
<tr>
<td>0.4</td>
<td>16</td>
<td>201</td>
</tr>
</tbody>
</table>

**TABLE 3: CONDITIONS OF OBSTRUCTION.**

<table>
<thead>
<tr>
<th>$\beta_{ob}$</th>
<th>$\beta_{ob}, h_{ob}$ [mm]</th>
<th>$x_{ob}$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.6</td>
<td>10</td>
<td>5, 9, 15, 20, 40</td>
</tr>
<tr>
<td>1.7</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>0.9</td>
<td>30</td>
<td></td>
</tr>
</tbody>
</table>

---

**FIGURE 5:** Supply flow rate of fan (A) versus distance of obstruction from fan inlet ($x_{ob}$) for different values of obstacle dimensions.

**FIGURE 6:** Supply flow rate of fan (B) versus distance of obstruction from fan inlet ($x_{ob}$) for different values of obstacle dimensions.
The supply flow rate of the fan decreases when $\beta_{ob}$ becomes smaller or the dimensions of the obstruction becomes bigger ($\beta_{ob}$ becomes smaller). The deterioration of the flow rate is caused by the increase of the pressure drop at the inlet and the obstruction. When $\beta_{in}$ or $\beta_{ob}$ becomes smaller, the flow pass becomes narrow. This causes the significant pressure drop. This tendency can be observed regardless of the type of the test fans.

We investigate the relationship between the supply flow rate and each opening area ratio. The deterioration of the flow rate becomes significant when $\beta_{ob}$ becomes smaller than 1.0. Even if $\beta_{ob}$ is larger than 1.0, the supply flow rate decreases when $\beta_{ob} = 0.9$. Here, Fig. 7 shows the relationship among the supply flow rate, $\beta_{in}$, and $\beta_{as}$. The supply flow rate approaches the maximum flow rate in the case that both $\beta_{ob}$ and $\beta_{in}$ becomes larger than 1.0. When either $\beta_{ob}$ or $\beta_{in}$ becomes smaller than 1.0, the deterioration of the supply flow rate is caused.

From these results, we can conclude that the supply flow rate of the fan is decreased according to the flow pass area. Of course, the fan's supply flow rate deteriorates when the number of the mounted components increases. The components increase the pressure drop in the enclosure regardless of their dimensions. However, when the flow pass area in electronic equipment becomes smaller than the fan’s flow area, the deterioration of the supply flow rate becomes significant. On the other hand, when the flow pass area in the enclosure is larger than 1.0, the deterioration of the flow rate can be inhibited.

Here, when the dimensions of the obstruction become larger, the distance between the fan and the obstruction $x_{ob}$ also affects the supply flow rate; the supply flow rate decreases when the position of the obstruction is near the fan. We previously reported that the fan performance itself deteriorates when the large obstruction is mounted in front of the fan [6]. In this case, the large obstruction affects the inlet flow to the fan and the fan performance itself may be deteriorated. In order to predict the correct supply flow rate regardless of the deterioration of the fan performance, the further investigation about the relationship between the obstruction near the fan and the fan performance should be additionally investigated.

### SUMMARY

In this article, the relationship between the supply flow rate of the axial fan in high-density packaging electronic equipment and the flow passage area is investigated experimentally by using the test enclosure. When the flow passage area in the enclosure becomes smaller than the fan's flow area, the supply flow rate of the fan significantly decreases. In order to avoid the deterioration of the supply flow rate, flow passage area should be larger than the fan's flow area. The obtained results may be used for estimating the supply flow rate of the fan easily in a design early stage.

For future study, an additional investigation of the pressure drop characteristic and the fan performance characteristic in high-density packaging electronic equipment may be meaningful in order to confirm the reproducibility of our results. Now, we are trying a development of detailed database of the pressure drop characteristic in high-density packaging electronic equipment. We will discuss whether our results can be applied regardless of the structure of electronic equipment. There is still room for developing methodology to apply our investigation to thermal design methods such as CFD analysis and flow and thermal resistance network analysis [9] in order to predict the supply flow rate of the fan in the enclosure rapidly and to shorten the thermal design period.

*Note: To view the nomenclature and subscripts for this article, see page 48.*

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Ultra-Thin Titanium Based Thermal Solution for Electronic Applications

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INTRODUCTION

As electronic system technology advances with continual increases in requirements leading to greater demand for higher power consumption, there have been growing challenges related to the design of thermal engineering and heat rejection technologies. The need for performance inevitably leads to operation of most electronic systems at the limits of the available thermal management technology. Today, one widely used method of thermal management in microprocessor and semiconductor chips is the heat pipe, due to its simplicity, reliability, practicality, and low-cost. However, conventional heat pipes have thermal transport and geometric limitations.

A thermal ground plane (TGP) is basically a flat heat pipe or vapor chamber type heat pipe. The TGP will expand upon the 2-phase cooling approach utilized in heat pipes and minimize issues relating to temperature gradients, increased weight, or added complexity. An important feature of the intended TGP technology is compatibility with existing chips and electronic devices without redesign of those systems; insertion of TGPs into these systems can provide new engineering margins yielding increased power use for the system, reduced operating temperatures, or reduced size of other components in the thermal management system.

PREVIOUS APPROACHES:

Early attempts to use TGPs for cooling electronics can be traced back to the 1970s [1]. In retrospect, the path forward has indeed been elusive. At first, simple cylindrical heat pipes were embedded in copper or aluminum plates. The heat pipes were bonded into holes drilled into the plate material. These devices were on the order of 10 mm thick. They were heavy, and thermal conductance was poor (~0.1 W/cm²-°C) due to the bondline thermal resistance between the plate and the heat pipes. In an attempt to provide a thinner TGP, thin-wall cylindrical heat pipes were flattened, and bonded between two aluminum facesheets. Although these TGPs were thinner (~4 mm overall thickness), thermal conductance remained poor, because of the bondline between the heat pipes and facesheets. Moreover, these TGPs were not tolerant to internal temporal working fluid pressure variations caused by temperature. The thin-wall flattened heat pipes tended toward their naturally rounded configuration at high operating temperatures associated with qualification tests (>100°C) or high altitude aircraft and spacecraft applications (>80°C). This resulted in flexing the circuit boards and associated solder joints.

Subsequently, vapor chambers [2] were developed. The vapor chambers were basically flat heat pipes that eliminated the individually embedded heat pipes of the previous designs, thereby eliminating the...
bondline between these heat pipes and facesheets. Vapor chamber TGPs typically used fine mesh wire screens or sintered powdered wicks leading to high thermal resistance to conduct heat into the wick material, at high heat flux levels, and excessive blockage of vapor space. For manufacturing reasons these vapor chambers were constructed of copper. However, copper has relatively poor structural properties, and this approach resulted in thick (~5 mm), heavy units. Moreover, copper has a higher coefficient of linear thermal expansion (CTE) relative to silicon (namely, 16.7 ppm/°C versus 2.6 ppm/°C, respectively) resulting in delamination of components and cracking in solder joints.

THE TITANIUM BASED APPROACH

The titanium based thermal ground plane (Ti-TGP) [3] was developed in response to advanced portable semiconductor and aerospace applications that are searching for ultra-thin, flat, lightweight and reliable thermal solutions.

Titanium has superior mechanical and electrical properties compared to copper or other conventional materials that have been used in heat pipes, Table 1. Titanium has: i) higher strength to weight ratio, ii) CTE closer to silicon which provides an improved thermal match between silicon based chips and the thermal solution, iii) higher corrosion resistance and, iv) compatibility with water.

Although the thermal conductivity of titanium is relatively lower than copper, the vapor chamber facesheets can be made 3 to 4 times thinner than a copper TGP. Therefore, the temperature drop for conducting heat into or out of the panel is not significant. For example, for a given budget thickness of a 4 mm TGP, the facesheet of titanium can be as thin as 500 μm, while the copper facesheet is required to be 1.5 mm thick in order to withstand the vapor pressure. That provides a 3 mm vapor space for the titanium TGP, while a copper TGP can only have a 1 mm vapor space. The temperature drop across a titanium facesheet would be ~2.5°C for a heat flux of 10 W/cm² compared with ~0.5°C for a copper TGP. This difference of ~2°C is a small penalty for the advantages of using titanium.

Generally, TGPs (flat vapor chambers) consist of two main structures; wick and back planes that are brought together, charged with working fluid, and sealed. The TGP cross-section and internal wick details are given in Figure 1. The overall thickness is calculated by adding the vapor space depth (VS_Depth) to the thickness of the wick and the back planes (T_WP and T_BP) as shown in Figure 1a. For a given TGP thickness budget, the vapor space depth is determined based on the required thermal metrics regardless of the material used. However, the thickness of the wick plane (excluding wick depth) and back plane are dependent on the material. These are the only two parameters that can be used to reduce the overall thickness. The high fracture toughness and yield stress of titanium compared to copper or aluminum allows both T_WP and T_BP to be reduced.

<table>
<thead>
<tr>
<th></th>
<th>Strength / Weight (kN-m/kg)</th>
<th>Yield Stress (MPa)</th>
<th>Density (kg/m³)</th>
<th>CTE (ppm/°C)</th>
<th>Thermal Conductivity (W/m-°C)</th>
<th>Electrical Resistivity (Ω·m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Titanium (99%)</td>
<td>288</td>
<td>830</td>
<td>4510</td>
<td>9.5</td>
<td>19</td>
<td>47.8E-08</td>
</tr>
<tr>
<td>Copper (OFHC)</td>
<td>24.7</td>
<td>70</td>
<td>8930</td>
<td>16.7</td>
<td>395</td>
<td>1.67E-08</td>
</tr>
<tr>
<td>Aluminum (606-T6)</td>
<td>214</td>
<td>400</td>
<td>2700</td>
<td>22.2</td>
<td>154</td>
<td>2.65E-08</td>
</tr>
</tbody>
</table>

**FIGURE 1**: (a) Ti-TGP operation (b) Micro-scale wicks (c) Coated nanostructure titania (NST).
and $T_{\text{wp}}$ to be as thin as 150 μm without suffering mechanical failure, resulting in an ultra-thin TGP (600 to 900 μm) that can withstand the internal pressure of the working fluid.

The multi-scale fabrication capability of titanium is another attractive feature. It can be either macro-machined using traditional machining processes or, micro-fabricated using micro-patterned masks and wet or dry etching to fabricate features having dimensions of 5 to 100 μm, Figure 1(b). Titanium can also be coated with nanostructure on the micro-scale wicks to generate a super hydrophilic surface (~0° wetting angle). Wettability of the titanium-water interface is an important factor in wicking velocity, which drives heat flow rate. The wick surfaces are enhanced by a chemical process to generate a highly porous surface called Nano-Structured-Titania (NST) [6], with pore features on the order of 100nm, Figure 1(c). The Ti-TGP is then hermetically sealed by micro-laser welding. [3,5]

The Ti-TGP has been customized for heat load capacities from 1W up to 1KW [3] with thermal conductance of 45 W/°C and 27 W/°C, based on average evaporator and condenser temperatures, for thickness of 5 and 3 mm, respectively. Figure 2 shows an ultra-thin and portless Ti-TGP, weighing 1 gram with thickness of 900 μm and overall dimensions of 50 mm X 8 mm. Figure 2a is a photograph of the plan view, whereas Figure 2b shows the edge view of the actual TGP.

Another attractive feature of titanium is scalability and conformality that allows the Ti-TGP to be designed and fabricated into predefined 2D and 3D shapes (Figure 3).

**FUTURE DIRECTIONS**

A new generation of semiconductor and micro-electronics devices are operating at higher power densities and consequently generating higher heat flux ($\geq 100 \text{ W/cm}^2$) capability. These applications are beyond the heat load capacity of current passive flat thermal ground plane solutions.

The future of advanced thermal management is moving toward active solutions to push the heat flux limit beyond passive solutions. Adaptive ultra-thin titanium-based active piezoelectric driven solutions that are able to extend the heat flux limit up to 150 W/cm² are currently being developed. Thermal solutions for SiC GaN MMIC are also currently being pursued with the aim of rejecting 1.2 kW/cm² heat flux from the GaN MMIC.

In summary, a range of thermal solutions will be available to accommodate a wide range of power and heat flux requirements.

**REFERENCES**


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Btech Corporation  
8395 Greenwood Dr. Longmont, CO 80503 US  
303-652-6448; www.btechcorp.com; Jay@btechcorp.com  

Cambridge Nanotherm Limited  
Homefield Road, Haverhill, Suffolk CB9 8QP, UK  
+44 (0)1440 765 520; Fax: +44 (0)1440 765 527  
info@cammarno.com; www.cammarno.com  

Capesym, Inc.  
6 Huron Drive, Suite 1B, Natick, MA 01760, US  
508-653-7100; Fax: 508-653-7155  
www.capesym.com  

Capovani Brothers Inc.  
704 Prestige Parkway, Scotia, NY, 12302, US  
518-346-0815; Fax: 042-862-9815; www.apack.net  

Caliente LLC  
1501 E. Berry St., Fort Wayne, IN 46803, US  
260-426-3800; Fax: 260-426-3838  
www.CalienteLLC.com; info@CalienteLLC.com  

Celsia Inc.  
3287 Kifer Road, Santa Clara CA, 95051 US  
408-577-1407; Fax: 408-577-1983  
gmeyer@celsiatechnologies.com  

Cambridge Nanotherm Limited  
Homefield Road, Haverhill, Suffolk CB9 8QP, UK  
+44 (0)1440 765 520; Fax: +44 (0)1440 765 527  
info@cammarno.com; www.cammarno.com  

Celsia Inc.  
3287 Kifer Road, Santa Clara CA, 95051 US  
408-577-1407; Fax: 408-577-1983  
gmeyer@celsiatechnologies.com  
www.celsiatechnologies.com/index.asp
### Harley Thermal LLC

### Heatron, Inc.
3000 Wilson Ave., Leavenworth, KS 66048, US 913-946-1394; Fax: 913-651-5352; www.heatron.com

### Henkel
14000 Jamboree Road, Irvine, CA 92606, US 714-368-8220; Fax: 952-934-2375 2100 Hoffman Way, Minneapolis MN, 55303 US

### Hoffman
www.hoffmanonline.com 952-934-8220; Fax: 952-934-2375 2100 Hoffman Way, Minneapolis MN, 55303 US

### Hyphen
809 Wellington St. N., Kitchener, Ontario, N2G 4Y7 Canada 519-749-6622, 519-744-1298 www.hyphenservices.com, sales@hyphenservices.com

### Ice Cube, Inc.
141 Wilson Ave., Greensburg, PA 15601, US 724-837-7600; Fax: 724-837-6385 www.icecube.com

### Iceotope Research and Development Limited
Advanced Manufacturing Park, Technology Centre, Brunel Way, Castlefield, Rotherham, South Yorkshire, S60 5WD, UK +44 114 254 1337; Fax: +44 114 254 1201; www.iceotope.com

### Indium Corporation
34 Robinson Road, Clinton, NY 13323, US 800-4-INDIUM; Fax: 800-221-5759; www.indium.com; askus@indium.com

### Insulfab
600 Freeport Parkway, Suite 150, Coppell, TX 75019, US 800-4-INDIUM; Fax: 800-221-5759; www.indium.com; askus@indium.com

### INTEGRATED Engineering Software
220-1821 Wellinton Ave., Winnipeg, Manitoba R3H 0G4, Canada 204-632-5636; Fax: 204-633-7780 www.integratedsoft.com; info@integratedsoft.com

### inTEST Thermal Solutions
41 Hampden Road, Mansfield, MA US 781-688-2300 sales@inTESTthermal.com; www.inTESTthermal.com

### Intermark US
1310 Tully Road, Ste. 117, San Jose, CA 95122, US 408-971-2035; Fax: 408-971-6033 sales@intermark-Us.com; www.intermark-US.com

### Ironwood Electronics
1335 Espanola Dr., Eagan, MN, 55121, US 952-229-8200; Fax: 952-229-8201 www.ironwoodelectronics.com; info@ironwoodelectronics.com

### Item Media - Electronics Cooling
1000 Germantown Pike, F-2, Plymouth Meeting, PA 19462 US; 484-688-0300; Fax 484-688-0303 www.electronics-cooling.com

### Jones Tech PLC
No. 3 Dong Huan Zhong Lu, BDA, Beijing 100176, China; +86 10 6786 2636; Fax: +86 10 6786 0291 JMC Products 10315A Metropolitan, Austin, TX 78758, US 512-834-8868; Fax: 512-834-8868 www.jmcproducts.com

### Kaneka Americas
6161 Underwood Road, Pasadena, TX, 77507, US Yukari Tanimoto; 832-741-3858 www.kaneka.com

### Kunze Folien GmbH
Rafflesiannalae 12a, Oberhaching, Bavaria, 82041, Germany; +49 89 666682-0; Fax: +49 89 666682-10; www.kunze-folien.com; sales@kunze-folien.com

### Laird Technologies
3481 Rider Trail South, Earth City, MO, US 63045, 636-898-6000; Fax: 636-898-8100; www.lairdtech.com

### LCR Electronics
9 South Forest Ave., Norristown, PA 19401, US 610-278-0840; Fax: 610-278-0935 www.lcr-inc.com

### Lytron
55 Dragon Court, Woburn, MA 01801, US 781-933-7300; Fax: 781-935-4529; www.lytron.com

### Malico
No. 5, Ming Lung Road, Yangmei 32663, Taiwan 886-3-4728155, ext. 1616; Fax: 886-3-4725979 inquiry@malico.com.tw; www.malico.com.tw

### Marian Inc.
1011 E. St., Clack Street Indianapolis, IN 46202 US 317-638-6525; Fax: 31-638-8684 www.marianinc.com

### Material Innovations Inc (MII)
15801 Chemical Lane, Huntington Beach, CA 92649, US 714-373-3070; Fax: 714-373-3091 www.matinnovations.com

### Maxx Technology, LLC
8270 S. Kyrene Rd., Suite 165, Tempe, AZ 85284 480-704-0284; Fax: 480-718-7390 www.maxxtechnology.com

### MC-21 Inc. (Metallic Composites for the 21st Century)
5100 Convair Drive, Carson City, NV 89706-0425, US 775-841-7112; Fax: 775-841-7112 www.mc21inc.com

### Mechanical Solutions Inc
2785 NW Uphur St., Portland, OR 97210, US 503-384-2965 www.msmthermal.com; ko@msmthermal.com

### Mechatronics
8152 30th Ave. SE, P.O.Box 5012, Preston, WA 98055; US; 800-453-4566; Fax: 425-222-5155 www.mechatronics.com

### MEN Micro, Inc.
24 North Main St., Ambler, PA 19002, US 215-542-9575; Fax: 215-542-9577; www.menmicro.com; sales@menmicro.com

### Mentor Graphics Corporation
Mechanical Analysis Division, 300 Nickerson Road, Marlborough, MA 01752, US; 800-547-3000 www.mentor.com/mechanical

### Mersen
317-638-6525; Fax: 636-898-6100; www.heatmanagement.com; inquiry@malico.com.tw; www.malico.com.tw

### MEnsor
374 Merrimac Street, Newburyport, MA 1950, US; 503-384-2965 www.msmthermal.com; ko@msmthermal.com

### MH&W International Corporation
575 Corporate Drive, Mahwah, NJ, US 07430-2300 201-891-8800; Fax: 201-891-0625 www.mhw-thermal.com

### Micropelt GmbH
Emmy-Noether-Strasse 2, Freiburg 79110, Germany; +49 761 1563370; Fax: +49 761 15633721 www.micropelt.com

### Microsanj LLC
3247 Kifer Road, Santa Clara, CA 95051 US; 408-256-5255; Fax: 408-765-7399; www.micropelt.com

### Minteq International, Inc.
Pyrogenics Group, 640 N. 13th St., Easton, PA 18042 US 610-250-3398; Fax: 610-250-3321; www.pyrogenics.com
<table>
<thead>
<tr>
<th>Company Name</th>
<th>Address</th>
<th>Phone Numbers</th>
<th>Website Links</th>
</tr>
</thead>
<tbody>
<tr>
<td>SNS Cooling Technology Inc.</td>
<td>90 Melford Drive, Unit #4, Scarborough, Ontario M1B 2A1, Canada</td>
<td>416-738-7797; Fax: 905-317-5305; <a href="http://www.snscoolingtech.com">www.snscoolingtech.com</a>; <a href="mailto:sales@snscoolingtech.com">sales@snscoolingtech.com</a></td>
<td></td>
</tr>
<tr>
<td>sp3 Diamond Technologies, Inc.</td>
<td>19605 Wyatt Drive, Santa Clara, CA 95054, US</td>
<td>408-492-0630; Fax: 408-492-0633; <a href="http://www.sp3diamondtech.com">www.sp3diamondtech.com</a>; <a href="mailto:info@sp3diamondtech.com">info@sp3diamondtech.com</a></td>
<td></td>
</tr>
<tr>
<td>Spectra-Mat, Inc.</td>
<td>180 Westgate Drive, Watsonville, CA 95776, US</td>
<td>831-722-4116; Fax: 831-722-4112; <a href="http://www.specsm.com">www.specsm.com</a></td>
<td></td>
</tr>
<tr>
<td>TA Instruments</td>
<td>159 Lukens Dr., New Castle, DE 19720, US</td>
<td>302-427-4000; 734-342-4900; Fax: 734-342-0931; <a href="http://www.thermoelc.com">www.thermoelc.com</a>; <a href="mailto:sales@thermoelc.com">sales@thermoelc.com</a></td>
<td></td>
</tr>
<tr>
<td>Timtronics</td>
<td>35 Old Dock Road, Yaphank, NY 11980 US</td>
<td>631-345-6509; Fax: 631-775-4023; <a href="mailto:info@timtronics.com">info@timtronics.com</a>; <a href="http://www.timtronics.com">www.timtronics.com</a></td>
<td></td>
</tr>
<tr>
<td>TTM Co., Ltd.</td>
<td>700 Gwanpyeong-dong, Yuseong-gu, Daejeon-si, Korea; +82-31-888-9258</td>
<td>Fax: +82-42-935-3752; <a href="http://www.coolttm.com">www.coolttm.com</a>; <a href="mailto:ttm@coolttm.com">ttm@coolttm.com</a></td>
<td></td>
</tr>
<tr>
<td>Universal Air Filter Company</td>
<td>Sauget, IL, 618-271-7300; <a href="http://www.uaf.com">www.uaf.com</a></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Verotec Inc.</td>
<td>33 Bridge St., Pelham, NH 03053, US</td>
<td>603-635-5199; Fax: 603-635-5299; <a href="http://www.verotec.us">www.verotec.us</a>: <a href="mailto:info@vettecorp.com">info@vettecorp.com</a></td>
<td></td>
</tr>
<tr>
<td>Wayfield-Vette</td>
<td>33 Bridge St., Pelham, NH 03076, US; 603-635-2800; Fax: 603-635-1000</td>
<td><a href="http://www.wayfield-vette.com">www.wayfield-vette.com</a></td>
<td></td>
</tr>
<tr>
<td>Wavelength Electronics Inc.</td>
<td>51 Evergreen Drive, Bozeman, MT, 59715, US</td>
<td>406-587-4911; Fax: 406-587-4911; <a href="http://www.teammwavelength.com">www.teammwavelength.com</a>; <a href="mailto:sales@teamwavelength.com">sales@teamwavelength.com</a></td>
<td></td>
</tr>
<tr>
<td>Waypoint Thermal Management, Inc.</td>
<td>150 River Road, Suite I-4A, Monroville, NJ 07045, US; 908-672-7362</td>
<td>Fax: 973-257-8999; <a href="http://www.waypointmanagement.com">www.waypointmanagement.com</a>; <a href="mailto:sales@waypointmanagement.com">sales@waypointmanagement.com</a></td>
<td></td>
</tr>
<tr>
<td>Wolverine Tube Inc. - MicroCool Division</td>
<td>100 Market St. NE, Decatur, AL 35601, US</td>
<td>256-580-3530; Fax: 256-580-3502; <a href="http://www.microcooling.com">www.microcooling.com</a></td>
<td></td>
</tr>
<tr>
<td>YS TECH US</td>
<td>12691 Monarch St., Garden Grove, CA, 92841, US</td>
<td>714-379-1400; Fax: 714-379-1010; <a href="http://www.ystechus.com">www.ystechus.com</a></td>
<td></td>
</tr>
<tr>
<td>Zalman Tech Co., Ltd.</td>
<td>#1037 Daeryoung Techno Town III, 448 Gusan-dong, Gumiunchon-gu, Seoul, 153-803, Korea; +82-2-2107-3232; Fax: +82-2-2107-3322; <a href="http://www.zalman.co.kr">www.zalman.co.kr</a></td>
<td></td>
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</tr>
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</table>
The Products & Services Index contains more than 50 categories to help find the products and services you need. Details of all the suppliers listed within each category can be found in the company directory, starting on page 40. To learn how to be included in this directory, e-mail editor@electronics-cooling.com.

Adhesives
- 3M Electronics Markets Materials Division
- Arlon Inc.
- Dow Corning Corporation
- Dymax Corporation
- Ellsworth Adhesives
- Henkel
- Insulfab
- Marian Inc.
- NAMICS Technologies, Inc.
- NuSil Technology
- ResinLab
- Rogers Corporation
- Timtronics

Advanced Materials
- Btech Corporation
- Element Six
- ERG Aerospace Corporation
- Hyphen
- MC-21, Inc. (Metallic Composites for the 21st Century)
- Indium Corporation
- Minteq International, Inc.
- Mintres BV
- Momentive Performance Materials
- NuSil Technology
- sp3 Diamond Technologies Inc.
- Surmet Corporation
- Taica Corporation

Air Conditioners
- Aqua Product Company
- Aspen Systems Inc.
- Caliente, LLC
- EIC Solutions Inc.
- Hoffman
- Ice Qube, Inc.
- MovinCool
- Pfannenberg
- Rittal Corporation
- TECO ThermoElectric Cooling America Corp.
- Vortec, an ITW Company

Air Filters
- Universal Air Filter Company

AlSiC Component
- CPS Technologies
- MC-21 Inc.

Baseplates
- CPS Technologies
- MC-21 Inc.
- Spectra-Mat Inc.
- Wolverine Tube Inc. - MicroCool Division

Blower/Fan Accessories
- Aavid
- AMCO Enclosures
- Caliente, LLC
- Cooltron Industrial Supply

Blowers
- Aavid
- Allied International
- Amerigon
- Ashland Electric Products Inc.
- Delta Electronics
- EAO Ltd.
- ebm-papst Inc.
- JARO Thermal
- JMC Products
- Nidec America Corporation
- Nuventix
- OLC Inc.
- Rosenberg USA Incorporated
- Sanyo Denki America Inc.
- SEPA EUROPE GmbH
- SUNON Inc.
- Waypoint Thermal Management, Inc.

Bonding
- Dymax Corporation
- Ellsworth Adhesives
- Rogers Corporation
- Scheugenpflug, Inc.

Chillers
- Aavid
- AMS Technologies
- Aqua Product Company
- Capovani Brothers
- Lytron
- Pfannenberg
- Rittal Corporation
- Thermokinetics Corp.

Circuit Assembly Materials
- Indium Corporation
- Nexlogic Technologies Inc.
- SinkPAD Corporation

Cold Plates
- Aavid
- Advanced Cooling Technologies Inc.
- Advanced Thermal & Environmental Concepts (ATEC)
- AMS Technologies
- Amulair Thermal Technology, Inc.
- Aspen Systems Inc.
- Baknor
- Celisa Inc.
- Delta Engineers

Coolers
- EPAC
- inTEST Thermal Solutions
- Lytron
- Malico Inc.
- MaxQ Technology LLC
- Minteq International, Inc.
- Niagara Thermal Products
- Parker Hannifin - Precision Cooling Systems Division
- Sapa Extrusions - North America
- Smart Heatsinks
- SNS Cooling Technology Inc.
- Summit Thermal System Co., Ltd.
- TECO ThermoElectric Cooling America Corp.
- Thermshield LLC - Division of Niagara Thermal Products
- Vette Corp.
- Wakefield-Vette
- Wolverine Tube Inc. - MicroCool Division

Composites
- Jones Tech PLC
- Material Innovations Inc. (MII)
- Spectra-Mat Inc.

Connectors
- Advanced Interconnections Corp.
- Indium Corporation
- Ironwood Electronics
- LCR Electronics
- Parker Hannifin Corporation

Connectors, Cable
- Amphenol Industrial Operations
- Aries Electronics
- Staubli Corporation

Cooling Courses/Seminars
- Alpha Novatech Inc.
- APACK Inc.
- Caliente, LLC
- CoolJag
- Delta Engineers
- Dynatron Corporation
- EXAIR Corporation
- MovinCool
- Nextreme Thermal Solutions
- Noctua at
- Parker Hannifin - Precision Cooling Systems Division
- TECO ThermoElectric Cooling America Corp.
- TE Technology Inc.
- Vette Corp.
- Vortec, an ITW Company
- Zalman Tech Co., Ltd.

Couplings
- Staubli Corporation

Education Courses/Seminars
- AERO NAV Laboratories
- Cradle
- Future Facilities Inc.
- Package Science Services LLC
### Enclosures
- AMCO Enclosures
- Aitech Defense Systems
- Chatsworth Products, Inc.
- Curtiss-Wright Controls Electronic Systems
- Elma Electronic, Inc.
- Extreme Engineering Solutions
- Hoffman
- Material Innovations Inc. (MII)
- MEN Micro, Inc.
- Pentair Equipment Protection
- Rittal Corporation
- STEGO, Inc.
- Youthen Technology Co., Ltd.

### Epoxy
- Ellsworth Adhesives
- NAMICS Technologies, Inc.
- Resinlab
- Rogers Corporation
- Timtronics

### Fans
- Ashland Electric Products Inc.
- DegreeC, Degree Controls Inc.
- Orion Fans

### Fan Controllers
- Cooltron Industrial Supply
- Gardtec incorporated
- Ice Qube, Inc.
- Mechatronics
- Orion Fans
- Pfannenberg
- Qualtek Electronics Corp.
- Rosenberg USA, Inc.
- STEGO, Inc.
- Universal Air Filter Company

### Fan Trays
- Ashland Electric Products, Inc.
- Delta Electronics
- EBM-papst Inc.
- Gardtec Incorporated
- Mechatronics
- Nidec America Corporation
- NMB Technologies
- ORION Fans
- Sanyo Denki America Inc.
- STEGO, Inc.
- Verotec Inc.
- Waypoint Thermal Management, Inc.

### Fans
- Aavid
- ALLIED International
- AMETEK Rotron
- AMCO Enclosures
- Ashland Electric Products Inc.
- Beijing Deepcool Industries Co. Ltd
- Cofan USA
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- Cooltron Industrial Supply
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- EBM-papst Inc.
- JARO Thermal
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- Mechatronics
- Nidec America Corporation
- Noctua.at
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- Pfannenberg Inc.
- Qualtek Electronics Corp.
- Rittal Corporation
- Rosenberg USA Inc.
- Sanyo Denki America Inc.
- SEPA EUROPE GmbH
- STEGO, Inc.
- SUNON Inc.
- Verotec Inc.
- Waypoint Thermal Management, Inc.
- YS TECH USA
- Zalman Tech Co., Ltd

### Gap Pads & Fillers
- AIM Specialty Materials
- Alfaitec GmbH & Co.
- AOS Thermal Compounds
- Aqua Product Company
- Brady Corporation
- Caliente, LLC
- Cool Polymers Inc.
- Fujipoly America Corp.
- Insulcom Corporation
- Internmark USA Inc
- Jones Tech PLC
- Kaneka Americas
- Kunze Folien GmbH
- Laird Technologies
- Marian Inc.
- Polymer Science Inc.
- Stockwell Elastomerics Inc.
- The Bergquist Company
- Timtronics

### Heat Exchangers
- Capovani Brothers Inc.
- Caliente LLC
- Curtiss-Wright Controls Electronic Systems
- Delta Electronics
- ERG Aerospace Corporation
- Heatron, Inc.
- Ice Qube, Inc.
- Lytron
- Thermacore Inc.

### Heat Pipes
- Aavid
- Advanced Cooling Technologies Inc.
- APACK Inc.
- Baknor
- Caliente, LLC
- Celsia Inc.
- Concept Group Inc.
- Delta Engineers
- Enertron Inc.
- JARO Thermal
- Mersen
- Radian Thermal Products Inc
- Sapa Extrusions - North America
- Thermacore Inc.
- Thermshield LLC - Division of Niagara Thermal Products
- TTM Co., Ltd.
- YS TECH USA

### Heat Sinks
- Aavid
- Alexandria Industries
- Alpha Novatech Inc.
- Amulaire Thermal Technology, Inc
- APACK Inc.
- Baknor
- Beijing Deepcool Industries Co. Ltd
- Caliente, LLC
- Celsia Inc.
- Cinch Connectors
- Cambridge Nanotherm Limited
- Cofan USA
- CoolJag
- Cool Polymers Inc.
- CTS Corporation
- Dynatron Corporation
- Element Six
- Enertron Inc.
- EPAC
- ERG Aerospace Corporation
- Fischer Elektronik GmbH & Co. KG
- Graftech International
- Heatron, Inc.
- Ironwood Electronics
- JARO Thermal
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- Malico Inc.
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- Thermacore Inc.
- Thermshield LLC - Division of Niagara Thermal Products
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- Youthen Technology Co., Ltd.
- YS TECH USA

### Heat Spreaders
- 3M Electronics Markets Materials Division
- Advanced Cooling Technologies Inc.
- Caliente, LLC
- Celsia Inc.
- CPS Technologies
- Graftech International
- Internmark USA Inc.
- Kaneka Americas
- Kunze Folien GmbH
- MC-21 Inc.
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- Mintres BV
- Momentive Performance Materials
- Polymer Science Inc.
- SinkPAD Corporation
- Smart Heatsinks
- SNS Cooling Technology Inc.
- Spectra-Mat Inc.

### Heaters
- Caliente, LLC
- STEGO, Inc.

### Infrared Imaging
- FLIR Commercial Systems, Inc.
- OptoTherm Inc.
- Palmer Wahl Temperature Instruments

### Interface Materials
- AIM Specialty Materials
- Alfaitec GmbH & Co.
- AOS Thermal Compounds
Arlon Inc.
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Dow Corning Corporation
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AIM Specialty Materials
Alfatec GmbH & Co.

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Mentor Graphics Corporation - Mechanical Analysis Division
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Mentor Graphics Corporation - Mechanical Analysis Division
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Concept Group Inc.

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Capovani Brothers Inc.
Degree Controls, Inc.
inTEST Thermal Solutions
Ironwood Electronics
Microsanj LLC
Palmer Wahl Temperature Instruments
Sensor Products Inc.
Teseq
Thermal Engineering Associates Inc.

Thermal Compounds
AOS Thermal Compounds
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Ellsworth Adhesives
Indium Corporation
Jones Tech PLC
Kaneka Americas
NAMICS Technologies, Inc.
ResinLab
Surmet Corporation
Timtronics
TTM Co., Ltd.
Zalman Tech Co., Ltd.

Thermal Design Services
Aavid
Advanced Thermal & Environmental Concepts
(ATEC)
Alpha Novatech Inc.
AMS Technologies
Baknor
CapeSym Inc.
Celsia Inc.
Chatsworth Products, Inc.
Cofan USA
Concept Group Inc.
CoolIT Systems Inc.
Daat Research Corp.
Degree Controls, Inc.
Enerdyne Solutions
Future Facilities Inc.
Harley Thermal LLC
Heaton, Inc.
Ironwood Electronics
Mechanical Solutions Inc.
Mentor Graphics Corporation - Mechanical Analysis Division
Mersen
Mintres BV
Package Science Services LLC
Radian Thermal Products Inc.
SinkPAD LLC
Smart Heatsinks
TDMG Inc.
Ten Tech LLC
TES International LLC
Thermal Engineering Associates Inc.
Thermal Solution Resources, LLC
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NOMENCLATURE

\[\begin{align*}
A & \text{area } [\text{mm}^2] \\
h & \text{depth } [\text{mm}] \\
d & \text{diameter } [\text{mm}] \\
h & \text{height } [\text{mm}] \\
l & \text{length } [\text{mm}] \\
Q & \text{volume flow rate } [\text{m}^3/\text{s}] \\
w & \text{width } [\text{mm}] \\
x & \text{distance } [\text{mm}] \\
\beta & \text{opening area ratio} \\
\Delta P & \text{pressure difference } [\text{Pa}] \\
\pi & \text{circular constant}
\end{align*}\]

SUBSCRIPTS

ch \text{chamber vs. surrounding} \\
cl \text{clearance around obstruction} \\
en \text{enclosure} \\
hub \text{fan hub} \\
in \text{enclosure inlet} \\
ob \text{obstruction} \\
s \text{strut} \\
tip \text{fan trip}
Alpha’s Next Generation Heat Sink
Custom or off-the-shelf.
Simple to complex.
Prototype to mass production.

Alpha’s New Push Pin Heat Sink Series!
Feature and design change for thermal & mechanical improvement.

Counterbore features added to fin side of the heat sink
Allows for push pin and spring attachment in height constrained applications.

Manufactured by MicroForging technology
Counterbores are created at the same time the fins are forged allowing for a lower cost than previous push pin heat sinks.

Wide catalog of standard parts and easy selection process
Extensive catalog of standard components and online design tools allow for simple selection of push pins and springs.
PF 40x40x28mm

- 40% Energy Saving
- 33% Low Vibration Reduced
- PWM Control Function
  - 40% Energy saving (100% duty)
  - 50% Energy saving (under 50% duty)